# The Effect of Combustion Parameters on the Nitric Oxide Emission in Direct Injection Type Diesel Engine

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This paper presents the investigation of influence factors on the output performance and the reduction of exhaust emission in the direct injection type diesel engine. In this work, the analysis of combustion products and combustion characteristics are investigated by numerical method and experiment under the various engine operating conditions. The combustion performance and exhaust emissions are analyzed in terms of the heat release, cylinder pressure and major exhaust emissions of engine. The accuracy of the prediction versus experimental data and the capability of the heat release, cylinder pressure and all the major exhaust emissions are demonstrated. The results of this study show that the combustion parameters have influence on the combustion processes and the nitric oxide emission in the direct injection type diesel engine. The nitric oxide concentration decreases with the increase of engine speed and the advance of injection timing.

**Key Words:** Fuel Properties, Combustion Characteristics, Exhaust Emission, Combustion Simulation

## 1. Introduction

Diesel engines are extensively used in road vehicles and marine transportation due to their superior fuel economy and higher performance over gasoline engines. Direct injection type diesel engines have occupied a dominant position in diesel engines for many years. However direct injection type engines have demerits of higher noise and louder vibration due to higher combustion pressure.

Diesel engine development engineers should have the knowledge on the combustion products, and the relationships between the emission factors of the combustion phenomena, fuel properties, mechanism of nitric oxide, and other factors on the combustion products in cylinder. The improvement of reduction of diesel engine emission, such as nitric oxide and particulate are also important. The activities concerned with the earlier development of all engines to improve the combustion efficiency and emission formations processes in combustion chamber have been extensively investigated (Fabio Bozza, 1990 ; Hiroyuki Hiroyasu, 1985.; Steven L. Plee, 1983 ; and Lee, 1995a, 1995b).

These papers deal with the effect of combustion parameters on the combustion characteristics of diesel engine. Therefore it is necessary to find the determination of influence factors on the nitric oxide concentration of diesel engine. In this point of a view, it is important to know the various factors on the exhaust emission in diesel engine.

In this study, the outcome of modeling results has shown less precise results comparing with the experiment, but it can clearly point out the effect of the operating conditions on the nitric oxide emission and engine performance.

Many simulation models have been developed for the processes of internal combustion engine. The model can be divided into two groups: steady-state model and unsteady-state or transient

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model. In this paper we used the unsteady-state model, where thermodynamic analysis and gas flow processes are simulated into two approaches such as quasi-linear and quasi-steady approach, in which engine thermo-fluid dynamics are treated by using the basic relations employing the filling and emptying method (Timothy P. Gardner, 1988 ; Hiroyuki Hiroyasu, 1985, etc.).

The objective of this study is to present the influence factors of combustion and exhaust emissions on the combustion characteristics and combustion products in a direct injection type diesel engine. The influence of combustion parameter and emission factors of the nitric oxide are investigated along with the parameters such as heat release, cylinder pressure, and the formation of nitric oxide of experimental engine. These simulations will provide design concepts and optimization studies in order to reduce the number of costly development tests.

## 2. Method of Investigation

### 2.1 The thermodynamic analysis

The engine performance is theoretically investigated by employing a filling and emptying model. Mathematical formulation of the model is derived by applying the first law of themodynamics. In order to study the effect of the combustion parameters on the nitric oxide emission and engine performance, the conservation of energy equation is used. It is assumed that the entire control volume consists of homogeneous equilibrium mixture of air and combustion products at each instant. The thermodynamic balance equation for the cylinder gas is given by

$$\frac{dE}{d\theta}=\frac{dQ}{d\theta}+\sum \dot{m}_i h_i-\frac{dW}{d\theta} \tag{1}$$

where,

$\frac{dE}{d\theta}$ ; the rate of change of total internal energy in the system

$\frac{dQ}{d\theta}$ ; the rate of heat transfer into the system through the system boundary.

$\dot{m}_i h_i$ ; the energy converted in or out of the system at flow rate $\dot{m}_i$.

$\frac{dW}{d\theta}$ ; the rate of mechanical work done by the system on its boundary.

The assumption of ideal gas and thermodynamic equilibrium are used. All the partial derivatives in the Eq. (1) can be evaluated from the gas property relationship. The internal energy of gas and the gas constant in the cylinder can be expressed by the function of pressure ($P$), temperature ($T$) and equivalence ratio ($\phi$) respectively by Foster(1985).

$$PV=mRT$$

$$V\frac{dP}{d\theta}+P\frac{dV}{d\theta}=RT\frac{dm}{dT}+mT\frac{dR}{d\theta}+mR\frac{dT}{d\theta} \tag{2}$$

$$u=u(T,\ P,\ \phi)$$

$$\frac{du}{d\theta}=\frac{\partial u}{\partial T}\frac{dT}{d\theta}+\frac{\partial u}{\partial P}\frac{dP}{d\theta}+\frac{\partial u}{\partial \phi}\frac{d\phi}{d\theta} \tag{3a}$$

$$R=R(T,\ P,\ \phi)$$

$$\frac{dR}{d\theta}=\frac{\partial R}{\partial T}\frac{dT}{d\theta}+\frac{\partial R}{\partial P}\frac{dP}{d\theta}+\frac{\partial R}{\partial \phi}\frac{d\phi}{d\theta} \tag{3b}$$

From the Eqs. (1), (2) and (3) the rate of gas temperature rise and the rate of gas pressure variation with respect to the time of crank angle are calculated by the solution of the ordinary differential equations.

$$\frac{dT}{d\theta}=\frac{C_o\frac{1}{V}\frac{dV}{d\theta}-D_o\frac{1}{m}\frac{dm}{d\theta}-E_o\frac{d\phi}{d\theta}+\frac{\sum Q+\sum mh}{m}}{\frac{\partial u}{\partial T}+\frac{A_oB_o}{T}} \tag{4}$$

$$\frac{dP}{d\theta}=\frac{\frac{dT}{d\theta}\left(\frac{1}{T}+\frac{1}{R}\frac{\partial R}{\partial T}\right)-\frac{1}{V}\frac{dV}{d\theta}+\frac{1}{R}\frac{\partial R}{\partial \phi}\frac{d\phi}{d\theta}+\frac{1}{m}\frac{dm}{d\theta}}{1-\frac{P}{R}\frac{\partial R}{\partial P}} \tag{5}$$

where,

$$A_o=\frac{P\frac{\partial u}{\partial P}}{1-\frac{P}{R}\frac{\partial R}{\partial P}},\ B_o=1+\frac{T}{R}\frac{\partial R}{\partial T},\ C_o=\frac{p\frac{\partial u}{\partial P}}{1-\frac{P}{R}\frac{\partial R}{\partial P}}-RT,$$

$$D_o=u+\frac{P\frac{\partial u}{\partial P}}{1-\frac{P}{R}\frac{\partial R}{\partial P}},\ E_o=\frac{\partial u}{\partial \phi}+\frac{P\frac{\partial u}{\partial P}}{1-\frac{P}{R}\frac{\partial R}{\partial P}}R\frac{dR}{d\phi},$$

The equivalence ratio of mixture can be calculated by considering that the total mass in the cylinder is the sum of the air mass flow through the intake and exhaust valve and mass fuel injected to the combustion chamber.

The combustion process has two districtively different modes, i. e., premixed-burning and diffusion-burning.

$$\dot{m}_{tb}=\dot{m}_p+\dot{m}_a \quad (6)$$

A phase proportionality factor $\beta$ is introduced. The phase proportionality factor $\beta$ is controlled by duration of the ignition delay ($ID$) and overall fuel/air (or mixture) equivalence ratio($\phi_t$).

$$\beta=\frac{m_p}{m_{tb}}=1-\frac{a\phi_t^b}{ID^c} \quad (7)$$

where $a$, $b$ and $c$ are suitable empirical constant and $ID$ is the ignition delay (D. N. Assanis, 1985 and N. Watson, 1984).

The combustion correlation are based on the diffusion-burning function $m_d(\theta_\tau)$ and the premixed-burning function $m_p(\theta_\tau)$ (Fabio Bozza et al., 1990),

$$m_d(\theta_\tau)=1-\exp(-C_{d1}\theta_\tau^{C_{d2}})$$
$$\dot{m}_d(\theta_\tau)=C_{d1}C_{d2}\theta_\tau^{(C_{d2}-1)}\exp(-C_{d1}\theta_\tau^{C_{d2}}) \quad (8)$$

$$m_p(\theta_\tau)=1-(1-\theta_\tau^{C_{p1}})^{C_{p2}}$$
$$\dot{m}_p(\theta_\tau)=C_{p1}C_{p2}\theta_\tau^{(C_{p1}-1)}(1-\theta_\tau^{C_{p1}})^{(C_{p2}-1)} \quad (9)$$

Therefore Eq. (6) become

$$\dot{m}_{tb}(\theta_\tau)=\beta\dot{m}_p(\theta_\tau)+(1-\beta)\dot{m}_d(\theta_\tau) \quad (10)$$

where $\theta_\tau$ is the time from ignition non-dimensionalized by the total time during the combustion process.

### 2.2 Equilibrium compositions

The equilibrium thermodynamic properties for the products of the combustion of fuel $C_\alpha H_\beta O_\gamma N_\delta$ and air are assumed. The products of combustion have influence on the equivalence ratio ($\phi$), temperature and pressure in a combustion chamber.

The appropriate balanced equation of chemical reaction may be written as,

$$\varepsilon\phi C_\alpha H_\beta O_\gamma N_\delta+0.210O_2+0.79N_2$$
$$\rightarrow \nu_1CO_2+\nu_2H_2O+\nu_3N_2+\nu_4O_2+\nu_5CO$$
$$+\nu_6H_2+\nu_7H+\nu_8O+\nu_9OH+\nu_{10}NO \quad (11)$$

Ten equations are obtained to determine the variables $\nu_1\sim\nu_{10}$. The Runge-Kutta Verner method is employed to solve this equation system.

### 2.3 The nitric oxide kinetics model

The amount of nitric oxide formation was found by applying the expanded Zeldovich mechanism to each element of the combustion processes. The temperature, pressure and air-fuel ratio in the premixed and diffusion combustion were used as the input data for the empirical equation used to calculate nitric oxide.

The mechanism of nitric oxide formations in the diesel engine is considered for three chemical reaction equations in connection with nitric oxide formation.

$$N+NO\overset{k_1}{\leftrightarrow}N_2+O$$
$$N+O_2\overset{k_2}{\leftrightarrow}NO+O$$
$$N+OH\overset{k_3}{\leftrightarrow}NO+H$$

The constants of reaction rate of positive direction $k_1$, $k_2$ and $k_3$, and the equilibrium rate $R_1$, $R_2$ and $R_3$ are given respectively as follows (Hiroyasu and Kodota, 1976),

$$k_1=1.55\times10^{13},$$
$$k_2=8.98\times10^3\times T\times\exp\left(\frac{-6520}{RT}\right),$$
$$k_3=4.2\times10^{13}, \quad (12)$$
$$R_1=k_1[NO]_e[N]_e$$
$$R_2=k_2[N]_e[O_2]_e$$
$$R_3=k_3[N]_e[OH]_e \quad (13)$$

where the subscript $e$ refers to equilibrium conditions.

By using the steady-state approximation for the concentration of N atom and assuming equilibrium O and OH concentration, the rate of change nitric oxide formation is given by,

$$\frac{d(NO)}{dt}=\frac{2R_1(1-X^2)}{X\dfrac{R_1}{R_2+R_3}+1},\quad X=\frac{[NO]}{[NO]_e} \quad (14)$$

### 2.4 Heat transfer

The model used for the rate of heat transfer is based on the consideration of convective and radiative heat transfer in the normal way for the combustion chamber.

$$\frac{dQ_w}{dt} = h_c A (T_g - T_w) + \zeta A \sigma (T_g^4 - T_w^4) \tag{15}$$

where,

$h_c$ ; gas to wall heat transfer coefficient

$T_g$ and $T_w$ ; gas and combustion wall surface temperature respectively

$A$ ; total area for heat transfer

$\zeta$ ; emissity

$\sigma$ ; Stephan-Boltzmann constant

The heat transfer between the cylinder gases and the inner wall surface is computed using the instantaneous gas temperature and the heat transfer coefficient ( Woschni, 1967).

$$h_c = \frac{C_1 P^{0.8}}{B^{0.2} T^{0.53}} \left[ C_2 V_p + C_3 \frac{V_s T_r}{P_r V_r} (P - P_{mot}) \right]^{0.8} \tag{16}$$

where,

$P$ ; instantaneous cylinder pressure

$B$ ; cylinder bore

$T$ ; instantaneous gas temperature

$V_p$ ; swept volume

$V_r$, $T_r$ ; reference condition, such as inlet valve closure

$P_{mot}$ ; motoring pressure

The motoring pressure is evaluated by assuming that the compression and expansion process are modelled by polytropic process.

$$\frac{P_{mot}}{P} = \left( \frac{V_r}{V} \right)^n$$

The $C_1$, $C_2$ and $C_3$ (Hiroyasu, 1985) are empirical constants that can be adjusted for local variation due to the intake swirl, the combustion chamber geometry and the radiation effect.

where,

$C_1 = 0.13$

$C_2 = 2.28$ (compression, combustion and expansion)

$C_3 = 0$ (compression and scavenging phase)

$C_3 = 3.24 \times 10^{-3}$ (direct-injection engines)

Annand (1963) has proposed a radiation term based on the average bulk temperature. The primary sources of radiative heat transfer in a diesel engine are the high temperature burned gasses and soot particles,

$$\frac{dQ_{rad}}{d\theta} = BA (T_g^4 - T_w^4) \tag{17}$$

where,

$B$ ; empirical radiation constant

$A$ ; total area for heat transfer

$T_w$ ; inside wall surface temperature of cylinder head, piston or linear

$T_g$ ; average bulk gas temperature

During the combustion process,

$B = \zeta\sigma$

where,

$\zeta$ ; emissity depending on the engine speed and load

$\sigma$ ; Stephan Boltzmann constant

The Eq. (17) become,

$$\frac{dQ_{rad}}{d\theta} = \zeta \sigma A' (T_g^4 - T_w^4) \tag{18}$$

## 3. Simulation Procedures

The models incorporate the above assumptions into the equation of energy. These equations are solved numerically using the predictor-corrector iterative technique to obtain the gas temperature, pressure, rate of heat release, and the rate of pressure rise, etc. as a function of crank angle.

The nitric oxide emission is simulated during the period of heat release and expansion process. In general, the consituent of nitric oxide appears during the mid-period of combustion. The difficulties for calculating nitric oxide emission are caused by the short of heat release.

For steady state operation, the differential equation must be solved simultaneously and must give a periodic solution. Initially, the data describing all geometric features of the engine are read. All fixed operating parameters such as wall temperature, and ambient temperatures are specified for each cycle calculation.

## 4. Experimental Apparatus and Procedures

The basic engine used in this work is a four stroke cycle, single cylinder, and naturally aspir-

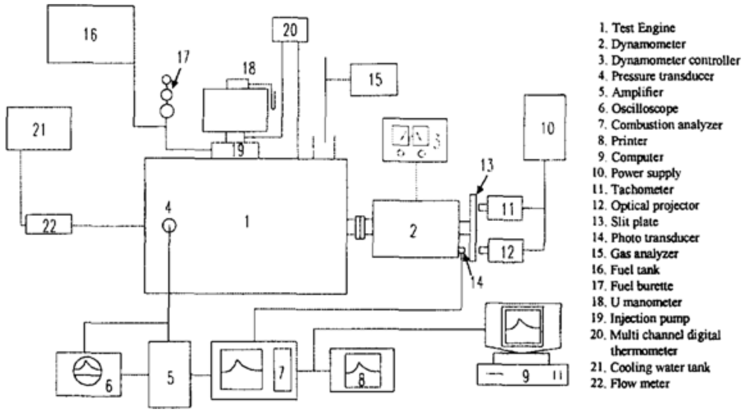



**Fig. 1** Schematic diagram of experimental apparatus

ated direct injection diesel engine with displacement of 1425cc. The engine has 17.4 compression ratio and rated power output of 7.36kW at 1200 rpm.

An experimental apparatus is shown schematically in Fig. 1. The tests were conducted on a single cylinder diesel engine coupled to an eddy current dynamometer. Combustion analyzer was composed of the pressure measuring device and the crank angle detector system. The cylinder pressure of engine was obtained by using a high pressure transducer and a crank angle detecting system.

In order to obtain the variation of combustion characteristics of diesel engine, the engine was investigated under the following conditions;

Engine speed ; 800～1500 rpm

Fuel injection opening pressure ; 170 bar

Cooling water temperature ; 50℃, 60℃, 70℃

Injection timing ; 220, 150, 100, 50 before TDC

## 5. Results and Discussion

Figure 2 shows a typical cylinder pressure diagram illustrating the difference between the measured cylinder pressure and those synthesized by the mathematical model. The agreement between the simulation and experimental results is good for the case of injection timing 15° BTDC and engine speed 1000 rpm.

During the combustion period, the rate of heat release is positive, and the burning process in two distinguishable stages as shown in Fig. 3. In the

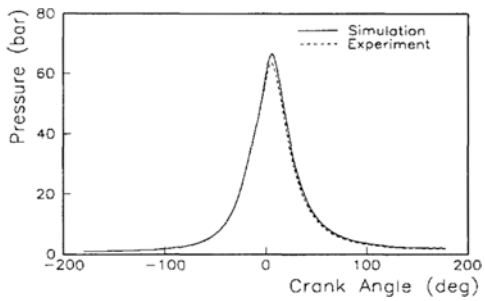



**Fig. 2** Comparison of simulation and experiment results for cylinder pressure

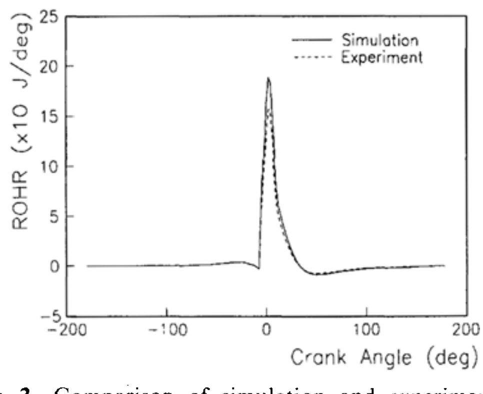



**Fig. 3** Comparison of simulation and experiment results for rate of heat release

first stage, the rate of heat release is generally very high for only a few degrees of crank angle. This corresponds to the period of rapid cylinder pressure rise because of the premixed combustion. The second stage corresponds to a period of gradually decreasing the rate of heat release and lasts for about 45 degrees of crank angle. This is the main heat release period and called the diffusion combustion. Diffusion process is the important factor to control the combustion process in diesel engine. In operating condition of engine speed, the rate of fuel injection, fuel properties and the injection pressure of fuel give an implication on the rate of heat release.

Figures 4~6 present the simulation results on the cylinder pressure, rate of heat release and the rate of pressure rise under the various ignition timing. The injection timing has a strong effect on the pressure and heat release pattern. The advance of injection timing brings about the increase of the cylinder pressure, and combustion characteristics.

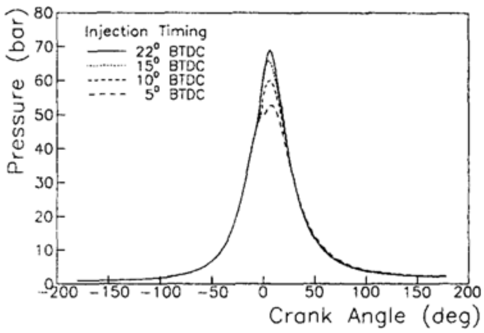



**Fig. 4** The effect of injection timing on the cylinder pressure

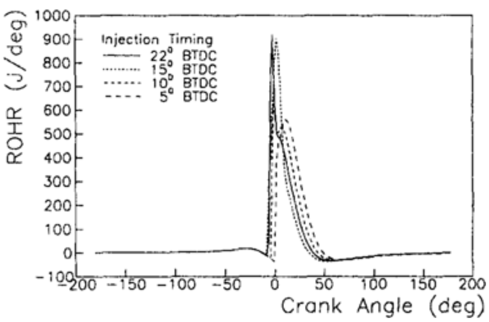



**Fig. 5** The effect of injection timing on the rate of heat release

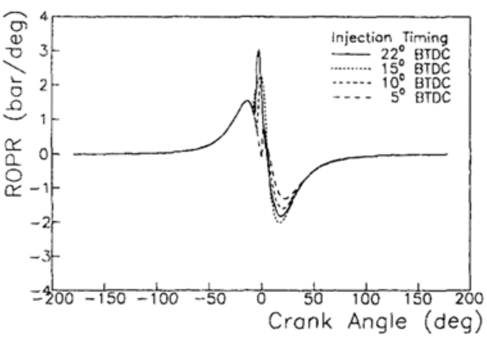



**Fig. 6** The effect of injection timing on the rate of pressure rise

Figure 7 shows a comparison of engine pressure with the effect of fuel injection amount. The peak pressure increased with injection fuel due to its increase in the amount rate of heat release in the combustion chamber. The cylinder perssure curve was shifted to the right and the peak pressure was increased slightly with the increase of injection fuel. As shown in Fig. 8, the rate of heat release increases in accordance with the injection fuel in the cylinder. The ignition delay, the period between the start of injection and the start of combustion are different from each amount of fuel injection. In this case, as already reported by Hiroyasu (1976), the reference amount of fuel in the engine was 40 mg per cycle at 1200 engine

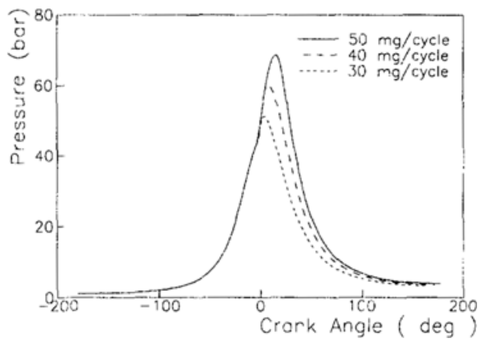



**Fig. 7** The effect of fuel injection amount on the cylinder pressure at engine speed 1200 rpm

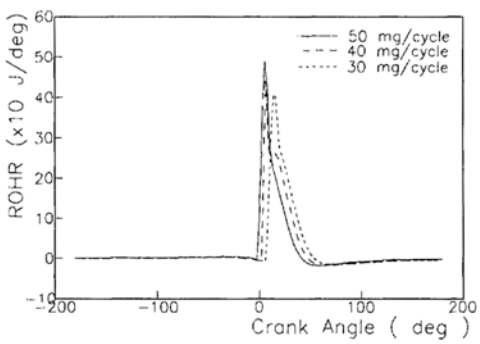
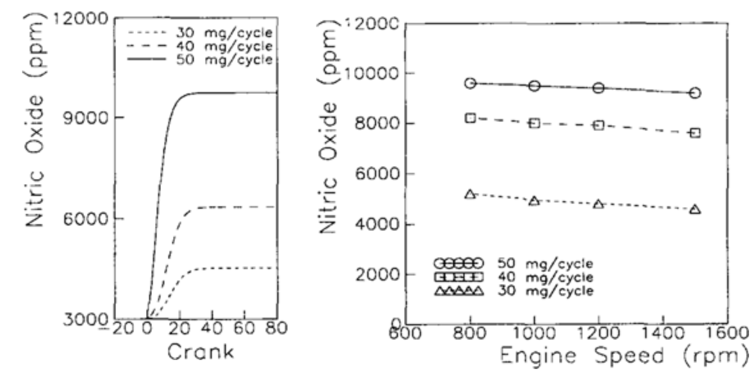



**Fig. 8** The effect of fuel injection amount on the rate of heat release at engine speed 1200 rpm




(a) Simulation results at 800 rpm (b) Experiment result

**Fig. 9** The effect of fuel injection on the nitric oxide emission

speed. As shown in figure, the increase of amount of fuel injection brings about the increase in the rate of heat release and the rate of pressure rise in the combustion chamber. Also the increase of fuel injection amount results in the increase of combustion energy in the cylinder. The cylinder pressure and the rate of heat release are improved by the increase of injected fuel.

Figure 9 shows the effect of fuel injection amount on the nitric oxide emission. It can be seen that, the increase of fuel amount in the combustion chamber will increase the formation of nitric oxide in the engine cylinder. The formation of nitric oxide increase because of nitric oxide is related with the premixed combustion. The cylinder temperature increases monotonically with the increase of fuel injection amount. Higher peak temperatures results in the increase of heat loss and increase of the nitric oxide emission.

The influence of engine speed on the mean value of nitric oxide in the engine cylinder is shown in Fig. 9(b). It can be seen that nitric oxide emission decreases with the increase of engine speed. The behaviour of this event can be explained by the combustion duration. At lower engine speed, the reactive gas provides a longer time for the nitric oxide to form and this causes in higher nitric oxide concentration in the combustion chamber.

The results suggest that the increase of fuel quantity brings about the higher concentration of nitric oxide as shown above figure. In addition, the rate of heat release of combustion products

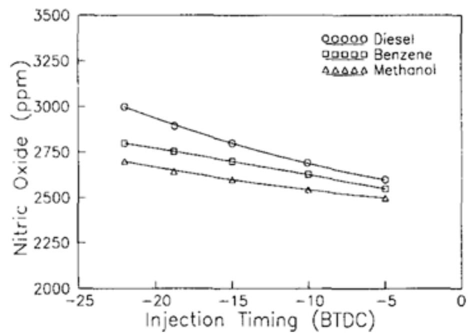



**Fig. 10** The effect of fuel injection timing on the nitric oxide at different fuel properties

exhaust emissions, dissociation and energy losses are dependent on the temperature and pressure of the cylinder. In this case the concentration of nitric oxide is affected by the heat release and gas temperature. It is considered that the combustion chamber type, surface temperature of combustion chamber and swirl flows in the engine greatly affect the performance of combustion system. The gas temperature of engine cylinder increases uniformly with the increase of fuel injection amount.

Figure 10 shows the effect of injection timing on the nitric oxide emission at engine speed 1500 rpm. As shown in figure, the nitric oxide emission of alternative fuels is lowered than that of diesel fuel. This can be explained by lower combustion temperature in alternative fuel.

Figure 10 also shows that, the concentration of nitric oxide is decreased in the retard of injection timing because of lower gas temperature in engine cylinder. The advance of injection timing have

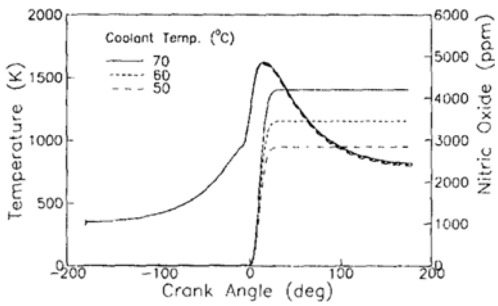



**Fig. 11** The effect of cooling temperature on the nitric oxide

influenced on the increase of fuel injection amount into the engine cylinder. The rapid increase of gas temperature due to the advance of injection timing results in a large amount of fuel evaporation before ignition. Therefore the optimum control of injection timing is very important to achieve an improvement of combustion process. The duration of ignition delay is one of the most important criteria, having great effects on the combustion process of engine, noise and the exhaust emission. As shown in figure, the ignition delay is a reduction factor of exhaust emission and engine noise such as nitric oxide and combustion noise. The combustion duration plays an important role in the rise of gas pressure and temperature. The effects of injection timing on the decrease of nitric oxide are similar in the results reached by M. Tsukahara, et al. (1989).

The late start of combustion due to retarding injection timing reduced the peak combustion temperature level. As the retard of fuel injection timing, combustion process progressively moves into the expansion process. This related to the decrease of local temperature in cylinder. Therefore, the variation of injection timing is an important factor that determines the level of nitric oxide concentration of exhaust emission. Further advance of the injection timing causes an increase in the peak temperature near the end of fuel evaporation.

The effects of coolant temperature on the gas temperature and nitric oxide of diesel engine at 1500 rpm of engine speed are shown in Fig. 11. The increase of the coolant temperature results in a small effect on the cylinder temperature but it increase the nitric oxide in the combustion chamber. The increase of coolant temperature brings about increasing the cylinder wall temperature. This results can be explained by the lower wet wall in the combustion chamber caused by higher coolant temperature. It is found that the increase of cylinder wall temperature is strongly dependent on the coolant temperature. When the coolant temperature is decreased, the nitric oxide concentration is decreased greatly as shown in figure. It can be estimated that gas temperature of engine influenced on the evaporation of spray.

## 5. Conclusions

The effect of combustion parameters on the nitric oxide emission combustion performance in direct injection type diesel engine have been investigated. This investigation deals with the effects of combustion parameter on the nitric oxide such as injection timing, engine speed, compression ratio and the amount of fuel injection.

The nitric oxide formation can be reduced by decrease of the peak pressure and the temperature in cylinder due to the retarding of fuel injection timing. An increase in the engine speed at given injection timing decreases the concentration of nitric oxide in the engine exhaust.

The increase of fuel injection amount increases the gas pressure and nitric oxide emission. Consequently, the combination of low injection amount with the retardation of injection timing reduces the nitric oxide concentration in the exhaust emission.

The concentration of nitric oxide increase in accordance with increase of coolant temperature under the constant operating condition.

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