Simultaneous Reduction of NO_x and Soot in Diesel Engines Using a New Mixture Preparation Method*

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We describe an experimental study on the simultaneous reduction of NO_x and soot in diesel engines achieved using a new mixture preparation method. To produce a lean uniform mixture, a new type of injection equipment was used with various injection timings. Between the late and very early injection timings, there was an injection timing range at which the amount of NO_x produced was extremely low and soot was not emitted. NO_x production was suppressed to a level one hundredth that of conventional DI diesel engines. However, with increasing load, NO_x production increased sharply and strong diesel knocking occurred. This indicated that NO_x production was strongly influenced by the process of mixture formation.

Key Words: Diesel Engine, Combustion, Combustion Phenomena, Ignition, Combustion Products

1. Introduction

In diesel engines where the fuel is injected to the cylinder under high temperature and high pressure conditions, the time history of the air-fuel mixture formation determines the engine output and the emission gas properties. Thus, we have assumed that the air-fuel mixture preparation with a different time history might dissolve the trade-off between NO_x and soot in the combustion process. Under this assumption, this experimental study was conducted. It is known that leaner air-fuel mixture or use of oxygen-contained fuel will greatly reduce the soot emission^{(1),(2)}, and that averaging of the flame temperature is effective in reducing the NO_x emissions. In the present study, we have used a new method of air-fuel mixture preparation that is not an extension of any of these published ideas. In other words, we have attempted to radically accelerate the diffusion of the

injected fuel, and determined the effects on the combustion process and the emission gas properties.

2. Concept of Air-Fuel Mixture Preparation

Figure 1 shows the history of formation of air-fuel mixture from injection to ignition in a conceptual manner. In the conventional combustion process in

Fig. 1 History of mixture properties before ignition

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which the fuel is injected to the cylinder under high temperature and high pressure conditions, the fuel is constantly heated by the ambient gas. On the other hand, diffusion of fuel is not promoted, which delays the formation of the mixture, during which period various chemical changes occur. As a result, the so-called thermal-cracked mixture is prepared before the ignition. If ignition starts under such condition, the thermal-cracked component induces radical combustion, making it difficult to obtain the desirable emission properties.

If thorough fuel diffusion is made to prepare the air-fuel mixture after fuel injection, the same heating condition will presumably cause immediate partial oxidation and result in moderate combustion. Thus, we have presumed that the combustion after the ignition will be controlled and NO_x and soot emissions will be reduced at the same time if the component change due to thermal cracking does not occur before ignition. This is the new air-fuel mixture preparation process that we are proposing.

This new process is based on the following concepts:

- (1) Injection is started earlier to promote fuel diffusion.
- (2) To avoid fuel collision with the cylinder wall resulting from fuel injection into the low-pressure state, the fuel spray with low penetration is injected from the large-bore injection valve at the minimum pressure without excessive atomization.
- (3) A guide portion to control scattering of the fuel spray is provided at the tip of the injection valve.
- (4) To avoid fuel decomposition during compression, the fuel is injected for the minimum injection duration to quickly prepare the air-fuel mixture.

3. Acceleration of Fuel Diffusion

A special design needs to be added to the fuel injection valve in order to achieve extensive and uniform injection of fuel in the cylinder space. Figure 2 indicates the shape of the fuel injection valve which incorporates a $\phi 1.9$ mm projection for fuel guide at the tip of the pin. The accumulator fuel injection

Fig. 2 Injection nozzle orifice configuration

system shown in Fig. 3 was used for fuel injection⁽³⁾, and the observation container filled with 2.1 MPa of nitrogen was attached to the injection valve to observe the fuel spray with the TV camera with high-speed shutter. Figure 4 shows the side view and the bottom view of the spray during the observation.

The fuel spray appeared to have spread in a disc-like shape, suggesting the formation of fuel injection with a high degree of diffusion. Figure 5 shows the calculated fuel spray travel after fuel injection. It was found that the fuel spray travel remained short from immediately after the injection, and the injected fuel

Fig. 3 Injection system

Fig. 4 Spray image

Fig. 5 Spray penetration

never reached the cylinder wall during the experiment. We concluded from above findings that this fuel injection valve was suitable for the purpose of this experiment since the fuel could be injected for a large space while avoiding the interference with the cylinder wall, and the spray speed was quickly reduced.

4. Test Engine

Figure 6 shows the single-cylinder direct-injection diesel engine with the cylinder bore of 102 mm and the stroke of 112 mm that was used in the experiment. The intake port has a straight shape to avoid air swirls. The test was conducted at the compression ratios of 14 and 18. The accumulator fuel injection valve was used and located at the center of the cylinder. The conical angle of the spray was set at 120° to expand the spray area to as much of the cylinder wall as possible. Although we first varied the injection pressure between 20 MPa and 50 MPa, no significant change was found in the spread of the spray. Thus, we carried out the combustion analysis at a relatively low pressure, and the engine performance analysis at a relatively high pressure. The ignition timing was also varied extensively from TDC and 150° BTDC. For each cylinder, the fuel injection volume per cycle was varied between approximately 10 mm^3 and 35 mm^3 . The fuel injection volume of 30 mm^3 on the test engine was equivalent to about 50% load. JIS #2 gas oil was used for the test engine and the diesel engine for comparison. The NO_x concentration was measured with the heated chemiluminescent analyzer, the HC concentration (methane equivalent) was measured with the heated flame ionization detector and the soot concentration was measured with the Bosch smoke meter.

Fig. 6 Schematic of the test engine

The bottom view system shown in Fig. 7 was used to observe the combustion inside the cylinder, and the high-speed camera was used to record the combustion phenomenon at each crank angle on the shadow photograph. Moreover, the TV camera with high-speed shutter equipped with image intensifier capable of capturing even a slight light emission was used in the case in which the combustion flame was not luminous. Piezo-type pressure indicator was used to measure the cylinder pressure at each crank angle for 400 cycles, and the pressure indicator diagram was analyzed.

5. Emission Gas Properties and Heat Release

Figure 8 shows the relationship between the injection timing and the emission gas properties or the indicated mean effective pressure. When the injection timing was advanced from the normal injection timing of around TDC, NO_x was increased and soot was reduced, showing the so-called trade-off between NO_x

Fig. 7 Optical system

Fig. 8 Exhaust emissions and indicated mean effective pressure

and soot. However, if the injection timing was further advanced, NO_x was sharply reduced, and the injection timing range without soot emission was found. Since HC increased with yet another advance in the injection timing, the useful range was limited to $50 - 60^\circ$ BTDC in this example. Even in this range, HC was greater than the normal DI diesel engine because thermal reaction of the fuel that spread over a wider area was delayed due to cooling by the ambient gas and as a result more unburned gas was emitted. In the ordinary DI diesel engine, the injection timing is around TDC, and ignition takes place before the fuel is fully diffused. Thus, the unburned fuel covered in flame is emitted as soot. In our opinion, HC will not necessarily increase when the fuel is fully diffused because the cetane number of the fuel is high at about 50, and the fuel will certainly ignite when the ambient gas reaches the ignition temperature by adopting appropriate compression ratio. Thus, it is suggested that there are certain fuel ingredients which temperature will not increase to the ignition temperature at the compression ratio of 18. The increase in HC in the case of earlier injection outside the ignition timing range is due to collision of the fuel with the wall surface. When the ignition timing was advanced from TDC, the indicated mean effective pressure increased with the equivalent injection volume, maximized at 30° BTDC and then decreased. Reasons for the decline in the indicated mean effective pressure from 30° BTDC include the fact that the heat release has started too early and the effective work to the piston has decreased and that the collision of the fuel with the wall surface prevents effective use of the fuel. Despite these factors that adversely affect the fuel economy, the indicated mean effective pressure at the injection timing of around 60° BTDC was equal to the value at the injection timing of around TDC. The fuel injection duration is closely related with wall surface collision and diffusion of fuel, which will be discussed later on. This combustion process is a low-temperature combustion, and its fuel efficiency is normally lower than the DI diesel engine.

Figure 9 shows the rate of heat release at various injection timings. At 60° BTDC which showed low NO_x , the initial heat release took place early as in the case of 40° BTDC. However, the timing of the subsequent major heat release was closer to TDC, and the reduction in the effective work was minimized. In other words, in the 60° BTDC injection and 40° BTDC injection, the fuel started partial reaction at a certain compression end temperature. However, the timing of the subsequent major heat release varied according to the extent of fuel diffusion. In the 60° BTDC injection, a major heat release took place after at tempo-

Fig. 9 Effect of injection timing on cylinder pressure and heat release rate

rary shut-down.

Then we investigated the NO_x emission characteristics under various loads at the ignition timing at which the fuel was well diffused so that NO_x and soot emissions were both close to zero. As shown in Fig. 10, the ordinary diesel engine shows the NO_x emission increasing at a certain gradient as the supplied fuel is increased. The gasoline lean-burn engine also showed a similar NO_x emission trend. This trend was reported to be more or less the same in all types of engines⁽⁴⁾. However, the NO_x emission characteristics were different when the fuel was more diffused. In other words, when the amount of fuel was small, the NO_x value was only one-hundreds or less of the ordinary diesel engine, and the low level of NO_x emission was sustained regardless of the amount of fuel. However, the NO_x emission sharply increased when the amount of fuel was increased.

The above results should be interpreted to suggest that the NO_x emission which normally increases

Fig. 10 Effect of load on NO_x

Compression Ratio = 18 Engine Speed = 1000 rpm
Injection Pressure = 20 MPa Fuel Quantity = 15 mm³/st. cyl.

Fig. 11 Comparison of combustion process

in proportion to the load can be controlled at a low level by improving the level of fuel diffusion.

6. Combustion Process

Figure 11 shows the results of the observation of the combustion phenomenon by means of shadowgraph. When the injection timing was advanced, the combustion process was totally different from the existing diesel combustion. A representative combustion photograph of the 60° BTDC injection is shown. In the 20° BTDC injection for comparison, the degree of diffusion was low, the spray was in cohesive stream, and the luminous flame combustion was observed. In the 60° BTDC injection, on the other hand, the spray shape disappeared before ignition and no luminous flame was observed since the fuel was well diffused.

In order to examine the combustion phenomenon in greater detail, we made an observation of the non-luminous flame section with the TV camera with high-speed shutter and image intensifier. The results are shown in Fig. 12. As the heat release rate pattern shows, there are two peaks of heat release for the 60°

Fig. 12 Combustion process (using image intensifier)

BTDC injection. This tendency can be seen more clearly by lowering the compression ratio from 18 to 14. The case of the compression ratio of 14 is shown here. During the first phase of the heat release, a very weak light first appeared. After a temporary shut-down, a strong light appeared and it was followed by continuous light emission strong enough to make us reduce the shutter speed of the image intensifier.

When we continued with the observation by reducing the shutter speed to about 1/300, scattered light emission appeared after the peak of heat release. The above observation indicates that the second phase of heat release takes place mainly with scattered fuel and it is a combustion of air-fuel mixture accompanied by strong light emission. On the other hand, the light emission in the first phase of heat release was stronger in the 40° BTDC injection which showed a high NO_x value than in the 60° BTDC injection. During the second phase of heat release that immediately followed the first phase, a strong light emission mixed with dot-like high intensity parts appeared, and we had to reduce the shutter speed to yet another digit smaller to about 1/3 000. Unlike the light emission in the 60° BTDC injection, the light emission in the second half of the phase two of heat release was located mainly in the central part of the cylinder. The reason for the temporary shut-down between the first phase and the second phase of heat release may be explained as follows: The first phase of heat release can be considered as the so-called cool flame, and the thermal reaction is slowed down since there are little active species that cause the subsequent thermal flame reaction in the condition where

Fig. 13 Effect of injection duration on NO_x and indicated mean effective pressure

the air-fuel mixture is sparsely distributed. In the ordinary diesel engine, the first phase and the second phase of heat release occur simultaneously and they can not be distinguished from one another.

On the basis of the above combustion process, the low NO_x value should be attributed to low-temperature combustion indicated by non-luminous flame and the short duration in which the entire fuel is burned.

7. Acceleration of Fuel Diffusion by Shorter Injection Duration

If the fuel is intensively injected for a short duration, turbulence of air by the spray will be intensified and the deceleration timing of the spray will be advanced, and the fuel will probably be diffused more thoroughly by the turbulence of air. Figure 13 shows the relationship between NO_x and the indicated mean effective pressure under various injection duration. The injection duration was adjusted by changing the fuel injection valve lift, and the injection valve diameter was kept constant. It is shown that the NO_x emission is compressed and the engine output is increased as the injection duration is shortened.

The relationship between NO_x and the engine output is the opposite of the results obtained with an ordinary diesel engine, which indicates that shorter injection duration has promoted diffusion of fuel. Thus, it may be concluded that the change in the NO_x emission is due to accelerated diffusion of fuel and that the change in the engine output is the result of subdued wall collision of the fuel.

8. Conclusion

The present study has found that the phenomenon that starts with fuel injection that leads to chemical reaction and ends with ignition can be totally different from the currently-seen phenomenon if the injecting duration and the injection timing that are beyond the normal range are chosen. In other words, we have demonstrated that the combustion characteristics can be greatly changed and NO_x and soot can be reduced simultaneously by radically changing the time process of air-fuel mixture preparation. We have also shown that a portion of the reaction kinetics of combustion which has been neglected may give a significant effect by changing the combination of the speed of mixture that affects diesel combustion and the speeds of heating and evaporation. On the basis of the knowledge on the combustion process that has been gained in the present study, we have presented new facts that are not part of our common knowledge.

(1) We have achieved uniform diffusion of oil droplet in the space by combining a short injection duration and early fuel injection, and gained the combustion characteristics of non-luminous flame that are different from the existing diesel combustion at a relatively low diffusion pressure.

(2) The case where heating takes place after the fuel is fully diffused and the case where the heating starts while the diffusion is incomplete have different combustion characteristics. Such an observation suggests the possibility of controlling NO_x and soot emissions, one of the representative characteristics of combustion, by controlling the process of air-fuel mixture preparation.

(3) Formation of NO_x can be dramatically reduced by accelerating the formation of air-fuel mixture before ignition.

A study on the above subject has just begun, but there are high expectations in the technology presented in this study because it is the technology of the future. We are planning another study on the subject in greater detail.

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