

# Study on Prediction of Four Stroke Diesel Engine Performance — Cycle Simulation Using Heat Release Model —

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## ABSTRACT

It is well known to diesel engineers that the heat release pattern is one of the most important factors affecting engine performance. Thorough research in heat release pattern has materially helped the progress in high-speed diesel engine development. This paper is based on the research conducted at KAIST and Daewoo Heavy Industry last year. The purpose of this paper is to determine the heat release pattern in combustion chamber of MAN M type, the famous low-noise engine.

Thermodynamic cycle simulation was performed using Whitehouse-Way's heat release pattern with modified coefficients and Annand's heat transfer model. Instantaneous temperature and pressure of gas in cylinder could be determined by the numerical solution of simultaneous equation of mass conservation, equation of energy conservation, and state equation of ideal gas. Calculated results were compared with measured values in some details emphasizing upon the factors affecting rate of heat release.

The agreement was fairly good and revealed why M type should have lower burning velocity at the early part of combustion inspite of high injection rate. Additional results by parametric studies were given in relation to fuel injection conditions for further application to engine development.

## INTRODUCTION

These days the problems of energy and environment pollution become serious day by day and the diesel engine, which was proved to be superior to gasoline engine with respect to fuel consumption and exhaust gas, can be adopted to the passenger car by increasing its application domain. Therefore, broad parametric-study on combustion in engine should be done for design and improvement of diesel engine, but since this procedure has some limitations on experiment and measurement which require a lot of time and money, the simulation of a working cycle becomes necessary.

Since the study by Lyn<sup>(1)</sup> in 1960, a lot of studies in many country have been accomplished and the basic idea of these cycle-simulation is to predict the pattern of heat release with time. In the diesel engine, the fuel which is injected in combustion chamber evaporates and burns in a very short time. And because its mode changes greatly in accordance with the shape of combustion chamber

and the forming method of air-fuel mixture, necessary information has been obtained from the pressure or heat release curve of experiments.

In 1956, Wiebe<sup>(2)</sup> succeeded in determining combustion function by using semi-empirical formula in order to simplify the calculation. But as this formula is dependent on time only, influence of fuel injection can not be considered in case of the diesel engine. In 1962 Lyn<sup>(1)</sup>, introducing the concept of ignition delay to obtain the relation of the prepared fuel for combustion among the injected fuel, tried to predict the rate of heat release under the several driving conditions. But the relation between chemical reaction velocity and fuel preparation rate was not clear and the effect of air-fuel ratio on combustion process was not considered. Therefore it is necessary to clarify the relations between various mechanisms in the whole cycle in order to rationalize the cycle simulation of engine performance as a prediction method. Specially it is important to establish the interrelations of factors effecting fuel preparation rate and reaction rate by considering the formation of mixture gas in combustion chamber. In 1971, the study of Whitehouse and Way<sup>(2)</sup> displayed that these problem could be partially solved by use of two experimentally determined formulas of fuel preparation rate and reaction rate including partial pressure effect of oxygen.

In this paper Whitehouse-Way's model was applied to calculate the variations of the pressure and temperature during one cycle in the combustion chamber of MAN-M engine, the famous low-noise engine which the German developed in 1955, and these results were compared with the experimental results under the same actual condition. Besides some parametric-study were included in this study. Also, semi-empirical coefficients for M-type combustion chamber whose method of mixture formation was unique were obtained and by comparing and examining the coefficients which were expressed by Whitehouse and Way based on single droplet model, indirect analysis about evaporation speed of fuel in M-type combustion chamber was tried.

## M-TYPE COMBUSTION

M-type combustion chamber shown in Fig. 1 has a spherical bowl in the top part of the piston and fuel is injected in the tangential direction to the chamber wall. According to Meurer<sup>(4)</sup>, about 95 percent of the injected fuel forms a thin layer on

the wall and moves around and evaporates on the wall by the strong swirl motion of the air. Once combustion starts, combustion gas near the wall moves to the center of the chamber due to the centrifugal force resulting from the strong swirl and that empty space is filled up by the air, enhancing combustion process. Figure 1(c) shows the trajectories of the fuel particle, air and combustion gas when the swirl motion is assumed to be potential flow, where  $\rho'/\rho \approx 400$  for fuel particle and  $\rho'/\rho \approx 300$  for the combustion gas. Therefore the evaporating mechanism of fuel in the M-type combustion chamber happens mostly on the wall and is quite different from those, by which the fuel is injected and evaporates in the air.

**THERMODYNAMIC CYCLE CALCULATION**

Modelling and Assumptions

In this paper, in order to investigate the variation of thermodynamic properties in the M-type combustion chamber, calculation was performed with the intake and exhaust ports closed under the following assumptions (refer to Fig. 2)

- 1) Working fluid in the combustion chamber is ideal gas.
- 2) Fuel is composed of only carbon and hydrogen and its compositions are expressed by the weight fractions.
- 3) Combustion process is assumed to be stoichiometric and thermal dissociation and NOx formation are ignored.
- 4) Properties in the combustion chamber are homogeneous (single zone model)

Besides, in order to predict the trapped air during the intake process, the procedure of air flow into the cylinder through the intake pipe from the intake manifold is assumed to be adiabatic.

Calculation of Intake Air

Under the assumptions of section 3.1, we can derive the following two differential equations for

pressure and temperature from the mass and energy conservation laws, apart from combustion process.

$$\frac{dP}{d\phi} = \frac{\gamma}{V} \left\{ RT^* \frac{dM_1}{d\phi} - P \frac{dV}{d\phi} \right\} \dots\dots\dots(1)$$

$$\frac{dT}{d\phi} = \frac{RT}{PV} (\gamma T^* - T) \frac{dM_1}{d\phi} - (\gamma - 1) \frac{T}{V} \frac{dV}{d\phi} \dots\dots\dots(2)$$

where  $T^*$  denotes the upstream temperature which is assumed to be  $T_1$  and  $T$  for  $dM_1/d\phi > 0$  and  $dM_1/d\phi < 0$  (reversing flow) respectively.

Assuming the adiabatic process,  $dM_1/d\phi$  can be expressed as follows,

$$\frac{dM_1}{d\phi} = \frac{A_1 \mu_1}{6N} \psi_1(P_1, P) P_0 \sqrt{\frac{2g}{RT_0}} \dots\dots\dots(3)$$

where  $\psi_1(P_1, P) = \sqrt{\frac{\gamma}{\gamma-1} \left\{ \left(\frac{P}{P_1}\right)^{\frac{2}{\gamma}} - \left(\frac{P}{P_1}\right)^{\frac{\gamma+1}{\gamma}} \right\}}$ ;

for  $\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}} < \frac{P}{P_1} \leq 1$

$$\psi_1(P_1, P) = \left(\frac{2}{\gamma+1}\right)^{\frac{1}{\gamma-1}} \sqrt{\frac{\gamma}{\gamma+1}} ;$$

for  $\frac{P}{P_1} \leq \left(\frac{2}{\gamma+1}\right)^{\frac{\gamma}{\gamma-1}}$

and when the reversing flow occurs (i.e.,  $P/P_0 > 1$ )

$$\frac{dM_1}{d\phi} = - \frac{A_1 \mu_1}{6N} \psi_1(P, P_1) P \sqrt{\frac{2g}{RT}} \dots\dots\dots(4)$$

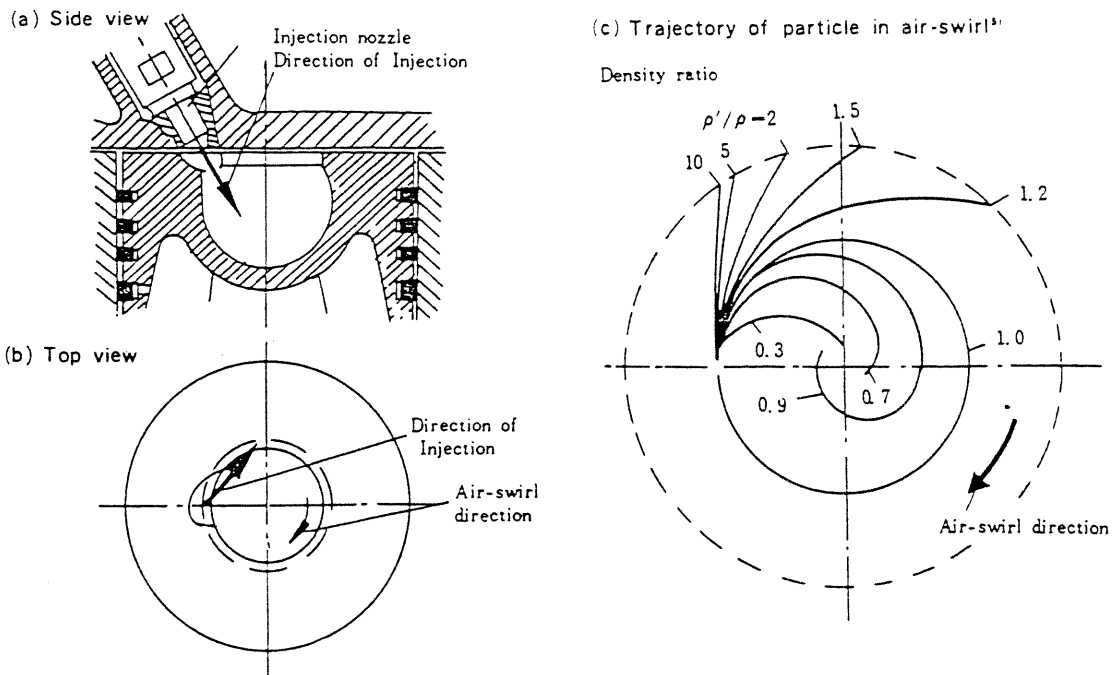


Fig. 1. MAN-M engine combustion chamber

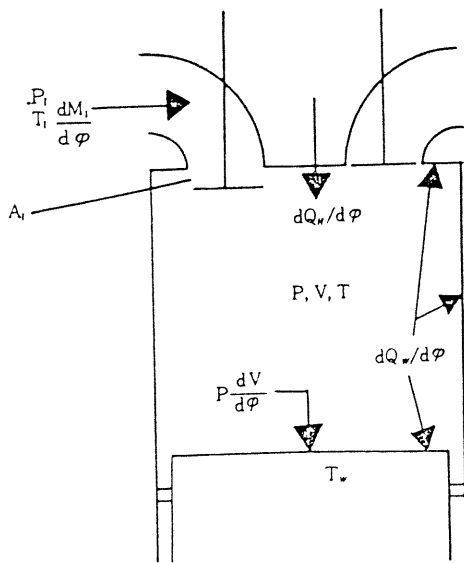


Fig. 2. Notation for the computation of the energy and mass balances for intake and combustion processes.

The trapped air in the cylinder can be calculated by obtaining  $dM_1/d\phi$  from the instantaneous pressure and temperature at each crank angle and integrating it up to the point of intake valve closing.

Model of Heat Release Rate

As mentioned in the introductory section, the prediction of heat release rate is based on the model proposed by Whitehouse and Way<sup>(3)</sup>, which is essentially a single zone model. This model is considered as widely acceptable after applied to various types of diesel engines and is expressed as

$$SP(\phi) = K_p SM_1(\phi)^{(1-x)} SM_u(\phi)^x P_{O_2}^m \dots\dots\dots(5)$$

$$SR(\phi) = K_R \frac{P_{O_2}}{N/T} \exp\left(-\frac{act}{T}\right) x \int \{SP(\phi) - SR(\phi)\} d\phi \dots\dots\dots(6)$$

where the coefficients  $x$ ,  $m$ , and  $act$ , being independent of the combustion type, retains its original values suggested by Whitehouse and Way and the parameters  $K_p$  and  $K_R$ , being greatly dependent on combustion type, are chosen by proper values from the analysis of pressure and heat release rate obtained from the experiment because there are no available data for the M-type combustion (refer to Table 1).

Calculation of Heat Transfer Rate

Heat loss from the actual engine cylinder is about 15% of total heat of the fuel and cannot be ignored. Assuming constant wall temperature from the fact in Meurer's report<sup>(4)</sup> that the cylinder wall in the M-type combustion chamber keeps its temperature about 330°C for the rapid vaporization of the fuel, heat transfer rate was calculated using the Annand's empirical formula<sup>(6)</sup> which takes into account the forced convection and the gray-

Table 1. Coefficients for Heat Release and Heat Transfer Modeling

x	1	
m	0.01	Ref. (3)
act	$1.5 \times 10^4$ °K	
$K_p$	$0.0052 \text{ bars}^{-m}$	Present
$K_R$	$3.0 \times 10^9 \sqrt{k}/\text{bars}$	Study
a	0.2	
b	0.7	Ref. (6)
c	$3.28 \times 10^{-8} \text{ J/sm}^2 \text{ K}^4$	

body radiation. This is

$$\frac{dQ_w}{d\phi} = A_s \left\{ a \frac{k}{D} (Re_D)^b (T - T_w) + c (T^4 - T_w^4) \right\} \text{ J/deg.} \dots\dots\dots(7)$$

where the experimental coefficients  $a$ ,  $b$  and  $c$  use the same values as obtained in the Dorman engine (refer to Table 1). The thermal conductivity of air,  $k$ , is corrected for the variation of the temperature by the following relation proposed by Davies and Fisher<sup>(7)</sup>.

$$k = \left\{ 2.56 + \frac{7.3 (T - 327)}{1000} \right\} \frac{1}{100} \text{ W/}^\circ\text{K} \dots\dots\dots(8)$$

And Reynolds number  $Re_D$  is defined as below, using the cylinder diameter as the characteristic length.

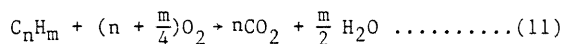
$$Re_D = \frac{\rho V_p D}{\mu} \dots\dots\dots(9)$$

where  $V_p$  is the mean piston velocity and  $\mu$ , the viscosity of air, is corrected by the Surtherland's formula

$$\mu = \mu_s \frac{\left(\frac{T}{273}\right)^{\frac{2}{3}}}{T + 117} \dots\dots\dots(10)$$

Thermodynamic Relations

Referring to the assumptions of section 3.1, the simple combustion process is considered. For one mole of fuel,



For a given number of moles of fuel, the number of moles of each component at any instant is determined by the model of heat release rate. Representing the specific enthalpy<sup>(9)</sup> as a polynomial of temperature for each component, the total internal energy  $E_i$  at time  $i$  is expressed as follows.

$$E_i = \left\{ \sum_k M_{k,i} R_{mol} \left( \sum_j u_{k,j} T_i^j - T_i \right) \right\} = \left\{ \sum_k M_{k,i} R_{mol} \left( \sum_j u_{k,j} T_s^j - T_s \right) \right\} \dots\dots\dots(12)$$

The properties in the cylinder can be obtained by solving the first law of the thermodynamics and the equation of state simultaneously with the control volume of the cylinder. At any instant  $i$ ,

$$\text{Equation of state : } P_i V_i = M_i R_{mol} T_i \dots\dots\dots(13)$$

The 1'st law of thermodynamics :

$$dQ)_i - dW)_i = dE)_i \dots\dots\dots(14)$$

where

$$dQ)_i = dQ_w)_i + dQ_{H'})_i$$

$$\cong (Q_w)_{i+1} - Q_w)_i + \alpha Q_T$$

$$\alpha = \begin{cases} 0 & \text{: before the ignition} \\ SR(\phi_i) & \text{: for } SP(\phi_i) > SR(\phi_i) \\ SP(\phi_i) & \text{: for } SP(\phi_i) < SR(\phi_i) \end{cases}$$

$$dW)_i \cong \left( \frac{P_i + P_{i+1}}{2} \right) (V_{i+1} - V_i)$$

The non-linear algebraic equation for  $T_{i+1}$  (or  $P_{i+1}$ ), obtained by combining the above equations (13) and (14) with the known values  $P_i, T_i$ , can be solved by an iterative method. In this paper Newton-Rapson method was applied and the convergence criteria was chosen to be  $10^{-2}$ . The numerical calculation was carried out with an increment of  $1^\circ$  crank angle up to the exhaust valve opening and then power, mean effective pressure and thermal efficiency can be obtained by the following equations.

$$PS = \oint PdV \dots\dots\dots(15)$$

$$P_{me} = PS / (V_{GVE} - V_{IVC}) \dots\dots\dots(16)$$

$$\eta_{th} = - \frac{PS}{Q_T} \times 100 \dots\dots\dots(17)$$

Figure 3 shows the flow chart of the cycle simulation and Table 2 displays the specification of the M-type engine.

Table 2. Details of Engine Used in Experiment and Model Calculation.

Engine Type	T/C, Direct Injection, MAN-M
Bore	0.123m
Stroke	0.150m
Length of Connecting Rod	0.275m
Compression Ratio	16.5
Operating Cycle	4 stroke
Fuel Used	Diesel
Number of Cylinders	6

## RESULTS AND DISCUSSION

### Comparison With Experimental Results

In order to examine the validity of the calculation, results are compared with the pressure curve obtained from the measurement by Daewoo Heavy Industry Co. Comparisons at 1400 and 1800 RPM as in Fig. 4 show that the time and magnitude of the highest explosion pressure are agreed quite well with each other, and that for the rest of the period after that point prediction is fairly good with

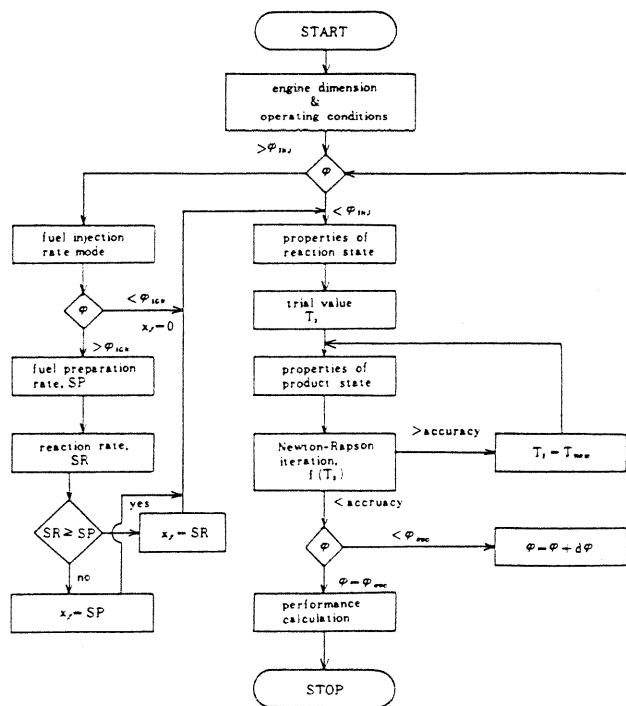


Fig.3. Diagram for Entire Calculation Procedure.

only about  $5 \text{ kg/cm}^2$  difference. Calculation results showing a little higher values during combustion process may be due to the assumption of the complete combustion and no thermal dissociation. Especially these phenomena are predominant during the diffusive combustion process. Proper values of  $K_p$  and  $K_R$  obtained from the experimental results are as follow.

$$K_p = 0.0052 \text{ bars}^{-m} \dots\dots\dots(18)$$

$$K_R = 3 \times 10^9 \sqrt{K} / (\text{bar} \cdot \text{sec}) \dots\dots\dots(19)$$

### Investigation of Combustion Process

Figure 5 shows the predictions for the variations of non-dimensional average fuel-film thickness  $\delta^*$ , rate of burning  $\alpha$  and pressure  $P$  when the fuel injection rate is assumed to be constant. Here assuming that the cylinder area occupied by

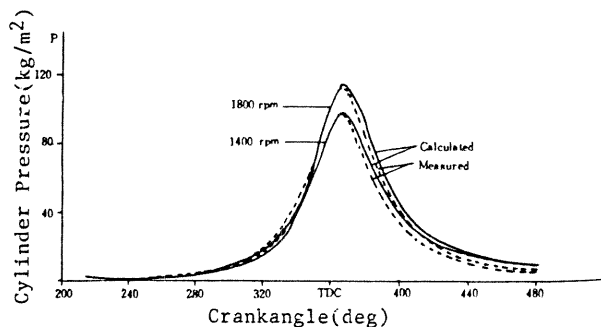


Fig.4. Comparison of calculated and measured pressure at 1400 & 1800 rpm.

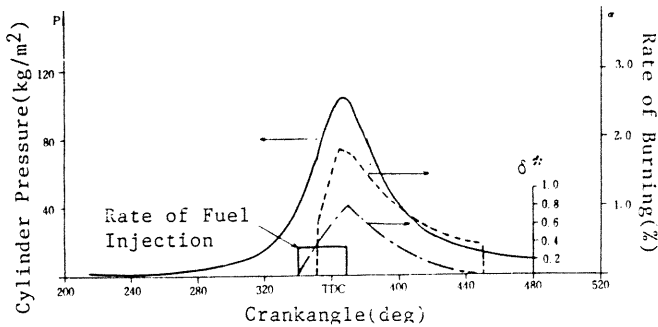


Fig. 5. Influence of fuel injection on average fuel film-thickness( $\delta^*$ ), rate of burning( $\alpha$ ), and pressure curves at 1400 rpm.

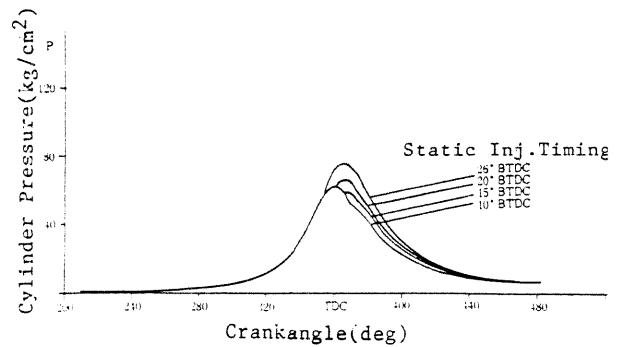


Fig. 6. Effect of injection timing change on the pressure curves at 1000 rpm.

the fuel is constant,  $\delta^*$  is defined by

$$\delta^* = \frac{\bar{\delta}}{\bar{\delta}_T} \dots \dots \dots (20)$$

During fuel injection  $\delta^*$  increases gradually until the end of the fuel injection, where it has a peak value and then decreases monotonically during the diffusion combustion process. In results it makes the whole distribution of  $\delta^*$  triangular. Comparing this with the experimental data by Urlaub<sup>(9)</sup> shows difference just after the injection. It seems because the area occupied by the injected fuel does not reach a critical value, and the film thickness at the measuring point appears larger than the average  $\delta^*$ . That is, due to non-uniform distribution of the fuel at the beginning of the combustion the actual amount of vaporized fuel is small in spite of high temperature of the cylinder wall. Calculated combustion-rate curve shows that, compared with an ordinary engine injecting the fuel in the air stream, the premixed combustion rate is relatively slow at the beginning of combustion and this makes the low noise operation possible. Entire combustion-rate curve becomes nearly triangular and it approaches the ideal triangle indicated by the experimental result of Meuer<sup>(1)</sup>. In case of Dorman engine adopting the evaporation in the air and vaporization on the wall from the results by Whitehouse and Way<sup>(3)</sup>,  $K_p=0.0115$  and  $K_R=1.2 \times 10^{10}$  at the beginning of combustion, which are larger than those of this study by double and four times respectively. Therefore the vaporization of fuel in the air is more useful than that on the wall, but it is thought that this makes a shock due to combustion large and the combustion incomplete when the air supply is not sufficient.

Influence of Fuel Injection Timing on Engine Performance.

As an application of this study the influence of the injection timing will be discussed. Figure 6 shows the comparison of the calculated pressure curves with the injection timings of 26°, 20°, 15° and 10° BTDC and the injection quantity fixed at 1000 RPM. It can be seen that the late injection makes the maximum explosion pressure and thermal efficiency decrease. Figure 7 represents the relation between the calculated maximum explosion pressure and indicated thermal efficiency for four kinds of engine speed and eight kinds of injection timing. Fuel injection quantity is almost same at

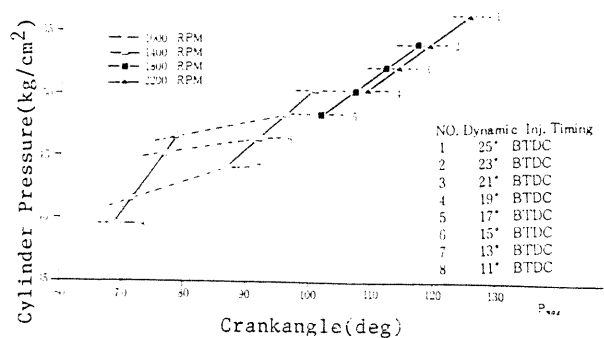


Fig. 7. Indicated maximum cylinder pressure vs. indicated thermal efficiency with varying fuel injection timings, rpm, and injected fuel amounts.

1400, 1800 and 2200 RPM but 20% smaller at 1000 RPM. At the same speed, earlier injection makes  $P_{max}$  and  $\eta_{th}$  increases as shown in Fig. 7 and with the same injection timing and injection rate, higher speed (i. e., 1400, 1800 and 2000 RPM) makes  $P_{max}$  increase but keeps  $\eta_{th}$  nearly same. This fact sets up the basis for giving the limitation on the determination of injection timing in order to obtain the required thermal efficiency per unit RPM when the maximum allowable pressure of the combustion chamber is fixed. In other words it is desirable to find the optimal condition under a given condition because the ideal engine requires low  $P_{max}$  and high  $\eta_{th}$  but the actual engine can not satisfy two requirements at the same time.

CONCLUSIONS

For an engine with the M-type combustion chamber a cycle simulation was conducted using Whitehouse and Way's heat release model which includes the effects of injection characteristics, evaporation speed and reaction rate and the thermodynamic properties at each instant were calculated numerically.

The results are the followings.

- 1) The values of  $K_p$  and  $K_R$  obtained in this study can be applied to the M-type combustion engine and are considered to be reasonable, compared with those for Dorman engine.
- 2) Compared with experimental results, the predic-

ted pressure showed the difference of about  $5\text{Kg}/\text{cm}^2$  because of the assumptions of complete combustion and no thermal dissociation. If a multi-zone modelling which considers the mechanisms of the combustion and the thermal dissociation in time and space is developed, the difference will decrease and the prediction of the exhaust gas components be possible.

- 3) Considering that in the early stage of injection the average fuel film measured by Urlaub was thicker than the calculated one, the fuel distribution on the wall of M-type combustion chamber would be non-uniform in the early stage of combustion and therefore the spacial distribution of fuel in the early stage of injection would influence deeply the initial combustion rate and the local air-fuel ratio.
- 4) The rate of burning curves using the obtained  $K_p$  and  $K_R$  in this study shows the triangular shape similar to the other experimental results.
- 5) Changing the injection timing with a constant injection rate and constant engine speed shows the earlier the injection occurs the higher maximum cylinder pressure and thermal efficiency can be obtained. As the engine RPM increases with the same injection timing, the maximum cylinder pressure increases but the thermal efficiency has nearly same value.

#### NOMENCLATURE

A	: area, $\text{m}^2$
a	: coefficient of Annand's equation
act	: coefficient of reaction rate equation, $^{\circ}\text{K}$
b	: coefficient of Annand's equation
C	: carbon
c	: coefficient of Annand's equation
D	: cylinder bore, m
E	: internal energy, J
g	: acceleration of gravity, $\text{m}/\text{sec}^2$
H	: hydrogen
k	: thermal conductivity, $\text{W}/^{\circ}\text{K}$
$K_p$	: coefficient of preparation rate equation, $\text{bars}^{-m}$
$K_R$	: coefficient of reaction rate equation, $\sqrt{\text{OK}}/\text{bars}$
M	: number of mols, mol
m	: number of hydrogen atoms in equivalent hydrocarbon fuel
m	: coefficient of preparation and reaction rate equations
$M_1$	: mass of airflow through intake valve
N	: nitrogen
N	: engine rpm, rev/min
n	: number of carbon atoms in equivalent hydrocarbon fuel
O	: oxygen
P	: pressure in cylinder, $\text{N}/\text{m}^2$
$P_{me}$	: mean effective pressure, $\text{N}/\text{m}^2$
$P_o$	: pressure in intake manifold, $\text{N}/\text{m}^2$
$P_{O_2}$	: partial pressure of Oxyzen, $\text{N}/\text{m}^2$
PS	: power output, PS
Q	: heat value, J
R	: characteristic gas constant, $\text{J}/\text{kg}^{\circ}\text{K}$
$R_{mol}$	: universal gas constant, $\text{J}/\text{kg}\text{-mol}^{\circ}\text{K}$
$Re_D$	: Reynolds number based on cylinder bore
S	: stroke, m
$SM_i(\phi)$	: rate of fuel injected up to crankangle $\phi$
$SM_u(\phi)$	: rate of fuel unprepared at crankangle $\phi$
SP( $\phi$ )	: preparation rate of fuel at crankangle $\phi$

SR( $\phi$ )	: reaction rate of fuel at crankangle $\phi$
T	: temperature in cylinder, $^{\circ}\text{K}$
$T_o$	: temperature in intake manifold, $^{\circ}\text{K}$
u	: polynomial coefficient of enthalpy
V	: volume in cylinder, $\text{m}^3$
$V_p$	: average piston speed, $\text{m}/\text{sec}$
x	: coefficient of preparation and reaction rate equation
$\alpha$	: burning rate
$\gamma$	: ratio of specific heat
$\delta$	: instantaneous average fuel film thickness at crankangle $\phi$ , mm
$\delta^*$	: nondimensional fuel film thickness ( $=\delta/\delta_T$ )
$\delta_T$	: total fuel-film thickness
$\eta_{th}$	: indicated thermal efficiency, %
$\mu$	: dynamic viscosity of air, $\text{kg}/\text{m}\cdot\text{s}$
$\mu_1$	: friction coefficient through intake valve
$\rho$	: density of air, $\text{kg}/\text{m}^3$
$\rho'$	: density of particle, $\text{kg}/\text{m}^3$
$\phi$	: crankangle, degree
$\psi_1$	: shape function determining mass flow rate through intake valve

#### Subscripts

H	: heat release
i	: number of time interval
j	: exponent of temperature
k	: component
S	: reference state
T	: total
W	: wall

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