

Improvement of Combustion Stability under Cold Ambient Condition by Mixture Control

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ABSTRACT

For diesel engine, lower compression ratio has been demanded to improve fuel consumption, exhaust emission and maximum power recently. However, low compression ratio engine might have combustion instability issues under cold temperature condition, especially just after engine started.

As a first step of this study, cold temperature combustion was investigated by in-cylinder pressure analysis and it found out that higher heat release around top dead center, which was mainly contributed by pilot injection, was the key factor to improve engine speed fluctuation.

For further understanding of combustion in cold condition, particularly mixture formation near a glow plug, 3D CFD simulation was applied. Specifically for this purpose, TI (Time-scale Interaction) combustion model has been developed for simulating combustion phenomena. This model was based on a reasonable combustion mode, taking into account the characteristic time scale of chemical reactions and turbulence eddy break-up. In addition, parameters of the ignition model and computational grids near glow plug were improved to apply under cold start conditions.

As a result, the result of this study reveal that controlling an equivalence ratio and temperature surrounding glow plug appropriately by multiple pilot injections was contributing to the improvement of combustion stability under cold start condition.

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INTRODUCTION

Diesel engine is the promising technology for the global environment thanks to its relatively high thermal efficiency thus less CO₂ emission. In recent years, compression ratio of passenger diesel engine has been decreasing to improve fuel consumption, output power and emissions [1, 2, 3, 4, 5]. However, low compression ratio may have combustion instability issues in low temperature condition. As a measure, an optimization of combustion chamber design, reduction of cooling loss, an in-cylinder gas temperature increase by exhaust gas recirculation and an air-fuel mixture distribution improvement by fuel injection strategy modification have been reported [6,7]. Concerning an air-fuel mixture distribution, recent studies have shown that a multiple injection was effective to improve in-cylinder mixture distribution because of relatively low fuel tip penetration. Also it was reported that a glow plug was the key component for combustion stability in low temperature condition and ignition was started surrounding glow plug by transparent

engine test results [8,9]. However, it has not been reported the effectiveness of a multiple injection focusing on an air-fuel mixture and temperature distribution improvement near glow plug. The purpose of this study was to clarify the relationship between an air-fuel mixture distribution and multiple injection modification by both experiment and 3D CFD simulation especially investigating closely near glow plug in low temperature condition.

TEST APPRATUS AND METHOD

Table 1 shows test engine specifications applied in this study. The test engine was 4 cylinder direct injection diesel engine and compression ratio was 15.0:1. In order to investigate the relationship between multiple injection strategy and combustion stability improvement in low temperature condition, combustion analysis was conducted using the special glow plug that was combined with in-cylinder pressure sensor. The relationship between surface temperature of glow plug and applied voltage was confirmed preliminary to the engine tests so that surface temperature of



glow plug was controlled to the target value by voltage control applying this relationship. Actual surface temperature of glow plug was also monitored by means of thermocouple embedded under tip of the heater of glow plug. A glow plug was located top of the combustion chamber as seen in Fig. 1 in this engine. Before experimental tests were conducted, the engine had been soaked in -15°C ambient temperature until coolant and lubricant oil temperature sank and were equal to the ambient temperature. The engine was motoring in constant speed by dynamometer and fuel was injected in designated injection patterns.

Table 2 shows the properties of test fuel. This study was specifically targeted to improve combustion stability in low temperature condition, therefore, Cetane number was set at relatively low level 46 as a severe condition.

Table 1. Test engine specifications

Engine Type	4 Cylinder Di Diesel
Displacement	2,488 cm ³
Bore	89.0 mm
Stroke	100.0 mm
Compression Ratio	15.0:1
Fuel Injection Equipment	Common Rail System
Nozzle Hole Diameter	0.158 mm
Number of Nozzle Hole	8
Swirl Ratio	3
Type of Glow plug	Rapid Ceramic
Turbocharger	VGT

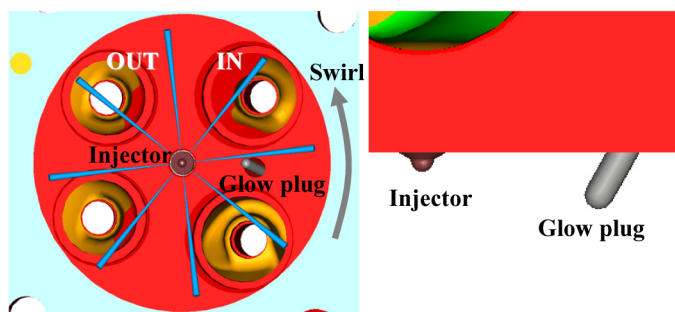


Fig. 1. Layout of glow plug on cylinder head

Table 2. Property of test fuel

Type of Fuel		Diesel
Cetane Number		46
Density		0.8321 g/cm ³
Viscosity		1.729 mm ² /s
Sulfur Content		10 ppm
Distillation	IBP	173 °C
	50%	240 °C
	FBP	301 °C

3D CFD SIMULATION

In this study, to investigate in-cylinder combustion under low temperature condition in detailed, TI (Time-scale Interaction) combustion model has been modified and applied [10]. TI combustion model was designed to be able to adapt various operating conditions of diesel engines. It consists of two kinds of combustion models; UCFM (Universal Coherent Flamelet Model) [11], which has been developed to predict mainly premixed combustion, and CTC (Characteristic Time-scale Combustion) model [12], which has been mainly dedicated to simulate diffusion combustion. Those two models were selected and applied by factoring the characteristic time scales of the chemical reactions and turbulent mixing into the scope of application of each model. In order to simulate low temperature engine starting operation condition, further improvement of the accuracy of ignition model was implemented and also a specific high temperature wall was introduced to imitate a glow plug and also surrounding computational grids of glow plug was modified.

Time-Scale Interaction Combustion Model

As outlined in Fig. 2, in the initial stage of diesel combustion, the air-fuel mixture that forms during the ignition delay begins to burn, thus consuming the oxygen in the fuel jet. After most of the oxygen in the fuel jet is consumed, the fuel is subsequently consumed by diffusion combustion in the region where the fuel jet surface mixes with the surrounding air. Dec [13] made this process clear on the basis of visualization experiments using laser-sheet imaging. Because the interior of the fuel jet is fuel-rich, the laminar burning velocity is extremely slow and the flame cannot sustain combustion. Consequently, the characteristic time scale of the chemical reactions is greater than that of macroscopic turbulent mixing, resulting in the local formation of a well-stirred combustion field. The fuel-rich mixture, which is generated by air entrainment at the upper stream area of the fuel jet, then goes into the fuel-rich combustion area [14]. These reasons presumably explain the occurrence of fuel-rich combustion that is rate-controlled by the chemical reactions including auto ignition.

Therefore, the new TI combustion model applies the UCFM to combustion in the region where the characteristic time scale of the chemical reactions is smaller than that of turbulence mixing and turbulence is the rate-controlling parameter. The CTC model is applied to combustion in the reaction rate-controlled region where the characteristic time scale of the chemical reactions is dominant owing to the influence of a fuel-rich or -lean mixture, among other factors. The CTC model determines the fuel consumption rate on the basis of the characteristic time scales of the chemical reactions and turbulence. The application range of the CTC model was defined on the basis of a non-dimensional Damköhler number Da , consisting of these two characteristic time scales. This approach enables the two combustion models to be applied to the regions matching the actual

combustion fields. Da is expressed as a ratio of the integral characteristic time scale of turbulent flow τ_t to the characteristic time scale of the chemical reactions τ_{kin} (i.e., $Da = \tau_t/\tau_{kin}$). When Da is less than a certain value, it is assumed that the progress of combustion contributes more to the characteristic time scale of the chemical reactions than to that of turbulent flow. Accordingly, the scope of application of the CTC model is defined as

$$Da \leq a \quad (1)$$

where $a = 1.0$ when the characteristic time scales of turbulence mixing and the chemical reactions are equal.

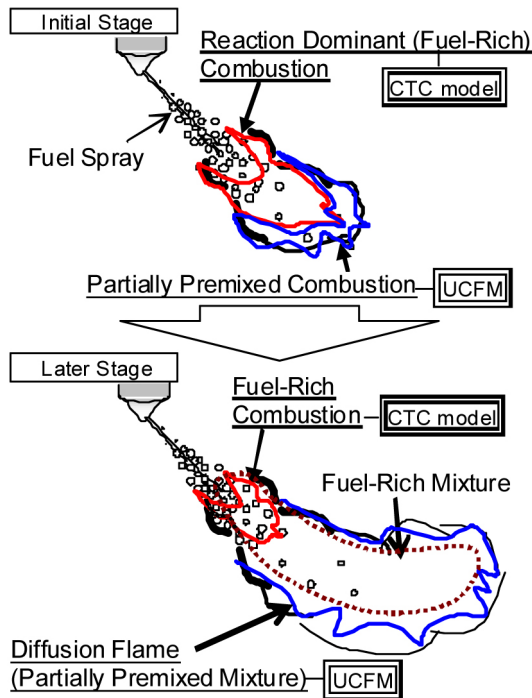


Fig. 2. Concept of TI combustion model

Combustion Model in the Turbulence Rate-Controlled Region

The results of Dec's visualization experiments [13] showed that the initial partially premixed mixture at an equivalence ratio from 0.78 to 2.02 forms the diffusion flame, which is the main combustion mode in diesel engines. Because the combustion reaction zone in this region is extremely thin, the diffusion combustion region in a diesel engine can be considered as an agglomeration of laminar flamelets using a flame front model [15,16,17]. Accordingly, the UCFM [11] developed by the authors as a high-accuracy premixed combustion model was applied to combustion in the region where turbulence is the rate-controlling parameter.

In the UCFM, the flame area is expressed macroscopically in terms of the flame area density Σ , and flamelets are expressed by solving the transport equation of Σ . The transport equation of Σ is written as

$$\frac{\partial \Sigma}{\partial t} + \frac{\partial u_i \Sigma}{\partial x_i} = \frac{\partial}{\partial x_i} \left(\frac{\nu_t}{\sigma_c} \frac{\partial \Sigma}{\partial x_i} \right) + S - D \quad (2)$$

where ν_t is the coefficient of turbulent viscosity, σ_c is the turbulent Schmidt number, S is the source term and D is the extinction term. A model of the source term S was newly created for the UCFM, and the model proposed by Bouldier et al. [18] was adopted for the extinction term D .

The source term S in Eq. (2) was modeled as shown below using the flame area growth S_T attributed to turbulent combustion and the growth S_L attributed to laminar combustion.

$$S = S_T + S_L \quad (3)$$

The turbulent combustion source term S_T is expressed in the UCFM as

$$S_T = \alpha_0 \sqrt{Ret} \frac{\varepsilon}{k} \Gamma \Sigma \quad (4)$$

where α_0 is a model constant, k , ε and Ret indicate the turbulent energy, the turbulence dissipation rate and the turbulent Reynolds number, respectively, and Γ denotes the intermittent turbulence net flame stretch (ITNFS) model [19], which expresses the stretching and quenching of the flame front by turbulence.

The source term S_L attributed to laminar combustion is expressed as shown in the following equation using the ratio of the combustion gas temperature T_b to the unburned gas temperature T_u and its product with the laminar burning velocity U_L , as well as the attenuation function expressed as the Karlovitz number and taking into account the Kilmo-Williams criterion [20].

$$S_L = \beta_0 \exp(-\beta_1 Ka) \frac{T_b}{T_u} U_L \Sigma^2 \quad (5)$$

where β_0 and β_1 are model constants.

In the UCFM, the fuel consumption rate ω_{Fchem} is derived as

$$\omega_{Fchem} = \min(Y_{TF}, Y_{TO2}/\zeta_{st}) \rho_u U_L \Sigma \quad (6)$$

using the unburned mixture density ρ_u , the hypothetical fuel mass fraction Y_{TF} and the hypothetical oxygen mass fraction Y_{TO2} , both of which will be explained later, the laminar burning velocity U_L and the flame area density Σ . Here, the notation ζ_{st} is the ratio of the oxygen mass fraction to the fuel mass fraction relative to a stoichiometric mixture ratio. It was hypothesized that the laminar burning velocity U_L of high carbon number fuels is approximately the same. The U_L of

iso-octane, whose flame speed is almost equivalent to chemicals contained in diesel fuel, was used in the calculations.

Combustion Model in the Reaction Rate-Controlled Region

In the region of reaction rate-controlled combustion, it is assumed that there are various fields where the characteristic time scale of the chemical reactions is greater than that of turbulent mixing, including a high-turbulence field and fuel-rich or -lean fields. For diesel combustion in particular, a fuel-rich field is mainly hypothesized, making it necessary to consider the mixing time of the un-reacted fuel and oxygen when determining the chemical reaction time. In addition, because the characteristic time scale of the chemical reactions increases due to the low oxygen concentration, it is necessary to consider both characteristic time scales simultaneously. Therefore, it was decided to use the CTC model [12], which takes both characteristic time scales into account locally, as the combustion model in the reaction rate-controlled region. The mixing of the un-reacted fuel and oxygen is expressed using the characteristic time scale of turbulence τ_t . Using the characteristic time scale of the chemical reactions τ_{kin} , the rate of reduction of the mass fraction of each species, dY_i/dt , can be expressed as shown in the following equation.

$$\frac{dY_i}{dt} = \frac{Y_i - Y_i^*}{\tau_{kin} + f\tau_t} \quad (7)$$

where Y_i^* is the momentary equilibrium concentration of each species. The characteristic time scale of the chemical reactions τ_{kin} and that of turbulence τ_t are expressed as an Arrhenius type function and as a vortex dissipation type function, respectively, as indicated in the following equations.

$$\tau_{kin} = A^{-1} [Fuel]^{0.75} [O_2]^{-1.5} \exp(E_A/RT) \quad (8)$$

$$\tau_t = cm_2 \frac{k}{\varepsilon} \quad (9)$$

Values of $A = 7.68 \times 10^9$, $E_A = 77.3$ kJ/mol and $cm_2 = 0.142$, as determined by Kong et al., [12] were used as the respective model constants. The notation f is a function having a value between 0 and 1, which is determined by the progress of combustion. It is expressed as shown below using the oxygen concentration Y_{O_2} and the hypothetical oxygen concentration Y_{TO_2} , which will be explained later.

$$f = \frac{1 - e^{-r}}{0.632} \quad (10)$$

$$r = 1 - \frac{Y_{O_2}}{Y_{TO_2}} \quad (11)$$

Based on the foregoing discussion, the fuel consumption rate in the TI combustion model is derived with Eq. (6) in the turbulence rate-controlled region and with Eq. (7) in the reaction rate-controlled region.

Mass Fraction Equations

The combustion gas temperature T_b and the unburned gas temperature T_u are needed to find the laminar burning velocity and the fuel consumption rate in each combustion mode. To calculate these temperatures, the mass fraction of each species in the burned region is found, in addition to solving the transport equations for the mass fraction of each species in the unburned region as well as their mass conservation equations and enthalpy conservation equations. The unburned region is defined as a hypothetical space in the TI combustion model so that the burned and unburned regions can be considered with one cell. The mass conservation equation for the fuel and the transport equation for the hypothetical unburned fuel are shown below.

$$\frac{\partial \rho Y_F}{\partial t} + \frac{\partial \rho u_i Y_F}{\partial x_i} = \frac{\partial}{\partial x_i} \left(\frac{\rho v_i}{\sigma_c} \frac{\partial Y_F}{\partial x_i} \right) + \omega_{Fchem} + \omega_{Fevap} \quad (12)$$

$$\frac{\partial \rho Y_{TF}}{\partial t} + \frac{\partial \rho u_i Y_{TF}}{\partial x_i} = \frac{\partial}{\partial x_i} \left(\frac{\rho v_i}{\sigma_c} \frac{\partial Y_{TF}}{\partial x_i} \right) + \omega_{Fevap} \quad (13)$$

where Y_F is the fuel mass fraction, Y_{TF} is the hypothetical unburned fuel mass fraction, U_L is the laminar burning velocity, ω_{Fchem} is the fuel consumption rate, and ω_{Fevap} is the increase in the fuel quantity due to evaporation. Y_{TF} is a variable that varies only due to fuel transport and is not changed by combustion.

Hence, $Y_{TF} = Y_F$ in the unburned state and $Y_{TF} > Y_F$ in the burned region.

The unburned combustion gas temperature T_u is derived by calculating it with the following enthalpy equation.

$$\begin{aligned} & \frac{\partial \rho h_u}{\partial t} + \frac{\partial \rho u_i h_u}{\partial x_i} \\ &= \frac{\partial}{\partial x_i} \left(\frac{\rho v_i}{\sigma_c} \frac{\partial h_u}{\partial x_i} \right) + \frac{\rho}{\rho_u} \frac{\partial}{\partial x_i} q_{ui} + \frac{\rho}{\rho_u} \left(\frac{\partial p}{\partial t} + u_i \frac{\partial p}{\partial x_i} \right) + \rho \varepsilon \end{aligned} \quad (14)$$

where p indicates the pressure and q_{ui} is a heat diffusion vector.

Optimization of Constant of the Shell Model

In order to predict ignition phenomenon in low temperature starting condition precisely, constants of shell model [21] was optimized using genetic algorithm. Objectives of optimization were experimental results of Ciezki's [22] shock tube ignition delay time using n-Heptane

and air mixture and for the further compensation, Tsurushima's [23] prediction results by means of chemical reaction model was applied. Optimization was carried out on wide range of conditions; an ambient pressure ranging from 1 to 8MPa, an equivalence ratio ranging from 0.5 to 2 and EGR ratio ranging from 0 to 50 %, considering the adaptation of various operation conditions. The modeFRONTIER, general-purpose optimization tool, was applied. To minimize objective function $\Delta\tau$ in formula (15), parameters of shell model (Af1, Af2, Af3, Af4, A1, A2, A3, Aq, Ab, At, Ef1, Ef2, Ef3, Ef4, E1, E2, E3, Eq, Eb, Et, X1, X3, X4, Y1, Y3, Y4) was optimized.

$$\Delta\tau = \sum (\tau_i - \tau_{shell}) \quad (15)$$

where τ is the ignition delay measured by Ciezki or calculated by means of Tsurushima's model, τ_{shell} is the calculated ignition delay using the Shell model. Fig. 3, 4 illustrates one of examples of calculated ignition delay before and after optimizations. In case of calculation results of before optimization deviated from measurements, while, after optimization, predicted values were in good agreement with measurements including cold temperature conditions.

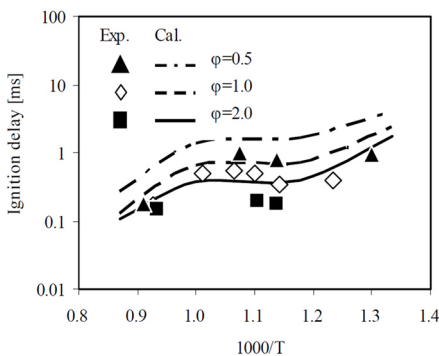


Fig. 3. Comparison of measured and calculated Ignition delay by using original shell parameter at an initial pressure of 4MPa, and equivalence ratios of 0.5, 1.0, 2.0

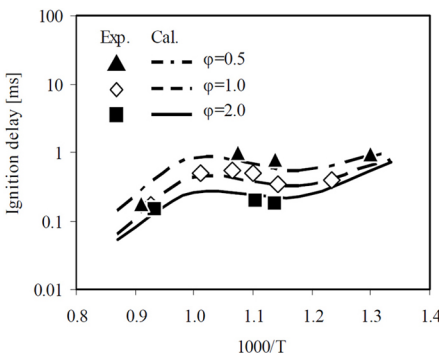


Fig. 4. Comparison of measured and calculated Ignition delay by using optimized shell parameter at an initial pressure of 4MPa, and equivalence ratios of 0.5, 1.0, 2.0

CALCULATION METHOD

Outline of Model

Calculations were performed in this study with the STAR-CD, general-purpose CFD analysis code, in order to validate the modified TI combustion model. The Renormalization Group (RNG) k- ϵ model [24], was used as the turbulence model. In the discretization procedure, the Implicit scheme was used as the time propagation method, the Monotone Advection and Reconstruction Scheme (MARS) was used to discretize the convection term, and the Pressure-Implicit with Splitting of Operators (PISO) algorithm was used as the solution for velocity and pressure.

The physical models used in this study are listed in Table 3. The Kelvin-Hemholz / Rayleigh-Taylor (KH-RT) hybrid model [25] was used as the spray model. This model combines the KH model of liquid jet breakup with the RT model of the breakup of liquid drops beyond the breakup length. Fuel viscosity that affect the breakup is adjusted with temperature. A shell model [21] was used for judging the low-temperature reaction mechanism below 1100 K and auto ignition.

Table 3. Configuration of simulation

Combustion	TI Combustion Model
Turbulence	RNG k- ϵ Model [24]
Drop Breakup	KH-RT Model [25]
Ignition	Shell Model [21]
Burnt Gas Dissociation	Post Chemistry Model [11]

Glow Plug Adaptation

In this study, rectangular high temperature wall was added to the model to imitate glow plug. Fig. 5 shows the computational grid of surrounding glow plug. According to the Oomori's measurement results [26], in order to predict temperature fraction including thickness of thermal boundary layer, grid size should be 0.5 mm or less. This time considering this requirement, the computation grid sizes were restricted from 0.13 to 0.3 mm.

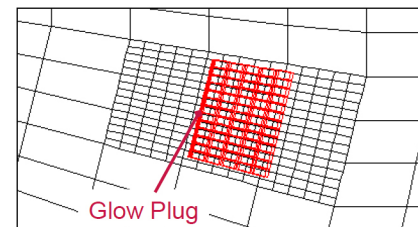


Fig. 5. Computation grid near glow plug

Fig. 6 depicts examples of the temperature fraction near glow plug simulated in this study. As showed in Fig. 6, temperature fraction including thermal boundary layer near