Modeling of Spray Combustion for the Prediction of Nitric Oxide Emission from Large-Sized Marine Diesel Engines*

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Abstract
A new model of spray combustion with a flame boundary and two NO formation regions was developed in order to predict NO emission from large marine diesel engines. The basic prediction characteristics were studied by using the actual engine specifications and several parameters such as a droplet initial diameter, a fuel injection period, and a flame angle. The simulation could show the change of chemical species composition in the inside and outside of a spray during their combustion. As a result, it was found that this model predicts that the NO formation is frozen after the end of combustion, and the flame angle influences the NO exhaust rate and the thermal efficiency more sensitively than the other parameters.

Key words: Marine Diesel Engine, Spray Combustion, Simulation, Modeling, NO Formation

1. Introduction

Large marine diesel engines may be forced to operate in different combustion conditions from those supposed at the time of their design. The reason is that the fuel has differences in the quality for every production location. If the conditions of the fuel injection, such as injection temperature and the injection timing, can be optimized according to the fuel quality at the time of the operation, the emissions of NO and others may be able to be reduced. If simulations for engine optimization become possible, it is expected that anyone can acquire the guideline of optimization easily without depending on experience and intuition. However, in order to optimize the fuel injection system according to the fuel quality, it is necessary to choose the optimum conditions from various conditions by predicting the engine performance. When it takes into consideration that the calculation is performed by an inboard workstation and the combination of calculation conditions becomes huge, the algorithm which performs each calculation for a short time is required. If the throughput of the present computers is taken into consideration, the computational models require solving the comprehensive states of sufficient level to give the guideline of optimization rather than solving the local states in the cylinder.

Authors have been investigating on the characteristics of the atomization and combustion of diesel sprays for large marine diesel engines. This study proposed the multi-region reactions model of the spray combustion based on these investigations in order to establish the practical cycle simulation technique which suits the above-mentioned purpose. Furthermore, the parameter study was performed by changing the conditions of the fuel injection system in order to clarify the numerical prediction characteristics of the computational models.
2. Computational models

Required computational models are a spray model, a fuel evaporation model, a chemical reaction model, etc. As a result of trying various models, it was found that if the models are simple, it will lead to a reduction of computing time, however, various faults will be encountered and there are limitations on the simplification. This study evaluated the validity of the following models which can obtain the stabilized solutions within the range of practical conditions.

2.1 Fuel model and fuel evaporation model

The fuel model assumes that the fuel is a hydrocarbon and its composition is expressed as $C_mH_n$. Here, $m$ and $n$ are determined from the element analysis of the fuel. The fuel evaporation model assumes that droplets are spherical, they evaporate according to an evaporation rate controlling model, and the evaporation rate is constant during their combustion.

2.2 Spray flame boundary model

The spray flame boundary model is shown in Fig. 1. In this model, the fuel injected from a nozzle hole forms a free spray, and the droplets in the spray evaporate during the penetration into the cylinder. Simultaneously with the injection, a flame boundary (refer as the free spray flame) is formed at the outer edge of the spray. In the case of large marine diesel engines whose engine speeds are about 60rpm, the ignition delay is from 0.01 to 0.04° in the crank angle. From the comparison with the calculation interval, the influence seems small even if the delay is assumed to be zero. The penetration distance of the free spray is calculated using the following assumptions: surrounding gas is entrained into the spray while the free spray penetrates, and the sum of the momentum of droplets and the momentum of gas components in the spray is always conserved equally to the momentum of the injected fuel.

![Fig. 1. Spray flame boundary model with a free spray flame and a wall spray flame.](image)

The cross-section of the spray $A_{sp}$ is given by Eq. (1).
\[ A_{spr} = \frac{\pi}{4} \left( d_{jet} + 2z_{spr} \tan \frac{\theta_f}{2} \right)^2 - d_{jet}^2 \]  

(1)

where, \(d_{jet}\) is the diameter of the jet at the nozzle exit of the fuel valve, \(z_{spr}\) is the penetration distance of the spray tip, and \(\theta_f\) is the spray flame angle.

The spray tip speed \(v_{spr}\) is calculated from Eq. (2) by using \(A_{spr}\) in Eq. (1). Here, it is assumed that the speed of gas components was equal to the speed of droplets.

\[ v_{spr} = -\frac{m_{jet} + \sqrt{m_{jet}^2 + 4\rho_{spr}A_{spr}m_{jet}v_{jet}}}{2\rho_{spr}A_{spr}} \]  

(2)

where, \(m_{jet}\) is the injected fuel mass in the calculation angle interval, \(\rho_{spr}\) is the average gas density in the spray, and \(v_{jet}\) is the speed of the jet at the nozzle exit.

If fuel droplets in the free spray collide with the cylinder wall before the droplets complete their evaporation, an impinged spray (refer as the wall spray) is formed. Furthermore, it is assumed that a wall spray flame boundary (refer as the wall spray flame) is formed at the outer edge of the wall spray. In addition, the calculation of the shape of the wall spray is omitted because it is unnecessary to calculate the reaction.

2.3 Multi-region chemical reactions model

The multi-region chemical reactions model is shown in Fig. 2. In this model, the free spray flame and the wall spray flame form the chemical equilibrium regions. Eleven chemical species of \(CO_2, H_2O, O_2, N_2, CO, H_2, O, OH, H, NO,\) and \(N\) come in the regions from the inside and outside of the spray. Nine chemical species except for NO and \(N\) become chemical equilibrium in the regions, and the species flow out of the regions. NO and \(N\) are not generated in the regions, but they are formed by the extended Zel’dovich mechanism at the inside and outside of the spray independently. However, the local calculation is not carried out but the overall values were calculated in the inside and outside of the spray respectively. Furthermore, the total in all regions is equal to the total amount of generation in the cylinder.

![Fig. 2. Multi-region chemical reactions model which consists of the spray flame boundary and two NO formation regions.](image)

When the spray expands, the outer gas passes through the spray flame boundary and it flows in the inside of the spray. The amount of inflow depends on the rate of volume
change. When the spray reduces, the inner gas passes through the spray flame boundary and it flows out of the spray. Because the angle of the spray flame is given, the gas volume which passes through the spray flame is fundamentally determined by the rate of change of the spray shape. If oxygen is insufficient for the stoichiometric burning of the fuel vapor, additional oxygen is supplied from the inside and outside of the spray to the spray flame boundary. The feed rates are proportional to the concentrations of oxygen in both regions. The method of the literature \(^{(9)}, (10)\) was used to calculate the chemical equilibrium concentrations.

Although the concentration of NO is calculable from Eq. (3) by using the extended Zel’dovich mechanism, the concentration of N is extremely small compared with other chemical species. Therefore, in order to acquire sufficient calculation accuracy, it is necessary that the calculation time interval \(Δt\) is less than \(10^{-6} \text{ s}\) \(^{(11)}\). In the case of 60rpm, this means that the calculation angle interval \(Δθ\) is less than \(3.6 \times 10^{-4} \text{ deg}\). Because a considerable number of the repeat of the cycle calculation is required until it converges, if this condition is applied, the computing time becomes huge. It cannot be said that it is practical. Although it is common to assume a quasi-steady state to N, the following method was tried in this study. The adiabatic flame temperature and the chemical equilibrium concentrations are calculated in every interval \(Δθ\). And, \(Δθ\) is further divided with the number of subdivision \(n_{div}\).

\[
\frac{d[NO]}{dt} = (k_{f1}[N_2][O] + k_{f2}[N]O_2 + k_{f3}[N][OH]) - (k_{b1}[NO][N] - k_{b2}[NO][O] - k_{b3}[NO][H])
\]  
(3)

where, the expression of \([A]\) means the mole fraction of species \(A\), \(t\) is the time, \(k\) is the reaction rate constant, and subscripts 1f to 3f mean the forward reactions in the extended Zel’dovich mechanism, and 1b to 3b mean the backward reactions in the extended Zel’dovich mechanism.

The transformation of Eq. (3) makes Eq. (4).

\[
\frac{[NO]_{i+1} - [NO]_i}{Δt} = k_{f1}[N_2][O] + k_{f2}[N]O_2 + k_{f3}[N][OH] - (k_{b1}[NO][N] + k_{b2}[NO][O] + k_{b3}[NO][H])
\]  
(4)

where, the subscriptions \(i\) and \(i+1\) respectively mean the states before and after the subdivided time interval \(Δt = Δθ/(n_{div} \omega)\), which corresponds to the subdivided angle interval \(Δθ/n_{div}\), and \(ω\) is the angular velocity.

If Eq. (4) is arranged, Eq. (5) is derived regarding NO. In the same way, Eqs. (6)~(11) are derived regarding N, O\(_2\), O, OH, and H. The values in the literature \(^{(12)}\) were used for the reaction rate constant. The reaction temperature is the average temperature in the cylinder.

\[
[NO]_{i+1} = \left\{ \frac{[NO] + Δt \left( k_{f1}[N_2][O] + k_{f2}[N]O_2 + k_{f3}[N][OH] \right)}{1 + Δt \left( k_{b1}[NO][N] + k_{b2}[NO][O] + k_{b3}[NO][H] \right)} \right\}_i
\]  
(5)

\[
[N]_{i+1} = \left\{ \frac{[N] + Δt \left( k_{f1}[N_2][O] + k_{f2}[NO][O] + k_{f3}[NO][H] \right)}{1 + Δt \left( k_{b1}[NO][N] + k_{b2}[NO][O] + k_{b3}[NO][H] \right)} \right\}_i
\]  
(6)

\[
[O_2]_{i+1} = \left\{ \frac{[O_2] + Δt \left( k_{b2}[NO][O] \right)}{1 + Δt k_{f2}[N]} \right\}_i
\]  
(7)

\[
[N_2]_{i+1} = \left\{ \frac{[N_2] + Δt \left( k_{b1}[NO][N] \right)}{1 + Δt k_{f1}[O]} \right\}_i
\]  
(8)
2.4 Pressure and temperature in the cylinder
The pressure in the cylinder $P_{\text{cyl}}$ is given by integrating Eq. (12). The temperature in the cylinder $T_{\text{cyl}}$ is given by Eq. (13).

\[
\frac{dP_{\text{cyl}}}{d\theta} = \frac{1}{V_{\text{cyl}}} \left( \kappa - 1 \right) \left( \frac{dQ_{\text{cyl}}}{d\theta} + \frac{dH_{\text{cyl}}}{d\theta} \right) - \kappa P_{\text{cyl}} \frac{dV_{\text{cyl}}}{d\theta}
\]  

(12)

\[
T_{\text{cyl}} = \frac{P_{\text{cyl}} V_{\text{cyl}}}{m_{\text{cyl}} R}
\]  

(13)

where, $\theta$ is the crank angle, $V_{\text{cyl}}$ is the volume of the combustion chamber in the cylinder, $\kappa$ is the specific heat ratio, $Q_{\text{cyl}}$ is the amount of net heat release, $H_{\text{cyl}}$ is the enthalpy change, and $R$ is the gas constant.

$Q_{\text{cyl}}/d\theta$ in Eq. (12) means the net heat release rate and it is given by Eq. (14). The first term of the right-hand side of Eq. (14) is the heat release by combustion. The second term is the heat loss to the cylinder wall. The Wall heat transfer coefficient was calculated by using Woschni’s correlation.

\[
\frac{dQ_{\text{cyl}}}{d\theta} = \frac{dQ_{\text{m}}}{d\theta} - \frac{dQ_{\text{m}}}{d\theta}
\]  

(14)

The supercharging system is assumed to be a constant pressure operation by using exhaust turbochargers. The turbine output $N_t$ and the compressor drive power $N_c$ are given by Eqs. (15) and (16).

\[
N_t = \frac{\eta_t}{2} c_v \rho_{\text{tg}} A_{\text{tg}} V_{\text{tg}}^3
\]  

(15)

\[
N_c = \frac{1}{\eta_c} m_{\text{comp}} \Delta h
\]  

(16)

where, $\eta_t$ is the adiabatic efficiency of the exhaust gas turbine, $c_v$ is the specific heat at constant volume, $\rho_{\text{tg}}$, $A_{\text{tg}}$, $V_{\text{tg}}$ is the density, the cross section, and the speed of the turbine passage gas, $\eta_c$ is the overall adiabatic efficiency of the compressor, $m_{\text{comp}}$ is the mass flow rate of the compressor passage air, and $\Delta h$ is the enthalpy difference between the inlet and the outlet of the compressor.

The mass flow rate of the air flowing into a scavenging air receiver is calculated by putting $N_t = N_c$. The scavenging pressure and the scavenging temperature are calculated by solving an equation of state by using this mass flow. However, the overall adiabatic efficiency of the compressor $\eta_c$ was fixed at 0.85. The adiabatic efficiency of the exhaust gas turbine $\eta_t$ was calculated automatically from the performance curve of the turbine.

3. Calculation characteristics
Because the experimental measurements of fuel atomization and combustion are difficult in large diesel engines, the values which are necessary in this simulation such as an
The initial diameter $D_{d0}$, an evaporation constant $K$, and a spray flame angle $\theta_f$ cannot be obtained. Then, calculations were performed under various combinations of these parameters, and suitable values were estimated by comparing the calculated engine performance such as the engine output and the NO exhaust rate with experiments. Figure 3 shows comparisons of calculated pressure and heat release rate histories in the cylinder with experiments using the actual marine diesel engine data. The main specifications of the engine are as follows: two-stroke cycle with seven cylinders, the rated output of 20000kW, the rated speed of 64.6rpm, the bore of 840mm, and the derating operation of 75% from the rated load. The rate of fuel injection was estimated from the fuel injection pressure history.

![Figure 3](image_url)

**Fig. 3.** Comparisons of calculated pressure and heat release rate histories in a cylinder with experiments. The injection start timing is $2.3^\circ$ ATDC. The injection period $\theta_{ip}=12^\circ$, $\theta_f=20^\circ$, $D_{d0}=12.3\mu m$, $K=0.0039 mm^2/s$, $\Delta \theta=0.01^\circ$, $n_{div}=100$.

Because the result of the element analysis of the fuel showed that carbon vs. hydrogen is equal to 86 vs. 14, it was assumed that $m$ vs. $n$ of $C_mH_n$ is 12 vs. 23.28. The measurement value of 40.63kJ/g was used as the heat of combustion of the fuel. All seven cylinders were calculated independently according to the firing order, and the cycle change of supercharging pressure was solved. If the initial cycle values of pressure in the cylinder and gas composition in the exhaust pipe became within the limits of a demand error ($1 \times 10^{-6}$ for the pressure, and $1 \times 10^{-5}$ for the concentrations), the calculation was judged to be converged. The calculated heat release shape agreed with the experiment very well during the main burning period from 2 to $20^\circ$. If after the spray collision to the cylinder wall was reduced to 30% of the value before the collision, the calculated result agreed with the experiment relatively well during the after-burning period from 20 to $40^\circ$. The difference is small. Because the actual fuel includes residual carbon and the calculation fixed the parameters such as $D_{d0}$ and $\theta_f$, this difference can be said to be the limit of this computational model.

Next, the fundamental characteristics of the computational model were investigated by unifying the calculation conditions except for the evaluation items. Figure 4 shows effects of the calculation interval $\Delta \theta$ and the number of subdivision $n_{div}$ on the NO exhaust rate of the whole engine $m_{NO}$. A uniform injection rate pattern was used for the fuel injection. The injection start timing was $2^\circ$ after TDC. The other conditions were the same as those of Fig. 3. Nine kinds of values, 0.01, 0.02, 0.025, 0.05, 0.1, 0.2, 0.25, 0.5, and 1$^\circ$ were used as $\Delta \theta$. This figure shows that if $\Delta \theta$ becomes small, the NO exhaust rate is converged on a steady value, and, if $n_{div}$ increases, the convergence improves. In the case of $n_{div}=100$, the
The difference between the values at $\Delta \theta = 0.2^\circ$ and $\Delta \theta = 0.01^\circ$ is 0.14%. It is found that the optimum number of subdivision is 100 because there is almost no difference between the values at $n_{\text{div}}=100$ and $n_{\text{div}}=1000$. In the case of $\Delta \theta = 0.01^\circ$ and $n_{\text{div}}=100$, the CPU time to converge was about 2 hours when AMD Opteron 2360SE 2.5GHz was used. The measurements showed that the CPU time was nearly proportional to $\Delta \theta^{-1.44}$. On the other hand, because the change of the computing time was small to $n_{\text{div}}$, it is found that this calculation technique is effective in shortening of the computing time.

Figure 4 shows the NO exhaust rate of the whole engine $m_{\text{NO}}$ as functions of the calculation interval $\Delta \theta$ and the number of subdivision $n_{\text{div}}$.

Figure 5 shows changes of the equilibrium concentrations of the minor gas components and the total reaction rates of all species in the free spray flame (fsf) and the wall spray flame (wsf) at $\Delta \theta=0.01^\circ$ and $n_{\text{div}}=100$. In the free flame, the rapid increment and the subsequent decrement in the concentrations are seen as decreasing of the spray flame region after the end of injection, $\theta \geq 14^\circ$. The concentrations decrease gradually in the wall spray flame.

Figure 5. Changes of the chemical equilibrium concentrations of the minor gas components and the total reaction rates of all species in the free spray flame and the wall spray flame.
Figure 6 shows changes of the concentrations of NO and N in the inside and outside of the spray. Although both NO and N begin to increase immediately after the flame generates, the remarkable changes appear after the end of fuel injection. In the free spray (fs), the chemical species concentrations in the spray reach the maximums at the time of extinction of the fuel spray. In the wall spray (ws) and the outside of the spray, they decrease after showing their peaks.

The flame temperature is one of the most important dominant factors of the chemical reaction. Then, in order to evaluate the effect of the flame temperature, a new parameter named the flame adiabatic degree $f_{ad}$ was defined. This parameter shows the influence of heat loss from the flame. If $f_{ad}$ is 1, the temperature is equal to the well-known adiabatic flame temperature. Figure 7 shows effects of $f_{ad}$ on $m_{NO}$, the maximum temperature of the free spray flame $T_{fs,max}$, and the maximum temperature of the wall spray flame $T_{ws,max}$. In this figure, $m_{NO}$ showed the local maximum at $f_{ad}=0.75$. Because $T_{fs,max}$ exceeded 4000K in...
the case of \( f_{ad} > 0.9 \), the physical-properties were calculated by the extrapolation method. From this reason, the calculation results in this range are the reference values. In these two sprays, there are the differences of fuel mass and gas volume which participate in the reaction. This is the reason why \( T_{wsf,\text{max}} \) exceeds always \( T_{wsf,\text{max}} \).

Figure 8 shows changes of the average concentration of NO in the cylinder \([\text{NO}]_{\text{cyl}}\), the flame temperature of the free spray and the wall spray \( T_{fsf} \) and \( T_{wsf} \), and the temperature in the cylinder \( T_{cyl} \) by using \( f_{ad} \) as a parameter. Both \( T_{fsf} \) and \( T_{wsf} \) reach their maximums in the cases of \( f_{ad} = 0.9 \) and \( f_{ad} = 0.75 \), but there is no clear difference in \( T_{cyl} \). NO of \( f_{ad} = 0.9 \) decreased after showing the maximum, and becomes lower than the value of \( f_{ad} = 0.75 \). After the flame extinguished, the reaction of NO was frozen and the concentration of NO converged on a steady value. One of the reasons is that \( \text{O}_2 \), \( \text{N}_2 \), \( \text{O} \), \( \text{OH} \), and \( \text{H} \) are not supplied newly to the reaction region of NO by the extinction of the flame which is the chemical equilibrium region. And another reason is the temperature fall in the reaction region. In order to investigate the characteristics under the conditions which do not exceed 4000K, \( f_{ad} \) is fixed at 0.85 from the subsequent calculation. If 15% of the fuel heat of combustion is lost because of cooling loss, the calculated concentration of NO (the 13% \( \text{O}_2 \) conversion, wet gas standard) was 2630ppm and the actual measured value was 2070ppm.

![Figure 8](image-url)

**Fig. 8.** Changes of the average concentration of NO in the cylinder \([\text{NO}]_{\text{cyl}}\), the flame temperature of the free spray and the wall spray \( T_{fsf} \) and \( T_{wsf} \), and the temperature in the cylinder \( T_{cyl} \) by using \( f_{ad} \) as a parameter.

Figure 9 shows effects of the injection period \( \theta_{ip} \) on \( m_{\text{NO}} \) and the thermal efficiency \( \eta \). \( \eta \) reached the maximum at \( \theta_{ip} = 13^\circ \), and \( m_{\text{NO}} \) reached the maximum at \( \theta_{ip} = 11^\circ \). If \( \theta_{ip} \) becomes very small, the fuel injection speed becomes large and most of the fuel droplets collide to the cylinder wall before they complete evaporation. As a result, the fall of the maximum temperature in the cylinder is caused because the heat release decreases. If \( \theta_{ip} \) is set more than 13\(^\circ\), it is possible to reduce a small amount of NO with the minimum decrement of \( \eta \).

Figure 10 shows effects of the initial droplet diameter \( D_{d0} \) on \( m_{\text{NO}} \) and \( \eta \). In the combustion under the evaporation rate controlling condition, \( \eta \) increases monotonously with decreasing of \( D_{d0} \), because if \( D_{d0} \) decreases, the maximum combustion pressure \( P_{\text{max}} \) and the indicated power will increase. The NO exhaust rate showed the peak at \( D_{d0} = 12.0 \mu \text{m} \). If \( D_{d0} \) decreases, the mass burning rate increases and the inside of the cylinder
becomes high temperature and high pressure. It is found that the concentration of NO decreases after the middle of combustion because the generation and subsequent decomposition reactions of NO become active at this time. This is similar to the cases of $f_{ad} = 0.75$ and $f_{ad} = 0.9$ in Fig. 8. Some reduction of NO can be promoted simultaneously with the improvement in $\eta$ by promoting the atomization of the spray. $P_{max}$ is 12.4MPa when $D_{d0}$ is 12.0$\mu$m, however, it reaches 13.0MPa at 11.1$\mu$m. It exceeds the allowable pressure. This means that there is a limit in the fine atomization.

![Fig. 9. Effects of the injection period $\theta_{ip}$ on the NO exhaust rate $m_{NO}$ and thermal efficiency $\eta$.](image)

![Fig. 10. Effects of the initial droplet diameter $D_{d0}$ on the NO exhaust rate $m_{NO}$ and the thermal efficiency $\eta$.](image)

Figure 11 shows effects of the spray flame angle $\theta_f$ on $m_{NO}$ and $\eta$. It is found that $m_{NO}$ decreases gradually after reaching the maximum at $\theta_f = 21^\circ$. On the other hand, $\eta$ increases monotonously to $\theta_f$. From these, $\theta_f$ can be said to be one of the important factors which dominate $m_{NO}$ and $\eta$. If $\theta_f$ becomes very large, the spray penetration speed becomes small and it decreases the number of fuel droplets which collide to the cylinder wall. As a result, the heat release increases, then the temperature in the cylinder increases, the formation and decomposition reactions of NO become active like the cases of $f_{ad}=0.75$ and $f_{ad}=0.9$ in Fig. 8. It is recommended that $\theta_f$ becomes more than $21^\circ$ by promoting the dispersion of a spray because $\eta$ decreases rapidly if $\theta_f$ becomes less than $21^\circ$. 
Figure 11. Effects of the flame angle $\theta_f$ on the NO exhaust rate $m_{NO}$ and thermal efficiency $\eta$.

Figure 12 shows effects of the mixing ratio of the wall spray $f_w$ on $m_{NO}$ and $\eta$. Here, $f_w$ is the ratio of the speed at which the wall spray mixes with surrounding gas to the speed at which the wall spray generates (quantity of the free spray which collides with the cylinder wall). $f_w=0$ means no mixing and $f_w=1$ means the wall spray mixes with surrounding gas immediately after the free spray collides with the cylinder wall. It is considered that the wall spray disappears shortly as the part of the spray is mixed with surrounding gas. There is a tendency in which $\eta$ is kept almost constant and $m_{NO}$ increases slightly against $f_w$. From this fact, it is found that the mixing ratio of the wall spray to surrounding gas is not significant compared with other factors.

4. Conclusions

In this study, the multi-region reactions model for spray combustion is proposed in order to predict the NO emission from large marine diesel engines. This model consists of the chemical equilibrium region corresponding to the spray flame boundary and two NO formation regions in the inside and outside of the spray. The characteristics of the numerical prediction by the model were investigated. As a result, the next conclusions were obtained. (1) The reaction calculation of NO by the subdivision is effective in shortening of the computing time, and the optimum number of the subdivision is 100.
(2) After combustion of the spray is completed, the reaction of NO is frozen and the concentration of NO is converged on a steady value.

(3) When the flame angle $\theta_f$ is 21°, the NO exhaust rate reaches the maximum. On the other hand, the thermal efficiency increases monotonously to $\theta_f$. It is recommended to make $\theta_f$ larger than 21°.

(4) The NO exhaust rate and the thermal efficiency have the maximum regarding the injection period. It is recommended to make the period larger than 13°.

(5) The NO exhaust rate has the maximum regarding the initial droplet diameter $D_{d0}$. The thermal efficiency increases monotonously as $D_{d0}$ reduces. Although it is recommended to make $D_{d0}$ smaller than 12μm, the lower limit exists from the relation with the allowable maximum combustion pressure.

(6) The influence of the mixing ratio of the wall spray to surrounding gas on the NO exhaust rate and the thermal efficiency is small.

References


