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# Diesel Combustion Analysis Based on Two-Zone Model*
## (Examination of Excess Air Ratio in Burned Zone)

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The burned zone excess air ratio $\lambda_d$ in the diffusion combustion process was analyzed using the two-zone model, and it was compared with the excess air ratio $\lambda_f$ estimated using a steady diffusion flame model of the fuel spray. It is found that $\lambda_d$ is dependent on the fuel spray penetration and ignition delay. If the premixed combustion fraction is less than 50%, the NO formation is minimally influenced by the excess air ratio during premixed combustion and is dependent on the excess air ratio $\lambda_d$ which determines the maximum temperature during diffusion combustion. It is clarified by analysis of the two-zone model that the large reduction in $NO_x$ due to timing retard is mainly caused by decreases in both combustion temperature and combustion pressure, and the small reduction in $NO_x$ occurring when the nozzle-hole diameter is decreased, is due to a small decrease in combustion pressure resulting from a decrease in the heat release rate during premixed combustion.

***Key Words***: Diesel Engine, Combustion Analysis, Two-Zone Model, Diffusion Flame Model, Excess Air Ratio, NO Formation, $NO_x$ Reduction Factor

### 1. Introduction

A revised two-zone combustion model was proposed in the authors' first paper[1] which can be used to analyze the processes of $NO_x$ formation and soot formation in diesel combustion. The special feature of the present two-zone model is that the excess air ratio in the burned zone is estimated on the basis of the measured time-histories of combustion pressure and heat release rate through an iterative calculation in which the calculated total cumulative NO formation was equated to the measured amount of exhaust $NO_x$. The calculated time-histories of the burned gas temperature and the cumulative soot formation were compared with the histories of the flame temperature and the $KL$ value which were measured simultaneously by the infrared two-color method. Since the calculated results agreed well with the measured ones, the present two-zone combustion model was found to be useful and effective for analyzing diesel combustion behavior except for the swirl effect on soot formation.

An important factor which determines the burned gas temperature is the amount of air entrained into the fuel spray, although there are a lot of factors which affect combustion in diesel engines. This factor is equivalent to the excess air ratio in the burned zone in the present two-zone model. In this paper, the authors examine how the burned gas temperature history and the cumulative NO formation are affected by the assumed excess air ratio pattern during the premixed combustion period, since it has been shown that NO is mainly formed during the premixed combustion period.

Secondly, a steady diffusion flame model is proposed to clarify which physical factor most affects the excess air ratio during diffusion combustion. The steady diffusion flame model consists of a steady fuel

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spray cone from a nozzle-hole, in which air is entrained into the spray cone from the surroundings and the fuel burns under the stoichiometric mixture condition. Comparing the excess air ratio estimated using the steady diffusion flame model with the one determined using the two-zone model, it is clarified that the excess air ratio during diffusion combustion is mainly dependent upon the fuel spray penetration and the ignition delay. Finally, the large reduction in exhaust $NO_x$ concentration resulting from timing retard and the small reduction in exhaust $NO_x$ concentration resulting from a decrease in nozzle-hole diameter are analyzed using the present two-zone model from the viewpoint of the excess air ratio during diffusion combustion.

## 2. Excess Air Ratio during Premixed Combustion

The excess air ratio during diffusion combustion was assumed to be constant during the whole period of diffusion combustion. Since the fact that good agreement was obtained between the calculated temperature history and the measured one as shown in the authors' first paper[1], the assumption seemed to be reasonable. In this paper, the effect of the excess air ratio during premixed combustion on the burned gas temperature and on NO formation is investigated by analyzing four typical patterns. Figure 1 shows the relationship between the heat release rate curve and the excess air ratio pattern assumed in the premixed combustion period. The analysis is simplest in the case of pattern P-1 because the excess air ratio is assumed to be constant from ignition to the end of combustion. However, the excess air ratio must be equal to unity at ignition because ignition occurs at the region of the fuel spray with the excess air ratio of unity[2]. In pattern P-2, after igniting under the condition that the excess air ratio is unity, that is, $\lambda_{po}=1.0$, the excess air ratio varies in accordance with a quadratic function. This curve is connected continuously to the curve for constant excess air ratio during diffusion combustion. Pattern P-2 seems to be the most realistic because the combustion process varies continuously. In pattern P-4, the excess air ratio averaged during the premixed combustion period is much closer to unity than in case of the other patterns, and pattern P-3 is intermediate between patterns P-2 and P-4.

Figures 2 and 3 show the changes in the calculated burned gas temperature and the calculated NO formation rate due to the excess air ratio pattern of the premixed combustion period under the condition of a constant diffusion combustion excess air ratio of $\lambda_d=1.50$. Figure 2 shows the results for the high-load operation condition (3 185 rpm, $P_{me}=0.83$ MPa) with a small fraction of premixed combustion, and Fig. 3 shows those for the low-load operation condition (1 750 rpm, $P_{me}=0.40$ MPa) with a large fraction of premixed combustion. In these figures, $dQ/d\theta$ denotes the heat release rate which is calculated from the measured combustion pressure history $P$. It is assumed that $dQ/d\theta$ and $P$ remain unchanged in the present calculation if the excess air ratio pattern is changed. $T_b$, $T_c$ and $T_u$ are the calculated gas temperatures of the burned zone, the in-cylinder average and

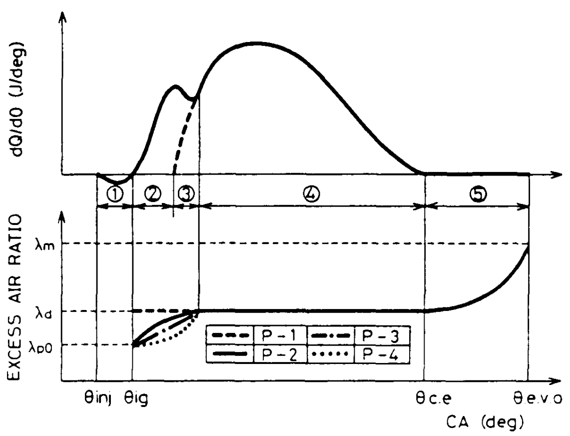


Fig. 1 Relationship between heat release curve and excess air ratio

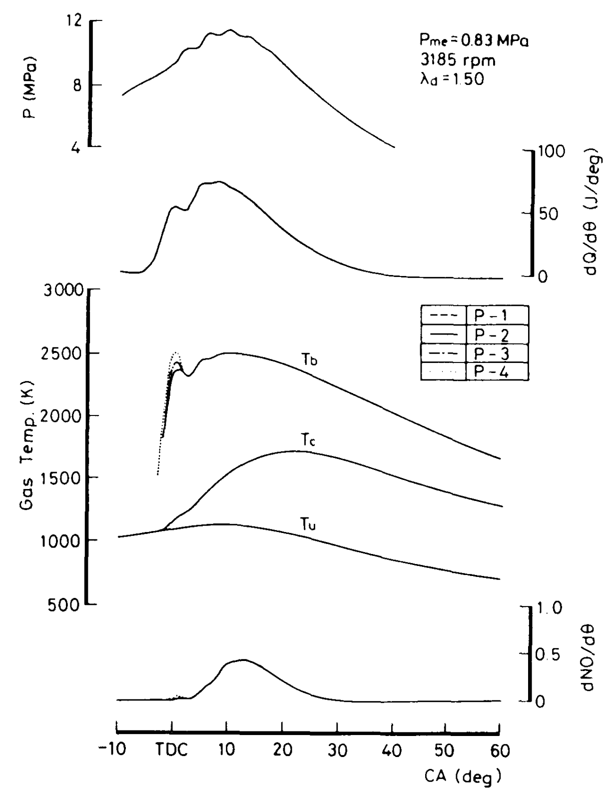


Fig. 2 Effect of assumed excess air ratio pattern (for the case of small premixed combustion fraction)



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the unburned zone respectively, and $dNO/d\theta$ is the calculated NO formation rate.

As the excess air ratio in the burned zone averaged during the premixed combustion period approaches unity with changing in the pattern from P-1 to P-4, the burned gas temperature increases. Also the maximum temperature $T_{pmax}$ during the premixed combustion period rises markedly and it is sometimes higher than the maximum temperature $T_{dmax}$ during the diffusion combustion period. As a result of this temperature rise, the NO formation rate increases significantly under the low-load condition but it is little affected under the high-load condition. It is reasonable that the excess air ratio during premixed combustion markedly affects the NO formation if the premixed combustion fraction is as large as 58% of the total cumulative heat release as shown in Fig. 3.

From a comparison of the time-history of the burned gas temperature calculated using the present two-zone model and that of the flame temperature measured by means of the two-color method[3], pattern P-1 or P-2 seems to be appropriate for the present model. Also it was shown by Kamimoto et al.[4] that the spike distribution of the flame temperature was not observed in the early stage of combustion, and the fraction of NO formed in the premixed combustion period was small judging from their measured time-history of $NO_x$ concentration obtained using the gas sampling method. In addition, the measured maximum flame temperature was as high as 2 500 K as shown in previous reports[3]-[5]. If pattern P-2 is adopted, the cumulative NO formation during the premixed combustion period is as much as 20% of the total amount of NO formed, even in the case of a large premixed combustion fraction as shown in Fig. 3.

Figure 4 shows the effect of the excess air ratio pattern on the maximum temperatures $T_{pmax}$ and $T_{dmax}$, the estimated diffusion combustion excess air ratio $\lambda_{do}$ and the NO fraction formed during the premixed combustion period. In the calculations, the total cumulative NO formation was equated to the measured amount of exhaust $NO_x$ under each experimental condition for all the excess air ratio patterns P-1 to P-4. These experimental data were obtained by varying the brake mean effective pressure from $P_{me}$=0.83 to 0.40 MPa under the condition of a constant engine speed of 3 185 rpm. Even when the premixed combustion fraction is less than 10%, as indicated by the open circles and open square symbols in Fig. 4, $T_{pmax}$ increases markedly when the pattern changes from P-1 to P-4. However, the cumulative NO fraction formed during the premixed combustion period is so small as to be negligible. If the premixed

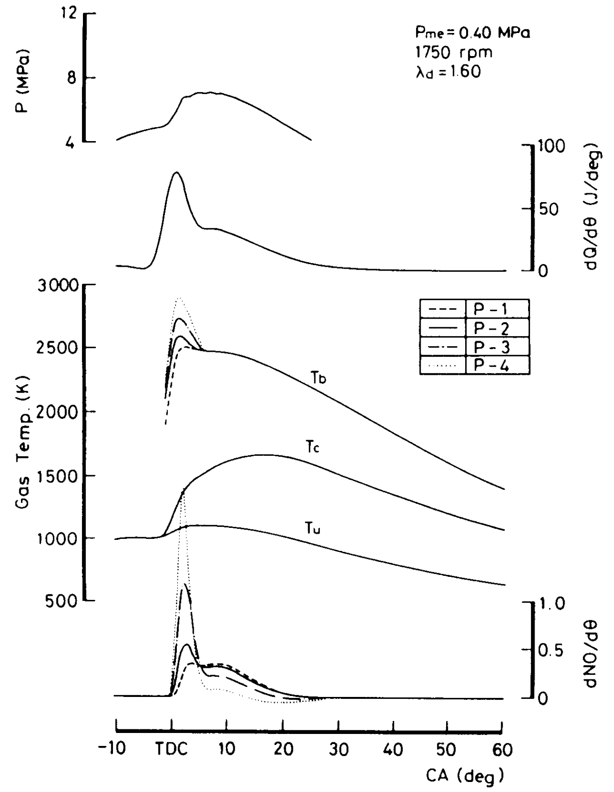


Fig. 3 Effect of assumed excess air ratio pattern (for the case of large premixed combustion fraction)

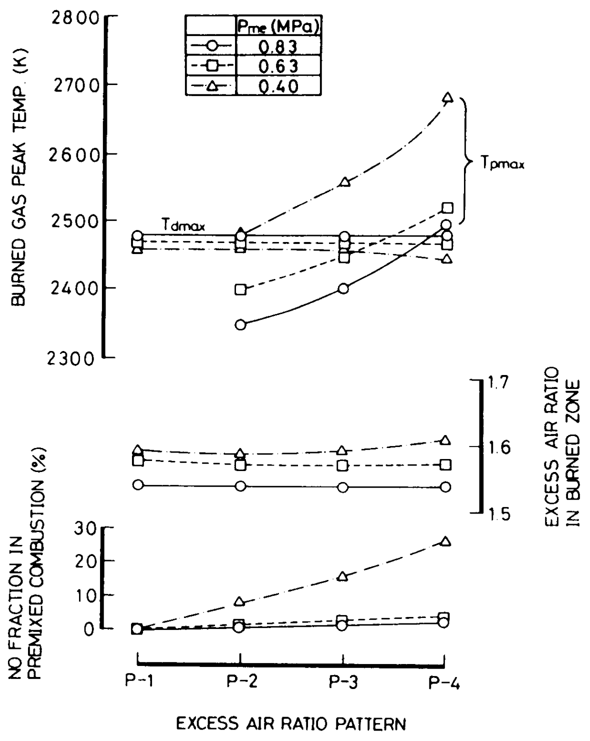


Fig. 4 Effect of assumed excess air ratio pattern on fraction of NO formed during premixed combustion



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combustion fraction is 33%, as shown by the open triangles, the cumulative NO fraction is less than 10% for pattern P-2, and less than 30% even in the case of pattern P-4. That is, the excess air ratio pattern in the premixed combustion period hardly affects the total amount of NO except in the case that the premixed combustion fraction is larger than 50%. Therefore, it can be said that the total amount of NO formed is dependent mainly upon the maximum burned gas temperature during the diffusion combustion period. In other words, NO formation is determined by the excess air ratio during diffusion combustion.

## 3. Excess Air Ratio during Diffusion Combustion

In the case of previously reported experimental results obtained under various engine operation conditions[3], the diffusion combustion excess air ratio $\lambda_{do}$ was determined using the present two-zone model in which the calculated total cumulative NO formation was equated to the measured amount of exhaust $NO_x$. The calculated diffusion combustion excess air ratio $\lambda_{do}$ is compared in Fig. 5 with the in-cylinder average excess air ratio $\lambda_m$. The in-cylinder average excess air ratio $\lambda_m$ varies greatly depending upon both engine speed and mean effective pressure while the diffusion combustion excess air ratio $\lambda_{do}$ hardly changes with the mean effective pressure but varies a little with the engine speed. The calculated values of $\lambda_{do}$ are divided roughly into two groups according to the engine speed, and looking at each group in more detail, it is found that $\lambda_{do}$ increases a little with $\lambda_m$ but there is no definite correlation between them. The experimental data obtained using marine diesel oil (A 40) and low-quality gas oils (ADO 40, ADO 45) with a low ignitability and a cetane index of 40 or 45 fall into the group corresponding to the engine speed of 1 750 rpm despite the fact that those data were obtained at an engine speed of 3 185 rpm.

In order to clarify which physical factor determines the excess air ratio during diffusion combustion, a steady diffusion flame model is proposed as shown in Fig. 6. This spray flame model consists of a conical fuel spray injected from a nozzle-hole, which has a half cone angle of $\phi$ and a length, $l_0$. Air is entrained continuously through the conical spray surface between points A and C. It is assumed that the fuel-air mixture begins to burn at point B on the cone surface, that the fuel continues to burn under the stoichiometric mixture condition $\lambda=1.0$ between B and C, in which interval the amount of burned fuel increases in proportion to the distance from B to C, and finally that all the fuel has burnt at cross section C. The distance $l_1$ between A and B is the fuel penetration length during the ignition delay period, and the distance $l_0$ between A and C is the fuel penetration length calculated by assuming that the average excess air ratio is unity at the spray tip cross section C.

In terms of the spray penetration length $\chi$ and the amount of air $G_a$ entrained into the fuel spray, the following equations were derived by Wakuri et al.[6]

$$\chi=(2c\Delta p/\rho_a)^{0.25}(td/\tan\phi)^{0.5} \tag{1}$$

$$G_a=G_f(2\tan\phi/c^{0.5})(\rho_a/\rho_f)^{0.5}(\chi/d), \tag{2}$$

where $G_a$ and $G_f$ are the flow rates of air and fuel which pass through the cross section at the penetration length $\chi$, $d$ is the nozzle-hole diameter, $c$ is the contraction coefficient of the nozzle-hole, $\Delta p$ is the pressure difference between the fuel injection pressure and the in-cylinder gas pressure, $t$ is time, and $\rho_a$ and $\rho_f$ are the densities of the in-cylinder air and fuel respectively. These equations were originally derived for the fuel spray in the combustion chamber of a large-bore engine. However, it is assumed that the fundamental concept of fuel spray penetration is also applicable in a small high-speed diesel engine,

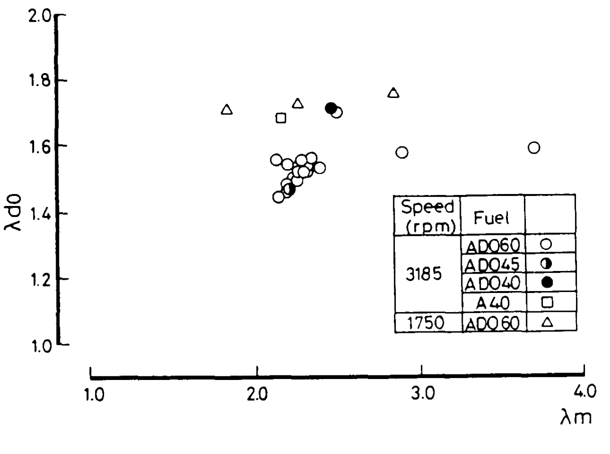


Fig. 5 Correlation between diffusion combustion excess air ratio and in-cylinder average excess air ratio

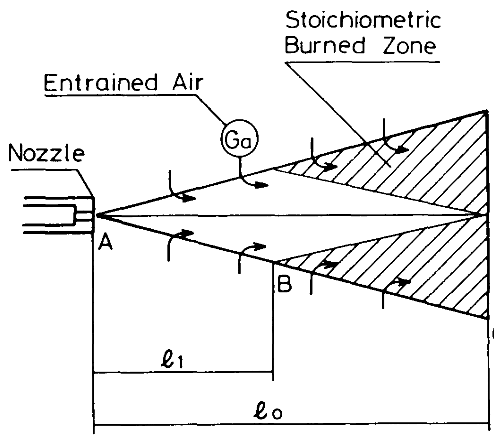


Fig. 6 Steady diffusion flame model of fuel spray

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although the fuel spray in a small-bore engine must be disturbed by impingement to the combustion chamber wall, swirl and squish. In other words, it is assumed in the diffusion flame model that the momentum of the injected fuel is conserved in the fuel spray region and in the burning region of the flame, and that the shape of fuel spray is a cone.

Under the above assumptions, the penetration lengths $l_i$ and $l_o$ are calculated using the following equations derived from Eqs.(1) and (2).

$$l_i=(2c\Delta p/\rho_a)^{0.25}(\Delta t d/\tan\phi)^{0.5} \quad (3)$$

$$l_o=d(c^{0.5}L_{th}/2\tan\phi)(\rho_f/\rho_a)^{0.5}, \quad (4)$$

where $\Delta t$ is the ignition delay time and $L_{th}$ is the theoretical air-fuel ratio.

The whole region of the spray cone between A and C is assumed to be a quasi-flame region in the present model, and the equivalent excess air ratio $\lambda_f$ of the heterogeneous spray flame is defined as the ratio of the amount of air in the whole cone to the amount of fuel burned in the region between B and C under the stoichiometric mixture condition.

$$\lambda_f=\frac{\text{Volume of air in the flame}}{\text{Volume of stoichiometric burned gas}}=(AC)/(BC)=l_o/(l_o-l_i) \quad (5)$$

Here, $\lambda_f$ defined by Eq.(5) is called the model diffusion flame excess air ratio. $l_i$ and $l_o$ are calculated by substituting the experimental values for $\Delta t$, $\Delta p$, $\rho_a$ etc. into Eqs.(3) and (4), where $\Delta t$ is the measured ignition delay, $\Delta p$ is the average nozzle pressure difference calculated from the measured injection duration and the measured amount of fuel injected, and $\rho_a$ is the average air density during the injection period.

Figure 7 shows the correlation between the model diffusion flame excess air ratio $\lambda_f$ and the diffusion combustion excess air ratio $\lambda_{do}$ determined using the two-zone model. There is some scattering in the correlation between them because the experimental data were obtained using various fuel oils and under various engine operation conditions of engine speed, brake mean effective pressure, injection timing and nozzle hole-diameter. However, $\lambda_{do}$ agrees quantitatively with $\lambda_f$ especially in the cases shown by open circles, and the values of $\lambda_{do}$ are in a narrow range between 1.45 and 1.75, which is almost equal to the value reported by Sakane et al.[7] under the low-load condition which is shown by open triangles, the injected fuel has burnt completely before reaching the penetration length $l_o$, and so the data are located away from the solid line. The same tendency is seen in the case of the low-quality gas oil.

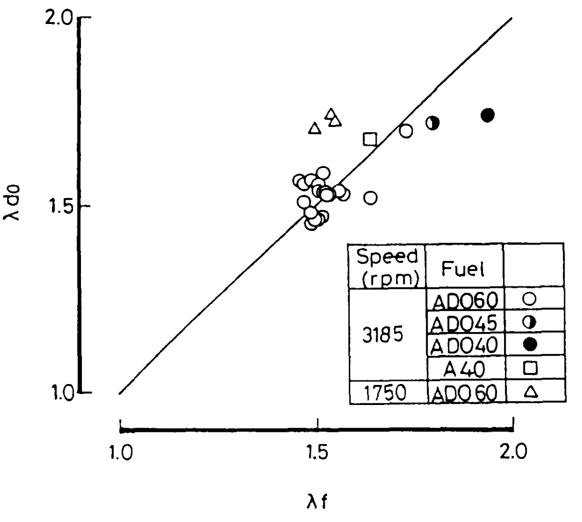


Fig. 7 Correlation between diffusion combustion excess air ratio and model diffusion flame excess air ratio

The modeled diffusion flame is not always sufficiently formed in the combustion chamber of an actual small diesel engine because the fuel spray is not conical due to the influences of swirl and squish. In addition, the fuel spray sometimes impinges on the piston cavity wall and the injection duration is not long enough to attain the penetration length $l_o$, especially under the low-load condition. Although the proposed diffusion flame model is very rough and hypothetical, the diffusion combustion excess air ratio based on the two-zone model analysis is almost equal to the model diffusion flame excess air ratio judging from the good agreement between $\lambda_{do}$ and $\lambda_f$. It can be concluded that the diffusion combustion excess air ratio is mainly dependent upon the spray penetration and the ignition delay. Furthermore, the diffusion combustion excess air ratio hardly varies when the injection timing and the nozzle hole-diameter are changed. However, it varies with the fuel properties and the fuel injection duration.

## 4. $NO_x$ Reduction Factor Analysis

### 4.1 Effect of injection timing retard

The $NO_x$ reduction factor was analyzed using the present two-zone model mainly by considering the excess air ratio in the burned zone in relation to the experimental results obtained under various timing retard conditions from 11 to 5 degrees before top dead center (BTDC) every two degrees. Figure 8 shows the measured time-histories of combustion pressure and heat release rate, and the time-histories of gas temperatures and NO formation rate calculated under the constant excess air ratio of $\lambda_d=1.50$. The calculated exhaust NO concentration is shown in Fig. 9 by solid lines corresponding to excess air ratios of $\lambda_d=1.4$, 1.5, 1.6, 1.7 and 1.8. In this calculation, the measured time-histories of combustion pressure and heat release rate shown in Fig. 8 were used for each excess air ratio. In



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Fig. 9, the solid circles show the measured exhaust $NO_x$ concentration and the open circles show the NO concentration calculated using the model diffusion flame excess air ratio $\lambda_f$ instead of $\lambda_d$.

According to the results calculated using the constant diffusion combustion excess air ratio shown by the solid lines in Fig. 9, the NO concentration is reduced markedly by timing retard. For example, in the case of $\lambda_d=1.5$, it is reduced from 1 200 ppm to 400 ppm by changing the timing retard from 11 to 5 degrees BTDC in crank angle, that is, it is about one-third of the original. This large reduction in NO concentration is caused by a decrease in the maximum burned gas temperature during the diffusion combustion period and, in addition, by a large decrease in the maximum combustion pressure due to the timing retard as shown in Fig. 8. That is, the $NO_x$ concentration can be reduced significantly by the simultaneous decreases in combustion pressure and temperature because the later the injection timing, the more the energy released by combustion is converted to expansion work rather than temperature rise. However, the reduction in the measured $NO_x$ concentration was smaller than the calculated one, as is shown by the difference between the solid circles and the solid line for $\lambda_d=1.5$. This is because of a small decrease in the diffusion combustion excess air ratio $\lambda_{do}$ caused by the timing retard. The model diffusion flame excess air ratio $\lambda_f$ remains almost unchanged by the timing retard as indicated by the open circles. There may be some other factors which decrease the diffusion combustion excess air ratio.

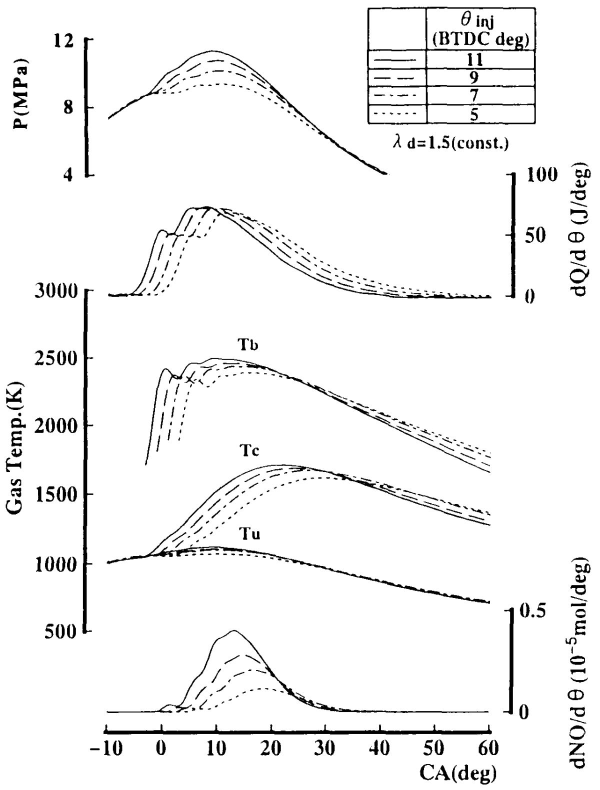


Fig. 8 Analysis of timing retard effect using two-zone model

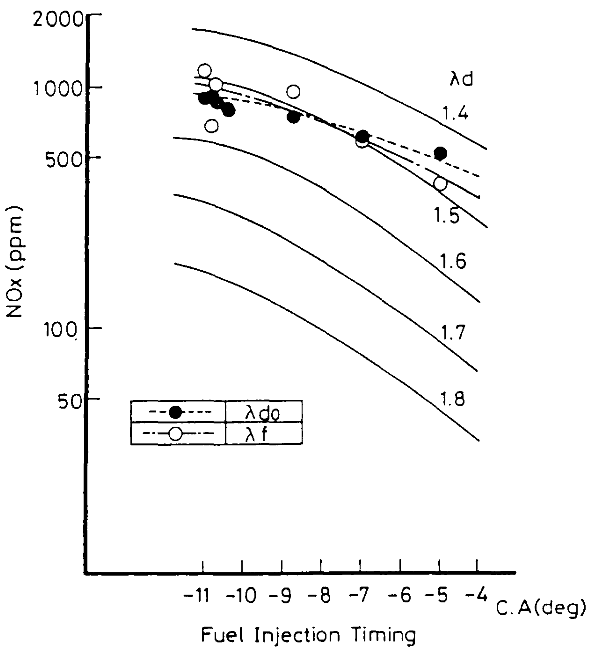


Fig. 9 Analysis of $NO_x$ reduction due to timing retard

### 4.2 Effect of nozzle-hole diameter reduction

Table 1 shows the measured values of the exhaust $NO_x$ concentration, the exhaust smoke density $D_s$ and the maximum combustion pressure $P_{max}$. The nozzle-hole diameter was decreased from 0.30 mm to 0.24 mm at an engine speed of 3 185 rpm and a brake mean effective pressure of 0.83 MPa. The estimated diffusion combustion excess air ratio $\lambda_{do}$ is also shown in Table 1. Figure 10 shows the measured time-histories of combustion pressure and heat release rate, and the calculated time-histories of gas temperatures and NO formation rate under the constant excess air ratio of $\lambda_d=1.50$. The smaller the nozzle-hole diameter, the lower the heat release rate in the premixed combustion period. As a result, a lower maximum combustion pressure was attained. However, the combustion duration was almost unchanged, even in

Table 1 Experimental results for nozzle hole diameter reduction

| d (mm) | NOx (ppm) | Ds (Bosch) | Pmax (MPa) | $\lambda_{do}$ |
|---|---|---|---|---|
| 0.30 | 1172 | 0.61 | 12.0 | 1.57 |
| 0.28 | 1198 | 0.77 | 11.9 | 1.54 |
| 0.26 | 1134 | 0.64 | 11.7 | 1.53 |
| 0.24 | 1035 | 0.61 | 11.4 | 1.52 |



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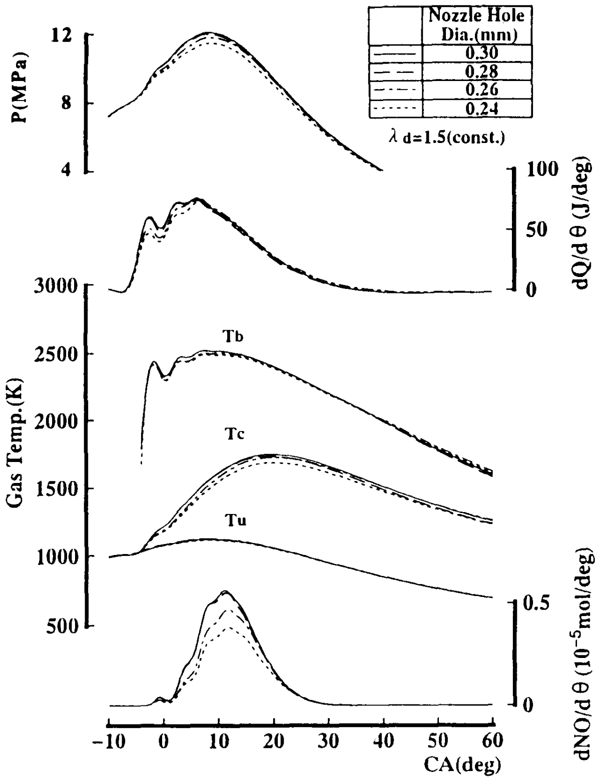


Fig. 10 Analysis of nozzle-hole diameter effect using two-zone model

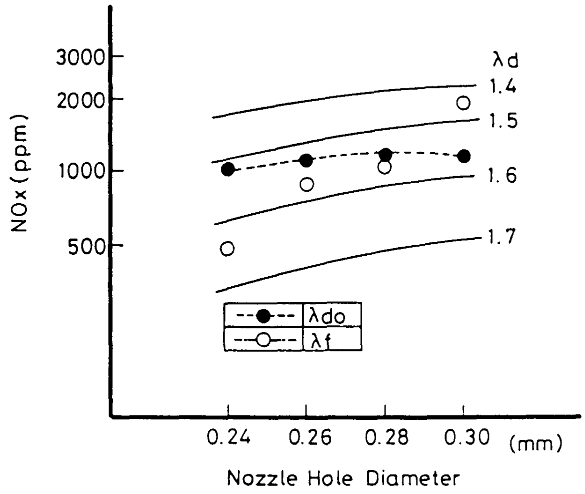


Fig. 11 Analysis of reduction in exhaust $NO_x$ due to nozzle-hole diameter

the case of the smaller nozzle-hole in which the injection duration increased.

In Fig. 11, the calculated NO concentration is shown by solid lines corresponding to $\lambda_d=1.4$, 1.5, 1.6 and 1.7. The solid circles show the measured exhaust $NO_x$ concentration and the open circles show the NO concentration calculated using the model diffusion flame excess air ratio $\lambda_f$ instead of $\lambda_d$. As shown in Table 1 and Fig. 11, the measured $NO_x$ and $D_s$ have maximum values at the nozzle-hole diameter of 0.28 mm, and both decrease simultaneously with further decrease in the nozzle-hole diameter. It is clearly seen in Fig. 10 that the NO formation rate decreases markedly while the burned gas temperature $T_b$ changes little with decreasing nozzle-hole diameter. This small reduction in $NO_x$ concentration is due to a decrease in combustion pressure near the maximum temperature timing, which results from a lower heat release rate in the premixed combustion period due to the lower initial injection rate resulting from the decreased nozzle-hole diameter. Furthermore, the smoke density also decreased slightly because the combustion duration was not increased in spite of the longer injection duration when the nozzle-hole was smaller.

According to Eq.(3), the spray length $l_1$ varies little with decreasing nozzle-hole diameter because the increase in pressure difference $\Delta p$ is canceled by the decrease in nozzle-hole diameter. Also, the ignition delay $\Delta t$ was changed little in the experiment. In addition, it is clear from Eq.(4) that the penetration length $l_0$ decreases in proportion to the nozzle-hole diameter. Consequently, the model diffusion flame excess air ratio $\lambda_f$ increases, and the calculated NO concentration is reduced markedly as shown by the open circles in Fig. 11. On the other hand, the measured exhaust $NO_x$ concentration shown by solid circles in Fig. 11 indicates a slight decrease in the diffusion combustion excess air ratio $\lambda_{do}$ due to a decrease in the nozzle-hole diameter, as is also seen in Table 1. This suggests that, in the actual combustion process, it is difficult to significantly change the excess air ratio in the burned zone simply by changing the nozzle-hole diameter.

## 5. Conclusions

It may be difficult to estimate the soot formation using the simple two-zone model because it is dependent strongly on the local conditions of gas temperature, excess air ratio, swirl and squish. However, the NO formation is mainly determined by the peak burned gas temperature. In the authors' first paper, the present two-zone combustion model was found to be useful and effective for analyzing diesel combustion behavior except for soot formation since the calculated time-history of the burned gas temperature agreed well with that of the measured flame temperature.

In this paper, the burned zone excess air ratio during diffusion combustion was determined using the two-zone model, and it was compared with the excess air ratio of the model diffusion flame in order to clarify which physical factor determines the excess



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air ratio in the burned zone. The effect of excess air ratio in the burned zone on NO formation was examined analytically in the premixed combustion and diffusion combustion periods using the two-zone model. Finally, the effects of timing retard and decrease in nozzle-hole diameter on $NO_x$ reduction were analyzed using the two-zone model. The results obtained are as follows:

(1) If the premixed combustion fraction is less than 50%, the excess air ratio during premixed combustion has little effect on cumulative NO formation. The total cumulative NO formation is mainly dependent upon the diffusion combustion excess air ratio which determines the maximum burned gas temperature in the diffusion combustion period.

(2) The diffusion combustion excess air ratio agrees quantitatively with the model diffusion flame excess air ratio but it does not correlate well with the in-cylinder average excess air ratio. It is found that the diffusion combustion excess air ratio is mainly dependent upon the fuel spray penetration and the ignition delay.

(3) The large reduction in exhaust $NO_x$ concentration due to timing retard is caused by decreases in both the maximum combustion temperature and the maximum combustion pressure in the expansion stroke.

(4) The small reduction in both exhaust $NO_x$ concentration and smoke density due to a decrease in nozzle-hole diameter is caused by a decrease in combustion pressure resulting from a decrease in the premixed combustion heat release rate without increasing the combustion duration.

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## References

(1) Ishida, M., Ueki, H., Matsumura, N. and Chen, Z. L., Diesel Combustion Analysis by Two-Zone Model (Part 1), Trans. Jpn. Soc. Mech. Eng., (in Japanese), Vol. 60, No. 573, B(1994), p. 1845

(2) Hiroyasu, H., Kadota, T. and Arai, M., Development and Use of Spray Combustion Modeling to Predict Diesel Engine Efficiency and Pollutant Emissions (Part 1), Trans. Jpn. Soc. Mech. Eng., (in Japanese), Vol. 48, No. 432, B(1982), p. 1606

(3) Ishida, M., Matsumura, N., Ueki, H., Ito, G., Kubota, S. and Ko, J.G., Measurement of Diesel Combustion by Optical Fiber Thermometer (Part 1), Trans. Jpn. Soc. Mech. Eng., (in Japanese), Vol. 58, No. 555, B(1992), p. 3482

(4) Kamimoto, T., Aoyagi, Y., Matsui, Y. and Matsuoka, S., The Effects of Some Engine Variables on Measured Rates of Air Entrainment and Heat Release in a DI Diesel Engine, SAE-Trans., Paper No. 800253 (1980)

(5) Yan, J. and Borman, G.L., Analysis and In-Cylinder Measurement of Particulate Radiant Emissions and Temperature in a Direct Injection Diesel Engine, SAE, Paper No. 881315 (1988)

(6) Wakuri, Y., Fujii, M., Amitani, T. and Tsuneya, R., Studies on Penetration of Fuel Spray of Diesel Engine, Trans. Jpn. Soc. Mech. Eng., (in Japanese), Vol. 25, No. 156 (1959), p. 820

(7) Sakane, A., Toshioka, S., Sumimoto, T. and Hamamoto, Y., Effect of Fuel Injection Pressure on Diesel Combustion, J. Marine Eng. Soc. Jpn., (in Japanese), Vol. 23, No. 1 (1988), p. 47



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