Combustion and Emission Characteristics in a Direct Injection Natural Gas Engine Using Multiple Stage Injection

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ABSTRACT

A direct injection natural gas engine has the possibility to achieve the fuel consumption and power corresponding to these of a diesel engine. The single cylinder test engine was equipped with a newly developed high-pressure electromagnetic injector. The possibility to stabilize the combustion by spark assist ignition and to reduce emissions, especially NOx, keeping high thermal efficiency was investigated by means of multiple stage injection with the test engine using the trial-fabricated electromagnetic injector. Moreover, the injected gas behavior in the combustion chamber was observed by means of laser shadowgraph method to investigate a mixture of air and fuel.

This research examines a method for producing a stratified fuel-air mixture suitable for ignition in the vicinity of a spark plug by splitting gas injection into two stages per cycle in addition to allowing us to visualize the flow of the fuel-air mixture until ignition.

1. INTRODUCTION

The introduction of natural gas engines as substitutes for diesel engines is one of many creative and effective methods of protecting our environment, and a direct injection natural gas engine has the possibility of ensuring the thermal efficiency and output characteristics equivalent to those of a typical diesel engine over a wide spectrum of loads and speeds. However, the earnest research on this type of engine, in both quality and quantity, is lacking when compared to pre-mixing type natural gas engines. (1)(2)

Initiation of combustion by the self ignition of natural gas method is difficult because of the high ignition temperature requirement. Because of that, a unique system for assisting the ignition process is necessary. (3)(4) This research examines the two-stage injection; a method for obtaining stable combustion by using spark ignition, which allows a wide range of options when setting spark conditions, such as timing and energy, as an ignition assistance process for direct injection natural gas engines.

In gas direct injection type combustion, the diffusion process of fuel greatly influences the combustion. In order to realize combustion stability over a wide range of loads, the fuel-air mixture needs to be stratified by utilizing the diffusion process of directly injected fuel so that a stable mixture suitable for ignition is formed in the vicinity of the spark plug regardless of the gas injection rate.

This research examines a method for producing a stratified fuel-air mixture suitable for ignition in the vicinity of the plug by splitting gas injection into two stages per cycle in addition to allowing us to visualize the flow of the fuel-air mixture until ignition. New measures to reduce NOx emissions must be devised because, in the stratified combustion, more NOx is emitted by locally produced excessively rich fuel-air mixtures, especially when at a low excess air ratio (λ) in comparison to uniform lean pre-mixing combustion. Accordingly, this research has been studied on measures for reducing NOx emissions in two-stage injection, while satisfying the minimum requirement for the stratification of the mixture to obtain a stable ignition.

2. EXPERIMENTAL METHOD

In order to stabilize ignition, it is necessary to form a fuel-air mixture with an excess air ratio (λ) of nearly 1 in the vicinity of the spark plug. (6) This research has been carried out to study a method where the fuel concentration of the ambient fuel-air mixture in the vicinity of the plug is increased by injecting a portion of the fuel into the cylinder at an early stage (hereafter termed the primary injection), so that the fuel-air mixture is easily stratified in the late stage injection for ignition (hereafter termed the secondary injection). This will give the mixture a distribution suitable for ignition in the cylinder by injecting fuel two times per cycle and adjusting the timing and rate of each injection.

Shioji et al. have investigated the influence of the nonuniformity of fuel-air mixtures on NOx concentration. Their calculations showed that the combustion of uniformly lean fuel-air mixtures emit lower NOx than nonuniform fuel-air mixtures. (7) As a result, in order to prolong the mixing time, the primary injection is performed just after the inlet valve is closed and the lean mixture is made uniform as much as possible before spark ignition. The secondary injection forms the mixture necessary for stable ignition (only in the vicinity of the plug), and then ignition takes place by spark. This injection method provides a higher probability of reducing NOx emissions in comparison to one-stage injection where the entire quantity of fuel for each cycle is injected at the same time. The two-stage injection is
realized by using an electromagnetic injector designed for high pressure natural gas that is sequentially actuated twice.

In addition, the state of the natural gas jets mixing until ignition has been visualized at the combustion chamber bottom by means of the shadowgraph method using an argon laser.

3. EXPERIMENTAL APPARATUS AND EXPERIMENTAL CONDITIONS

3.1 Experimental Apparatus

The test engine is a modified version of the single cylinder diesel engine shown in Table 1. The engine speed is kept constant by an electric dynamometer. The fuel injection quantity is controlled by varying the width of electrical pulses, with the λ signals of a monitoring exhaust sensor. Fig.1 shows the composition of the experimental apparatus. Standard 13A natural gas specified for city gas, is injected into the engine cylinder from the injection nozzle through the pressure regulator from a gas cylinder filled with gas at a pressure of 20 MPa.

Using a bottom-view type visualization device employing an observation mirror installed in the extended piston, gas jets can be observed from the piston side through a quartz glass window attached to the piston head. A reflecting mirror is stuck on the combustion chamber inside surface of the cylinder head. Fig.2 shows the optical system. The maximum output of the argon laser is 3.7 W and the images of injection are recorded by a high speed video camera.

NOx concentrations are measured by a CLD analyzer, CO by NDIR, and HC by HFID.

### Table 1 Single cylinder engine specification

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore X Stroke (mm)</td>
<td>108 X 115</td>
</tr>
<tr>
<td>Displacement Volume</td>
<td>1053 cc</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>12</td>
</tr>
<tr>
<td>Chamber type</td>
<td>Direct Injection</td>
</tr>
<tr>
<td>Intake valve timing</td>
<td>21° (BTDC)49° (ABDC)</td>
</tr>
<tr>
<td>Exhaust valve timing</td>
<td>56° (BBDC)14° (ATDC)</td>
</tr>
<tr>
<td>Cooling method</td>
<td>Water cooling</td>
</tr>
<tr>
<td>Nozzle</td>
<td>6X φ0.41 mm</td>
</tr>
<tr>
<td>Swirl Ratio</td>
<td>1.2</td>
</tr>
</tbody>
</table>

3.2 Combustion Chamber

A deep dish type combustion chamber was adopted to utilize a squish. The combustion chamber has an offset of 7.2 mm from the nozzle center and a compression ratio of 12. Fig.3 shows the plan and a cross section of the combustion chamber. The plan is the view from the piston top, and the cross section includes both the piston and cavity centers. The shape of the combustion chamber is a reentrant type and only the gas-injection directions are indicated by bold full lines in both figures. The swirl direction from the top is clockwise. (From the bottom, it is counterclockwise.)

To ensure that even a small fuel injection rate can produce a fuel-air mixture having a proper λ in the vicinity of the plug by using the collision of jets with the combustion chamber wall, the cavity is positioned as close as possible to the spark plug. Accordingly, as shown in Fig.3, the middle point of the spark plug is located 7.8 mm away from the nozzle (marked with ⧎) at a space angle of 60° between two injection directions.

3.3 High Pressure Electromagnetic Injector for Natural Gas

A compact electromagnetic injector for high pressure gas was trial-fabricated. It allows a wide freedom of choice in selecting the injection start timing and injection periods. Fig.4 shows the injector, which can operate with a maximum fuel gas pressure of 20 MPa and an engine speed of 3,000 rpm. The nozzle hole diameter is 0.41 mm and the number of holes is six. Since the other dimensions of the nozzle are nearly equal to those of a diesel oil injector, the head does not need to be
modified greatly.

3.4 Experimental Conditions

The air intake was fully open with wide-open throttle and the engine speed was kept at a constant 1,000 rpm. The cylinder’s internal pressure was sampled successively in each of 350 cycles to analyze combustion. For statistical evaluation of cyclic variation, the maximum cylinder pressure (Pmax) and the indicated mean effective pressure (IMEP) were used as the coefficient of variations (COV) defined as the standard-deviation/average. For instance, the COV of IMEP is expressed by COVIMEP.

The cycles that give the same IMEP during motoring are counted as misfire cycles. The ratio of the misfire cycles to the total measured cycles is defined as a misfire ratio (Rm).

The fuel-feed pressure (Pf) by which the flow rate becomes critical at Pmax in this experiment is given to be 14 MPa. Consequently, at a Pf higher than 14 MPa, the fuel injection quantity can be controlled by the needle opening period alone in any arbitrary injection timing regardless of cylinder pressures as long as both fuel temperature and Pf are constant. The Pf has been set to be a constant 15 MPa. A has been controlled so as to be between 1.3 and 3.7 and the swirl ratio at 1.2.

For the injection start timing of the two-stage injection, in order to prolong the duration for mixing fuel after the primary injection, the primary injection is started at -125° ATDC just after the close of the air intake valve and the secondary injection is started at either -30° ATDC or -15° ATDC.

Two-stage injection is abbreviated as SI (split injection). One-stage injection, where the full quantity is injected at the primary injection and has no secondary injection, is abbreviated as EI (early injection). One-stage injection, where the full quan-

![Fig. 5 Shadowgraph images of one-stage injection (LI)](image1)

![Fig. 6 Shadowgraph images of two-stage injection (SI)](image2)
tity is injected at the secondary injection and has no primary injection, is referred to as LI (late injection). In SI, the primary and secondary injection quantities have been divided to be usually 50 percent respectively, however, in some cases the injection rates were also changed.

The spark-discharging conditions of the igniter were set with an induction discharging duration of 3 ms constant and an spark energy of about 30 mJ, and for all the experiments the spark timing was to be MBT.

The images produced by the shadowgraph method were recorded at 4,500 pps by a high speed video camera, stored in a mass memory storage system, and image-processed by computer.

4. RESULTS AND CONSIDERATIONS

4.1 Observation by Laser Shadowgraph Method

Fig. 5 shows the shadowgraph images of the one-stage injection system at a fuel-feed pressure of 15 MPa. For comparison, however, those at 10 MPa are also shown. The swirl ratio is 1.2, the injection start timing -30° ATDC, and injection duration 4 ms. The swirl direction is counterclockwise. The spray angle is 13.5° at 10 MPa, and 14.7° at 15 MPa in the condition of -26° ATDC at an ambient gas pressure of 0.91 MPa. At 10 MPa, the boundaries of the jets with the ambient air in the downstream near the wall of the combustion chamber are not clear, and the jets drift in swirls. At 15 MPa, the boundaries between the jets near the nozzle and the ambient air is clear, but the mixing becomes active by the turbulence of jets due to the collision of the jets with the wall of the combustion chamber or observation window glass constituting the bottom surface of the combustion chamber. The tendency to mix with the ambient air in the downstream of the jets at 15 MPa was confirmed to be more remarkable than that at 10 MPa.

Fig. 6 shows the shadowgraph images in the two-stage injection at a fuel-feed pressure of 15 MPa. The swirl ratio is 1.2. The primary injection is started at -125° ATDC and the secondary injection at -30° ATDC. The injection duration is 2.6 ms for both injections. The swirl direction is counterclockwise.

For the primary injection, because the suction stroke takes place just after the air intake valve is closed, the ambient air pressure is less than the atmospheric pressure. It is suspected that water vapor by the temperature drop due to adiabatic expansion effect may be observed. The spray angle is 8.4° at -121° ATDC. Also the dim contrast of the jets indicates that the spatial differential of the density gradient is small in primary injection, because the contrast of the shadowgraph is proportional to the secondary differential of the gas density.

The flow in the combustion chamber continues from the completion of the primary injection to the start of the secondary injection. Further finer shape turbulence is produced at the beginning of the compression stroke, so that the uniformity of the fuel-air mixture increases.

In the secondary injection, the ambient gas pressure is 0.9 MPa at the start of the injection and 1.7 MPa at the end. The spray angle in the secondary injection is larger than that in the primary injection. It is 15.3° at -26° ATDC. The clear contrast of jets indicates that the spatial differential of the density gradient during the secondary injection is large. In the later period of the injection, turbulence-mixing actively occurs as a result of the collision of jets with the combustion chamber wall or the observation glass in the combustion chamber bottom.

4.2 Stabilization of Combustion by Two-Stage Injection

Fig. 7 shows the results of the influence of the split ratio, R1. The split ratio shows the ratio of the primary injection quantity to the total injection quantity. The influence was investigated under the condition that the primary injection quantity is larger than the secondary one in order to reduce NOx emission.

The primary injection start is at -125° ATDC, the secondary injection start is at -30° ATDC, and spark is at -30° ATDC. λ is 1.3 or 2, and Rs 1.2. When λ =1.3, no big difference is found. When λ =2, the most stable state occurs at R1=1/2. When λ =2, however, the secondary injection duration at R1=2/3 reaches the minimum injection duration of the injector because of the small injection quantity. The needle movement of the injector becomes unstable in this condition. As the result, the injection quantity fluctuates greatly. Accordingly, the latter experiments were basically carried out under the condition of R1=1/2 when λ is chosen to be 1.3 to 2.7.

Fig. 8 shows the changes in the COVimep, Rm, indicated thermal efficiency, and indicated effective pressure vs. λ. The split ratio of SI is 1/2. The values of COVimep is highest for EI, second for LI, and smallest for SI, that is, the two-stage injection gives the most stable combustion. In the case of λ > 2.1, combustion fluctuates so greatly that the misfire ratio rises sharply under all injection conditions. Indicated thermal efficiencies and indicated effective pressures are nearly the same in SI and LI but the thermal efficiencies in 1.6 < λ < 1.8 are slightly lower in EI.

Fig. 9 shows the typical indicated pressures and heat release rates at λ =1.8 and Rs=1.2. The split ratio of SI is 1/2. The spark timing of MBT is earliest for EI, second for SI, and latest for LI. In EI, mixing is promoted in between fuel injection and spark. As the result, the combustion start timing becomes earlier. Consequently, most of the heat release takes place before the top dead center of the compression stroke, so that the energy is not used effectively as output and the thermal efficiency decreases.

![Figure 7](image_url)
4.3 Exhaust Gas Characteristics of the Two-Stage Injection System

Fig. 10 shows the concentration of the emission gases vs. $\lambda$ at $Rs=1.2$. The injection timing is the same as Fig.8. The split ratio of SI is $1/2$. SI has nearly the same levels of emission gases as LI, while EI emits higher NOx concentration at $\lambda < 2$. In EI, the Pmax value is higher as shown in Fig.9. Most of the heat release occurs before the top dead center. Accordingly, the NOx concentration is high at $\lambda < 2$. In SI, the concentration of NOx is on nearly the same level as LI. NOx-reducing effect in this condition was not remarkably recognizable as shown in the figure. Accordingly, analysis was then carried out to focus on the possibility of reducing NOx by delaying the secondary injection start timing so that the spark in SI takes place with the same timing as that of LI.

4.4 NOx Emission Reduction by Two-Stage Injection

Fig.11 shows the indicated thermal efficiency, COVimep, and exhaust gas characteristics when the secondary injection start timing and spark timing is delayed in the range of $\lambda =1.3$ to 2.7. The conditions of LI, SI, SI2, and SI3 are shown in Fig.12. The split ratios were set at the maximum within COVimep<5%. In EI, no ignition takes place with the same spark condition, so EI is not shown in the figure.

SI2 remarkably reduces the NOx concentration, however, since the spark timing is after the top dead center, the thermal efficiency drops sharply and THC and CO concentration rise as the fuel-air mixture becomes lean. On the other hand, SI3 reduces NOx concentration by half in the range of $\lambda =1.3$ to 1.7 by putting spark timing forward to $-1^\circ$ ATDC from that of SI2. Also the combustion in SI2 is more stable than those in LI and SI.

Fig.12 shows the indicated pressures and heat release rates in LI, SI, SI2, and SI3. SI2 reduces NOx concentration because the combustion temperature is limited owing to its low heat release rate, but the thermal efficiency is lower because the heat releasing duration is longer. On the other hand, SI3 shows that the detrimental influence on the thermal efficiencies is small because the heat releasing patterns are similar to those of LI and SI although the heat release start timing is delayed.

Two-stage injection method has the capacity to easily vary the secondary injection timing and spark timing, because the secondary injection quantity in two-stage injection is less than the injection quantity in one-stage injection. Consequently, two-
stage injection can reduce NOx emission in the range of $\lambda = 1.3$ to 1.7 without lowering thermal efficiencies by the early primary injection and the retard of the secondary injection timing and spark timing.

5. CONCLUSION

This research was conducted to study the stabilization of combustion and the possibility of reducing NOx emissions by using two-stage injection in a direct injection natural gas engine. The visual analysis of gas jets, using the shadowgraph method, was also achieved. As a result, the following conclusions were reached:

(1) The spacial differential of density gradient in primary injection of two-stage injection is small and the uniformity of the fuel-air mixture increases by the fine turbulence produced after the compression stroke begins. The spacial differential of density gradient in the secondary injection is large and spray angle increases. The mixing between fuel and air is promoted by turbulence due to the collision of jets with the combustion chamber wall surfaces.

(2) Two-stage injection method reduces the fluctuation of combustion over a wide range of $\lambda$, in comparison to one-stage injection. Therefore, two-stage injection method stabilizes combustion.

(3) NOx emission concentration can be reduced by about one half in comparison to that of one-stage late injection, in the range of $\lambda = 1.3$ to 1.7 without deteriorating the thermal efficiencies. This substantial reduction could be achieved by delaying the secondary injection timing in the two-stage injection and spark timing.

ACKNOWLEDGMENT

For this research, Prof. T. Kamimoto of the Tokyo Institute of Technology graciously provided a wealth of invaluable instruction, and Tokyo Gas Co. Ltd., supplied the necessary natural gas fuel. The author would like to thank them for their cooperation.

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