International Journal of Automotive Technology, Vol. 12, No. 5, pp. 639−644 (2011)
DOI 10.1007/s12239−011−0074−0

EFFECT OF A 2-STAGE INJECTION STRATEGY ON THE COMBUSTION AND FLAME CHARACTERISTICS IN A PCCI ENGINE

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(Received 12 March 2010; Revised 5 April 2011)

ABSTRACT−Recently, to reduce environmental pollution and the waste of limited energy resources, there is an increasing requirement for higher engine efficiency and lower levels of harmful emissions. A premixed charge compression ignition (PCCI) engine, which uses a 2-stage type injection, has drawn attention because this combustion system can simultaneously reduce the amount of NOx and PM exhausted from diesel engines. It is well known that the fuel injection timing and the spray angle in a PCCI engine affect the mixture formation and the combustion. To acquire two optimal injection timings, the combustion and emission characteristics of the PCCI engine were analyzed with various injection conditions. The flame visualization was performed to validate the result obtained from the engine test. This study reveals that the optimum injection timings are BTDC 60° for the first injection and ATDC 5° for the second injection. In addition, the injection ratio of 3 to 7 showed the best NOx and PM emission results.

KEY WORDS : Flame visualization, Emissions, Premixed charge compression ignition (PCCI), NOx, PM

1. INTRODUCTION

It is essential to reduce the amount of NOx and particulate matter (PM) emitted from diesel engines because they are harmful to the environment. Many ongoing studies focus on high pressure fuel injection systems (e.g., the common-rail system) and after-treatment techniques of diesel engines to reduce the emission of NOx and PM simultaneously; however, such a goal is undoubtedly difficult to attain.

Compared to conventional diesel engines, the premixed charge compression ignition (PCCI) diesel engine presents a new combustion concept with lower emissions and higher thermal efficiency (Kathi *et al*., 2002; Neely *et al*., 2004). This combustion mechanism is designed to achieve compression ignition by a fully premixed homogeneous charge instead of the diffusion combustion. A 2-stage injection strategy, including an early injection and a late injection, gives PCCI a leaner and more homogeneous mixture and a lower combustion temperature than the conventional diesel engines, which leads to a decrease in the NOx and PM levels. However, the PCCI engine still has some drawbacks such as the limited driving range, the difficult ignition timing control and the increase in CO and HC due to decreasing the in-cylinder pressure and temperature at the early injection timing. Such issues must be overcome for practical use.

Technically, the combustion principle of PCCI relies on a well-mixed lean mixture and no diffusion flame. Its combustion is initiated at the same time throughout the entire area of the combustion chamber. The homogeneous mixture restricts the emissions of PM, which is generated due to locally excessive fuel richness. The thermal efficiency is also expected to increase because the time loss decreases when the diffusion combustion is absent.

Hino Motors conducted many studies on a split injection strategy in a single cylinder direct injection engine to improve its emission performance (Yokota *et al*., 1997). Toyota Motor Co. studied a smokeless mechanism by reducing the temperature (Akihama *et al*., 2001). UNIBUS (UNIform BUlky combustion System) of Toyota is a technology using an early (BTDC 50°) and a late injection (ATDC 13°) (Hasegawa *et al*., 2003).

In this study, a 4-cylinder diesel engine was used to analyze the combustion and emission characteristics of the PCCI strategy with a 2-stage injection. In addition, the flame visualization technique was employed to observe the PM formation and to support the engine test analysis. Based on these data, we obtained the optimum injection timings and injection ratio for the two injections, which allowed a simultaneous decrease in NOx and PM in the PCCI engine.

2. METHOD OF MEASUREMENT

2.1. Engine Performance Test

Figure 1 shows the schematic of a common rail direct

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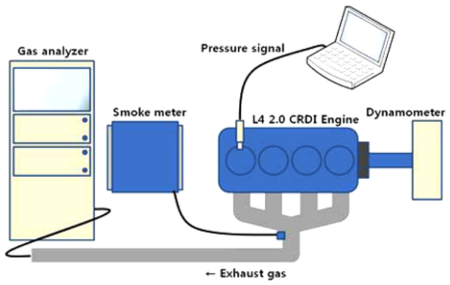

Figure 1. Schematic of the engine testing apparatus.

Table 1. Engine specifications.

Description	Specification
Engine type	4-valve, 4-cylinder
Bore × Stroke	83 × 92 mm
Displacement volume	1991 cc
Compression ratio	17.3
Fuel injection system	CRDi
Injection pressure	1600bar (max)
Number of injector hole	7

injection (CRDI) type engine apparatus. The detailed specifications of the engine are listed in Table 1. The dynamometer used for this experiment is a Meiden 220kW EC dynamometer. A 3600-pulse encoder is installed at the crank shaft, and a TDC sensor is installed at the cam shaft to control the fuel supply and to analyze the combustion.

The injection timing, injection pressure and injection quantity are controlled with an injector controller of TEMS Co. The coolant and the fuel are set at 82°C ± 2 and 40°C ± 0.5, respectively. The fuel flow rate is measured by a mass flow meter (CFM 010) from Micro motion Co., the Horiba MEXA-7100D is used to analyze the emissions, and the smoke density is obtained by an AVL smoke meter 415S.

In addition, a pressure sensor is installed in place of one of the glow plugs to measure the combustion pressure. The combustion pressure graph, the rate of heat release (ROHR) and the indicated mean effective pressure (IMEP) are measured by a combustion analyzer from MTS Co. in real time. All of the data is acquired as the average value of the measurements in 10 sec when the engine is in steady state.

Table 2. Experimental conditions.

Engine speed	1400 rpm	
Injection pressure	1000 bar	
BMEP	4 bar	
Injection timing	1st	BTDC 60°
	2nd	BTDC 5° ~ ATDC 10°
Injection method	2-stage injection (early & late)	

In this study, the fuel is injected in the middle of the compression stroke and again near the TDC for the 2-stage injection to make a sufficiently premixed mixture. Table 2 shows the experimental conditions to determine the characteristics of combustion and emission by the 2-stage injection. The engine has SCV, EGR and VGT and their influence on the engine performance will be tested in future studies.

2.2. Flame Visualization

Figure 2 shows the composition of the visualization engine system. With an elongated piston, the flame photographs are taken from the bottom of the combustion chamber and a window for visualization is attached at the crown of the piston. This visualization window is made of quartz, which has a 95% transparency. The window is designed to have a safety coefficient of 1.7 at the maximum pressure of 10 MPa.

The injector used is the same type as that of the engine test. The conventional diesel pistons have a reentrant shape, but the piston for this study has a bowl shape with a flat surface to ensure the stability of the quartz and to prevent the distortion of the flame images.

Image processing is a useful way to assess the combustion characteristics of the flame. In this study, we used MATLAB as the image processing tool to analyze the flame images captured by a high speed camera and to investigate the flame characteristics quantitatively. Some examples of image processing are shown in Figure 3. The root-mean-square (RMS) image processing method was used during the entire combustion period to compare the flame shape. This method helped us understand the

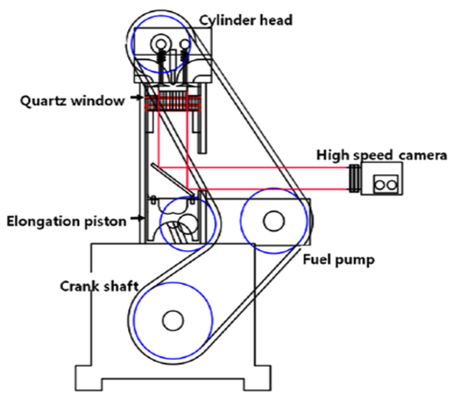

Figure 2. Schematic diagram of the visualization engine system.

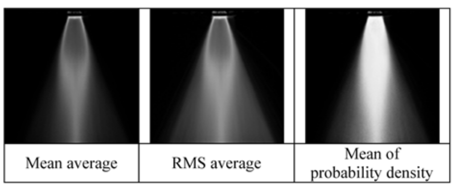

Figure 3. Examples of image processing.

periodicity of the flame while producing high-contrast images, whereas the mean averaging method could not easily produce a high contrast image because of the periodicity. Thus, we used RMS to calculate the flame area and the dispersion in the cylinder bowl. The principle of RMS image processing is based on the following equations (Oh *et al.*, 2008):

$$S_{mean}(x,y) = \frac{\sum_{k=1}^{n} S_k(x,y)}{n} \quad (1)$$

$$S_{RMS}(x,y) = \sqrt{\frac{\sum_{k=1}^{n} \{S_k(x,y) \times S_k(x,y)\}}{n}} \quad (2)$$

where $S_{mean}(x, y)$ is the arithmetic mean image obtained from the arithmetic average of the pixels, $S_{RMS}(x, y)$ is the root mean square image obtained from the value of the pixels in each frame, $S_k(x, y)$ is the foreground image after being corrected and denoised, and n is the number of images.

3. RESULTS AND DISCUSSION

First, the two injection timings were selected by analyzing the emissions and the BSFC characteristics at various injection timings with the dynamometer tests. Then, the optimal fuel injection mass ratios for the two timings were obtained by analyzing the combustion characteristics for different injection mass ratios. Finally, these optimal injection conditions were verified by the flame visualization.

3.1. Effect of Various Injection Timings on Emissions and BSFC Characteristics

Figure 4 and 5 show the effects of different injection timings on the characteristics of BS (Brake Specific) emissions and BSFC (Brake Specific Fuel Consumption) in the split injection mode, using the same injection mass for the first and the second injections. For the first test, the first injection timing was changed while the second injection timing was fixed at ATDC 5°. As the first injection timing approached the TDC, the BS emissions and the BSFC increased. We believed that if the first injection timing was close to the TDC, we could avoid the collisions of fuel droplets against the cylinder wall and increase the combustion temperature. Based on the BS emission and BSFC results, the best performance occurred when the first injection timing was set at BTDC 60°.

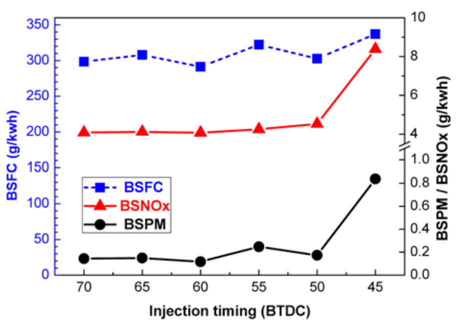

Figure 4. Effect of the first injection timing on the emission and BSFC.

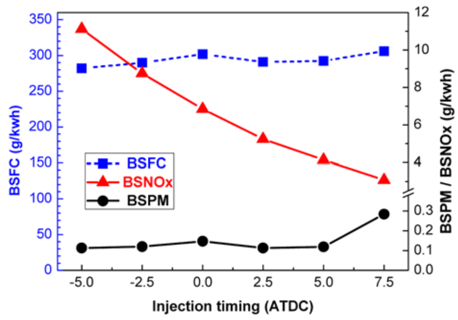

Figure 5. Effect of the second injection timing on the emissions and the BSFC.

With the first injection fixed at BTDC 60°, we varied the timing of the second injection for the second test. As the second injection timing was retarded, the NOx emission decreased, BSFC increased, and PM did not change much initially but also increased afterwards.

These results are attributed to the reduced combustion efficiency caused by a decrease in the combustion temperature. Based on the two tests, the first injection at BTDC 60° and the second at ATDC 5° represent the optimal timing for the split injection strategy without considering the injection ratio.

3.2. Effect of Injection Timing on Flame Characteristics

Figure 6 shows the effect of the second injection timing on the flame characteristics in the combustion chamber. When the second injection timings were set at BTDC 5° and TDC while the first injection timing was fixed at BTDC 60°, the diffusion flame occurred at the typical diesel combustion. However, the diffusion flame disappears when the second injection timing is retarded further after TDC. Thus, retarding the second injection timing increases the time for a more homogeneous mixture formation, resulting in a low PM generation.

The RMS image processing method developed for the research was also used to compare the combustion characteristics of the different second injection timings, as illustrated in Figure 7. The combustion flame is distributed homogeneously in the combustion chamber and PCCI combustion is noticeable when the second injection timing is retarded after TDC.

3.3. Effect of Various Injection Mass Ratios on Combustion and Emission Characteristics

In addition to the injection timing, variations in the injected fuel mass ratio also affect the characteristics of combustion, emissions and BSFC in the split injection strategy. To investigate these effects, the fuel mass portion of the first

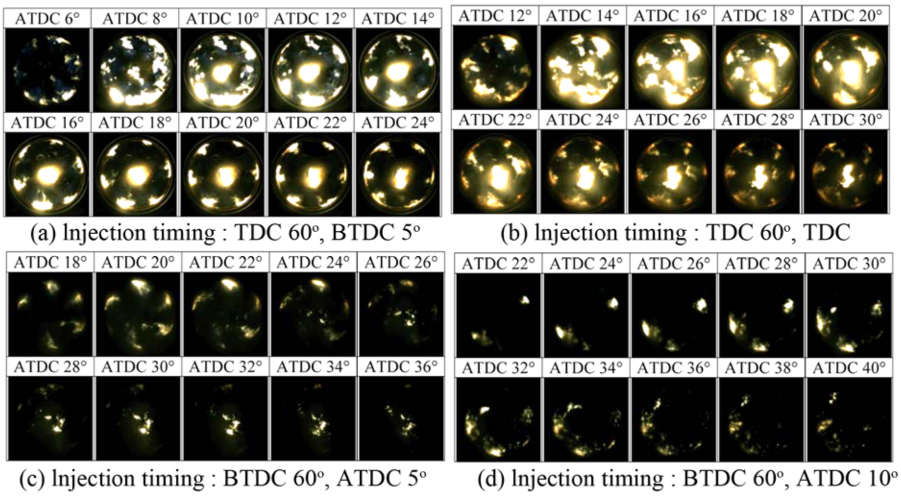

Figure 6. Flame characteristics as a function of the injection timing change.

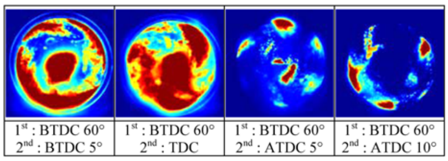

Figure 7. Comparison of RMS images between various second injection timings.

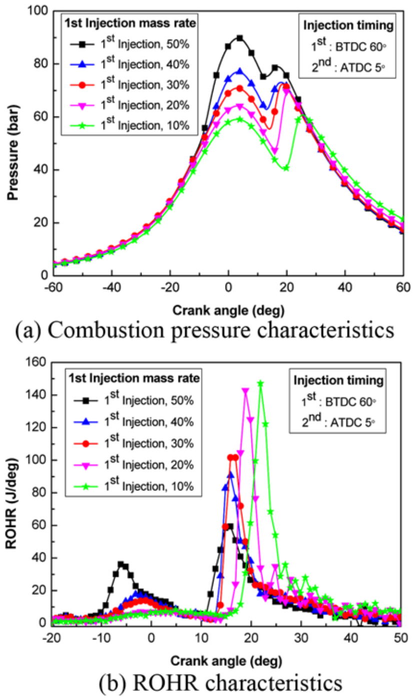

Figure 8. Effect of injection mass ratio on combustion pressure and ROHR.

injection was varied from 10% to 50% of the total fuel mass with the first and second injection timings fixed at the values determined above.

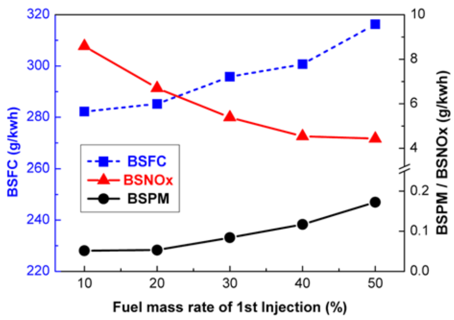

Figure 9. Effect of injected fuel mass ratio on emissions and BSFC.

Figure 8 shows the effects of the fuel mass ratio of the first to the second injection on the in-cylinder pressure and the ROHR as a function of the crank angles. A decrease in the fuel mass portion of the first injection reduced the cylinder peak pressure, which consequently caused a longer ignition delay for the second combustion, making the premixed mixture formation conducive to a highly efficient second combustion.

Figures 8 and 9 show that increased fuel mass portion of the first injection leads to a rapid combustion prior to TDC, resulting in a decrease in combustion efficiency. Moreover PM emission increased because the first combustion raised the in-cylinder gas temperature and shortened the ignition delay time before the second combustion occurred. On the contrary, the ROHR peaks of the second combustion became higher when the first injection mass portion was reduced, implying that an increase in the fuel mass portion of the second injection results in raising ROHR, causing the NOx emission to increase as shown in Figure 9.

Figure 9 also shows that decreasing the first injected fuel mass portion leads to a decrease in the BSFC and BSPM, except for the BSNOx emission. As discussed above, we believe that such effect occurred because the lower fuel mass portion of the first injection promoted a more

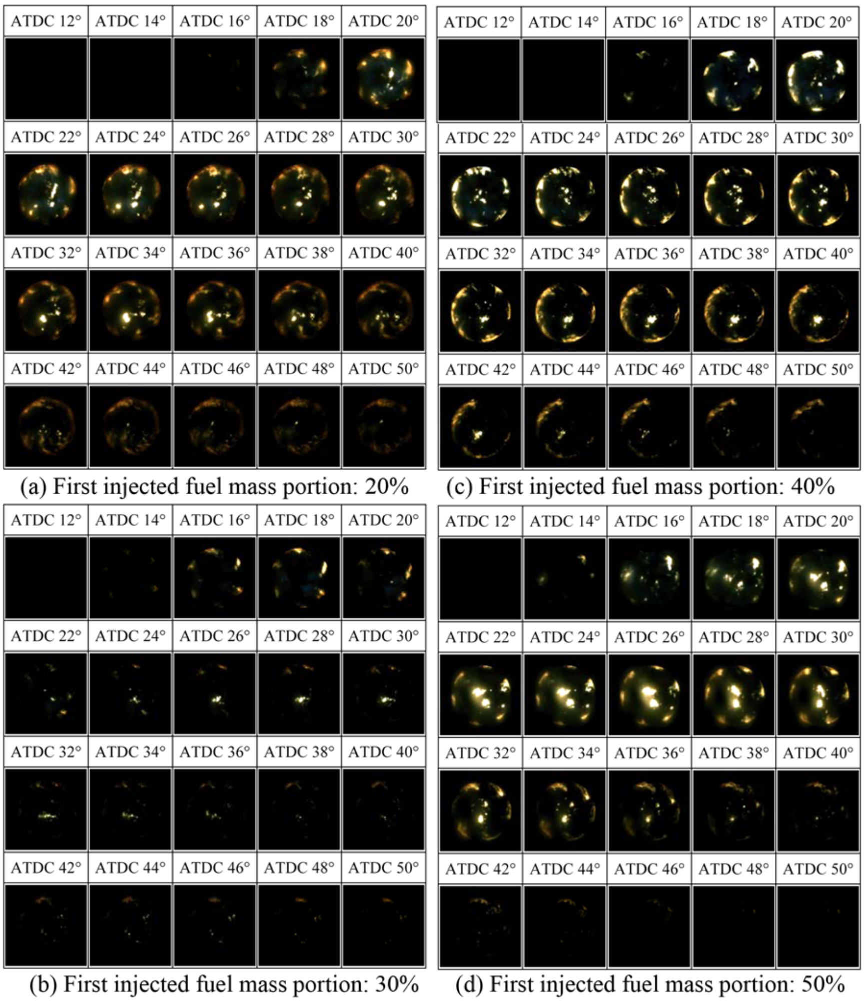

Figure 10. Flame characteristics as a function of the injection mass ratio.

efficient combustion; however, a higher NOx emission level appeared because of the increased second injection mass portion. In addition, when the fuel was injected at BTDC 60^{o}, the pressure and the temperature inside the cylinder were low enough that the fuel spray could penetrate well. Therefore, the first injection with a high fuel mass portion caused wall wetting and not enough air-fuel mixing time, causing PM emissions to increase. This effect can also be explained in terms of the flame images in Figure 10.

3.4. Effect of Injection Mass Ratio on Flame Characteristics

Figure 10 shows the effect of the injected fuel mass portion on the flame in the combustion chamber. This visualization indicates that when the first injection mass portion reduced, the ignition was delayed, and the combustion efficiency increased; however, the peaks of ROHR and NOx emissions also increased because the increased second injection mass portion was similar to the conventional diesel combustion. On the contrary, an increase in the first injection portion led to an increase in the ambient temperature, which reduces the ignition delay and the mixing time of the second injected fuel and air. As a result, the diffusion flame appeared and caused the PM emission to increase.

The first injection portion of 20% seems to have the brighter flame than that of 30%, representing a higher ROHR characteristic and a higher peak combustion temperature, which causes an increase in the NOx emission. At the first injection portion of 30%, neither the rapid combustion flame nor the diffusion flame appears in the combustion chamber attributed to the uniform mixture distribution. When the first injection portion was 40% and 50%, the diffusion flame that generates PM appears because of a short ignition delay and

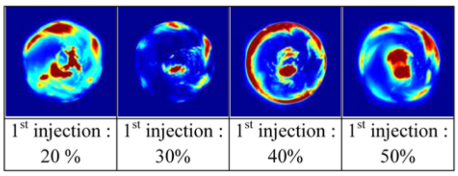

Figure 11. Comparison of RMS images according to the first injection mass portion.

a mixing time.

The results of combustion characteristics are confirmed by the combustion RMS images in Figure 11. Evidently, the first injection mass portion of 30% gives the most uniform mixture and the best combustion flame.

In conclusion, the optimal injection timings are BTDC 60° and ATDC 5° and the optimal first injection mass portion is 30% for the 2-stage injection strategy used in this research.

4. CONCLUSIONS

This study was to find the optimal injection conditions for a PCCI engine. Using a 4-cylinder engine and a visualization engine with a common rail direct injection system, we analyzed the effect of various injection conditions on the characteristics of flame, emissions and combustion performance. We found the followings:

(1) The NOx level increased with a retarded first injection timing because the combustion temperature increased. The high combustion temperature results in a decrease in the ignition delay for the second injected fuel. Therefore, poor mixture formation is a direct reason for a high PM emission. Based on the observation of both BSFC and BS emissions, BTDC 60° was chosen for the first injection timing in the 2-stage injection scheme.

(2) The combustion efficiency decreased with a low combustion temperature when the second injection is retarded. The second injection timing of ATDC 5° gives the best overall results.

(3) When the first injection mass decreased, the NOx emission increased with the increased ROHR. Based on the combustion and emission performances, the first injection mass portion of 30% was chosen for the best.

(4) From the flame images, the first injection mass portion of 30% appears to be the most suitable ratio because it has the lowest diffusion flame among all ratios and the first injected fuel has a minimal influence on the main combustion.

ACKNOWLEDGEMENT–This work was supported by the “Parts and Materials Technology Development Program”, “Non-Road 50kW Diesel Engine Development for Tier4 Project” funded by the Ministry of Knowledge Economy(MKE, Korea)”.

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