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EFFECT OF EGR RATIO TO CONTROL PARTICULATE MATTER EMISSION FROM DIESEL ENGINE COMBUSTION BY USING HIGH –SPEED COMBUSTION MODEL

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ABSTRACT

Diesel engine attracts market attention due to its high thermal efficiency among other IC engine. To design diesel engines adapted to future exhaust gas regulations, it is important to develop a driving mode simulator to estimate vehicle performance and exhaust emissions. For the driving mode simulator with simple and high-speed calculation characteristics, the authors use a diesel combustion model based on the Hiroyasu model. In this study, we examine the effect of EGR on exhaust gas emission, like: Ethylene, CO, NOx, etc. For this we are varying EGR ratio during operation cycle. We use Tsurushima reduced kinetic model together with characteristic mixing time scale. Direct introduction of the reaction model to the basic combustion model did not give good simulation result for experimental data. However by introducing mixing time scale to take into account local heterogeneity of the mixture, the simulation accuracy was significantly improved. The result shows possibility of simulating the rates of heat release and emissions including ethylene and CO for the different EGR ratio.

Keywords: EGR, Combustion Model, Ethylene, CO, Nox ,Diesel Engine

1. INTRODUCTION

To design diesel engines to comply with future regulations of exhaust emissions, it is necessary to optimize combinations of combustion chamber configurations, common rail injection patterns, EGR, and after-treatment technology such as urea SCR systems and DPF. It is also necessary to control the optimum operating mode of the engine system during transient operation. However, considerable time and cost are necessary to develop optimization and control logic for the total engine systems by experiments with real engines. One of the development goals would be establishing a total engine simulation system which can predict the emission characteristics and engine performance including the after-treatment with real time processing speed [1]. For such a simulation system, a combustion model is required to possess both processing speed and prediction accuracy. This study develops a combustion model for use in high speed total engine simulation. The model is based on the combustion analysis program HTB-DECS with a phenomenological model developed by Hiroyasu et al.[2-4] due to its high speed computation characteristics. The paper improves the sub-models of the program such as air entrainment to the spray and ignition delays. The present model was also extended to deal with premixed compression ignition combustion, and this paper provides details of the combustion model and comparisons between experimental and calculated results of a single cylinder engine.

2. THE OUTLINE OF THE BASIC MODEL

The basic model has the spray structure shown in Fig. 1 [2-4]. The fuel spray is divided into the model is comprised of elements containing a fixed amount of fuel, and air is considered to be entrained into the elements. It does not consider combustion chamber configuration, and this effect is taken into account by adjusting some of the constants in the model. Migration length and speed of the spray elements are determined from
equations of the penetration length of the free spray established experimentally. The amount of air entrainment into an element is set so that the momentum of the injection is conserved. The air entrainment amount is corrected when collisions with the combustion chamber wall plane and ignition were generated. The spray mixed with air starts to evaporate and ignition takes place. Then, the combustion progresses across the spray elements with the rate limited by either the air entrainment or the evaporation of the fuel. It integrates the heat release of each element, and calculates heat release rates and pressure changes in the combustion chamber.

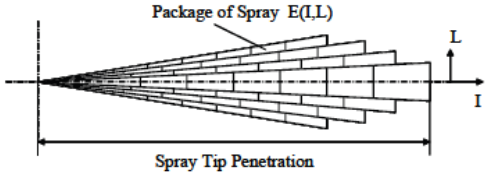

Fig 1. Schematic outline of the spray distribution

The composition of the combustion gas is calculated on the basis of the temperature history and fuel vapor quantity, combustion gas amount, and air quantity of each element by equilibrium calculations. NOx emissions are calculated by the Extended Zeldvich mechanism, and the soot emission model developed by Hiroyasu et al. is included. With the model, it was possible to calculate engine performance and emission characteristics with short times of computation by inputting engine dimensions, operating conditions, and other parameters. The model includes several empirical constants, and air entrainment coefficients, for example, must be adjusted in relation to the operating conditions and engine configuration.

The present paper investigates the effect of high speed on engine performance and the model comparing it with experimental results, and applies necessary modifications to minimize the adjustment procedure of the model constants for different operating conditions.

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3.1 Introduction of the Schreiber Model

Table 1 shows the model of Schreiber et al. The *F* represents a fuel, n-dodecane was the fuel in their model. Reactions 1 and 2 are high temperature oxidation reactions and reactions 3 to 5 are low temperature oxidation reactions. The end product is expressed by *P*, and *X, Y, and I* are chemical species which represent intermediate products forming in the process of the oxidation. The *X* is compounds which form by thermal decomposition of the fuel, *Y* is intermediate product such as *OH* radicals which appear in the reaction process, and *I* is also an intermediate product mainly the peroxides of the fuel. The reaction rate of each reaction is as follows:

Table 1: Five-step reduced reaction scheme

Reaction	A Mol.m3.Sec	Modified A Mol.m3.Se	H_{298} (KJ/mol)	E /R [K]
1	5.0E08	1.5E09	709.9	18050
2	7.0E06	7.0E06	-6709.9	7200
3+	3.5E09	3.0E08	-53.9	19500
3-	6.0E27	1.5E27	53.9	37500
4	6.0E07	6.0E07	-60.0	5000
5	1.0E09	1.0E13	-5943.1	16500

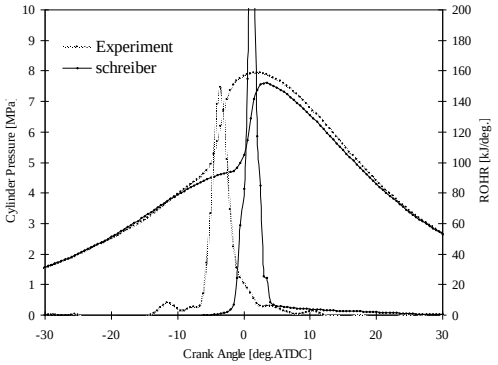

Fig 2. Comparison of Schreiber and experimental result: 1350 rpm and injection timing at –27.CA

$$R_1 = K_1[F](p/p_0)^{0.5}$$
$$R_2 = K_2[X][O_2][M]$$
$$R_{3+} = K_{3+}[F][O_2][M](p/p_0)^{-2.2}C_{3+}$$
$$R_{3-} = K_3[I](p/p_0)^{-3.5}$$
$$R_4 = K_4[I]C_4$$
$$R_5 = K_5[O_2][Y]$$

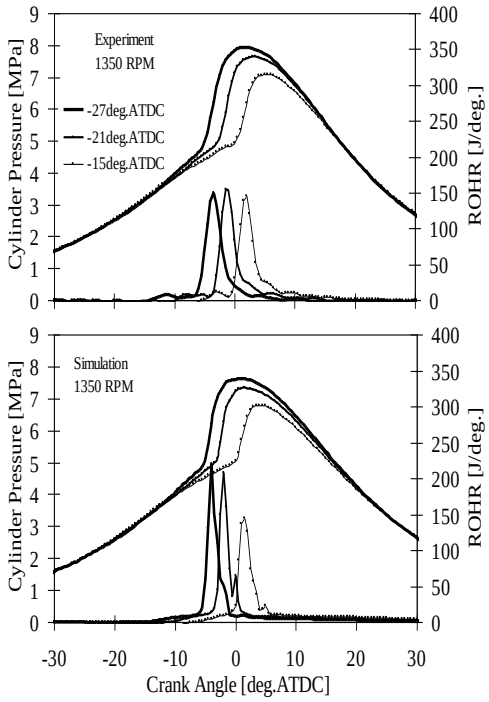

Fig.3 R.O.H.R and cylinder pressure for simulations and experimental results for various injection timings. (Reaction optimized) 1350 rpm

The *k* parameters in each reaction equation are the rate constants of the reaction, and is given by *k=A exp(-E/RT)*; [*M*] is the total concentration of all species, and is obtained from the state equation as *[M]=p/(RT)*; the value of *P0* is 1MPa; and *C3+* and *C4* are correction factors for the cetane value. To verify the prediction accuracy with the model of Schreiber et al., the

experimental results by the single cylinder engine B were compared with the calculated results.

The engine operation was at 1350rpm, 50Mpa injection pressure, injection quantity 20mm3/st, EGR rate 55%, and the fuel injection timing was varied. With the original set of values proposed by Schreiber, shown in Table 2, there were large differences in the heat release timings when compared with the experimental result.

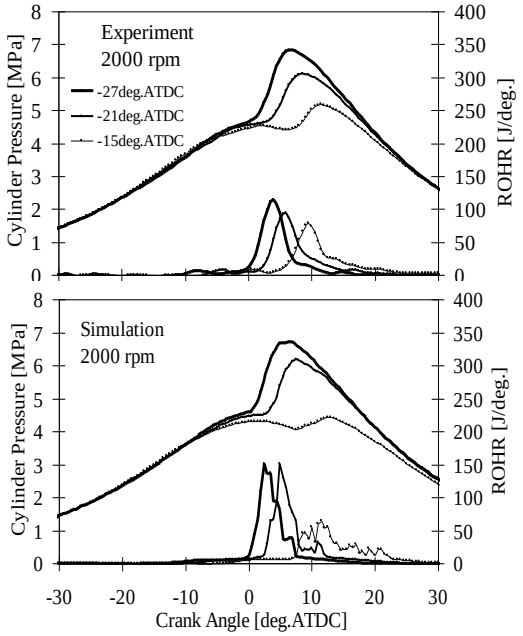

Fig 4. R.O.H.R and cylinder pressure for simulations and experimental results for various injection timings. (Reaction optimized) 2000 rpm

Thus, the *A* factors in Table 2 were corrected to fit the experimental results of the -27deg. ATDC fuel injection timing (Figure.2) The corrected set of frequency factors *A* is shown in Table 2 as a modified value of A. Figure 3 shows the heat release rates and cylinder pressures with the fuel injection timings -27,-21,-15deg.ATDC, and the top is the experimental results, and the bottom the calculation result for 1350 rpm .In the same way figure. 4 shows for rpm 2000.In the experimental result the heat release timing becomes earlier when advancing the fuel injection timing from -18 to -24deg. ATDC, and the peak of the heat release rate rises slightly. At the injection timing of –15deg. ATDC for 2000 rpm , the heat release becomes slower than the 1350 rpm. Before the main combustion showing the low temperature oxidation reaction. The simulation indicates the appearance of the low temperature combustion and the general trend in the positions of the peak heat release, however there are still large differences between experimental and the calculated results. To improve the degree of simulation the following mixing time model was introduced, considering local heterogeneities of the pre-mixed fuel.

3.2 The Characteristic Mixing-Time Model

In the previous section, the spray elements were assumed homogeneous, and the chemical reactions take place with this assumption. With this homogeneous model, the fit of the heat release rates, even when the reaction constants were adjusted, was poor. Some reports have proposed the existence of local heterogeneities in the fuel concentration, and this may be reason why the combustion is slower than the calculations [9, 10]. Thus, heterogeneity was introduced in the computational elements, and the chemical reaction rates were modified with this mixing time scale as shown in Eq. (10), similar to the manner in Ref.[11]. The *Rchem* is the reaction rate for a homogeneous element, and *Rmix* is the reaction rate corrected by a characteristic time scale of mixing, τ_{mix}

$$R_{mix} = \frac{dt}{dt + \tau_{mix}} R_{chem} \qquad \text{Eq.(1)}$$

The *dt* is the computational time step, and when the time step is large compared with the mixing time scale, the element can be regarded as having a uniform mixture. The characteristic time of mixing τ_{mix} is a function of the heterogeneity scale *l* and turbulence intensity *u'*, and equation (11) was assumed. The heterogeneity scale *l* was also assumed to be in the order of the turbulence scale. As the air entrainment into the computational element can be correlated with the local turbulence scale and intensity as in Eq. (12)

$$(\tau_{mix})_{spr} = A_{spr} \frac{\dot{m}_{spr}}{u'^3_{spr}} \qquad \text{Eq.(2)}$$

$$(\tau_{mix})_{cyl} = A_{cyl} \frac{\dot{m}_{cyl}}{u'^3_{cyl}} \qquad \text{Eq.(3)}$$

$$\frac{1}{\tau_{mix}} = \frac{1}{(\tau_{mix})_{spr}} + \frac{1}{(\tau_{mix})_{cyl}} \qquad \text{Eq.(4)}$$

The adjusting constants, *Aspr* and *Acyl* in the above equations, were calibrated to fit the specific results of the experiments. The turbulence velocity *u'spr* in Eqs. (10) was set to 0.3*um* for the spray, based on the theory of a free jet as shown in Ref. [12] , where *um* is the velocity of the spray element. The *u'cyl* was determined from the average value in the cylinder calculated by KIVA simulation without combustion.

3.3 The Calculations using the Characteristic Mixing Time Model

The above characteristic mixing time model was applied to the model of Schreiber et al., and the experimental results of the pre-mixed compression ignition combustion was again compared with the calculated results, for engine operating conditions similar to those in Fig. 4. In the calculations, the frequency factor *A* in the reaction rate equation and characteristic time mixing model constant *Aspr*

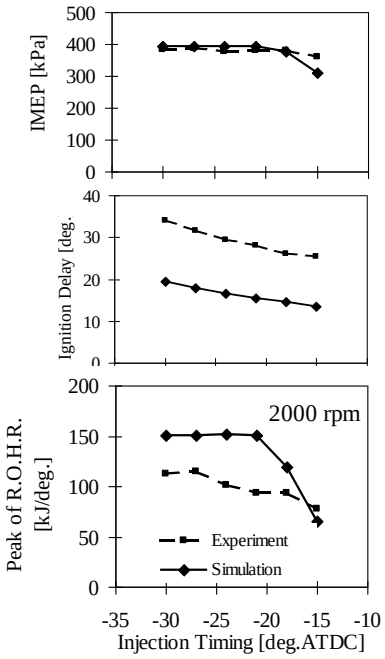

Fig.5 NOx, Unburned fuel and Soot for simulations and experimental results for various injection timings. (Reaction optimized) 2000 rpm

Acyl were set to fit the experimental results at the fuel injection timing -27deg. ATDC. Figure 5 shows the calculated results of cylinder pressure ,ignition delay and Peak of R.O.H.R after introducing τ_{mix}.

As a result it was possible to simulate the pre- mixed compression ignition combustion accurately by introducing the characteristic mixing time. Formation of soot is well simulate with experimental result and percentage of unburned hydrocarbon have good relationship up to –17deg.ATDC.The computational time including the whole set of modifications was about 10 seconds with a personal computer with a Pentium 4 and 2.8GHz.

4. CONCLUSIONS

The phenomenological model devised by Hiroyasu et al. was further developed into a combustion model for use in a total engine simulation system with short computational times. The developed model was evaluated for two different speed of engines to determine the characteristics and prediction accuracy of the model. The simplified and chemical reaction model of Schreiber et al. was included to simulate the pre-mixed compression ignition combustion, and was compared with the experimental results. The results may be summarized as follows:

(1) The improved model was accurately able to simulate combustion in the cylinder by minimal tuning of the model constants for different engines, and it was confirmed to be accurate in predicting performance characteristics and NOx quantities.

(2) To simulate pre-mixed compression ignition combustion, the simplified chemical reaction model of Schreiber et al. and a mixing time characteristics model were included, and it became possible to simulate two step combustion with good accuracy. With this model the parameters to be adjusted are the frequency factors *A* in the reaction rate equations and the characteristic mixing time constants *Aspr and Acyl*, which are constant for a engine.

(3) The model is able to forecast about Ignition delay, Soot and percentage of unburned fuel, which is important for total engine simulation system.

(4) EGR ratio effect greatly to engine performance.

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