# Combustion Simulations of Direct Injection Diesel Engine Having 'M' Type Combustion Chamber

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ABSTRACT

In this paper, a combustion simulation model for a 4-stroke, direct-injection diesel engine is described.

The combustion modeling was based on the assumption that the combustion process was taken to be only a heat addition process and the in-cylinder conditions are spatially uniform. As the heat release model, the well-known Wiebe's combustion function and the Whitehouse-Way's heat release model were studied and successfully employed to the combstion simulation.

A comparison has been made between the predicted results of the cylinder pressure and the rate of heat release and the measured data, and showed that the predictions agreed well with the measured data.

## INTRODUCTION

The thermodynamic cycle analysis and the performance prediction of diesel engines using the engine cycle simulation programs has become an established part of the diesel engine research and development. Particularly, the engine simulation programs based on the single-zone combustion model are widely used by most engine manufacturers because of simplicity and low operating cost in spite of their shortcomings in accuracy. These programs can readily utilized for the prediction of cylinder firing pressure for the stress analysis of main engine parts, investigation of engine design options and the optimization of aspects of design via parameter studies, etc.(1)*

In the present work, the combustion models based on the Wiebe's combustion function(2) and the Whitehouse-Way's heat release model(3)(4), which are usually used in the single-zone combustion model, have been made and the calculation results have been compared with the measured data, in order to obtain the appropriate engine simulation program for the tested engine.

* Numbers in parenthese designate References at the end of paper.

## EXPERIMENTATIONS

In the present study, the test engine as shown in Table 1. was used. This was a 4-stroke, watercooled, in-line 6-cylinder, direct injection and turbocharged diesel engine having 'M' type combustion chamber.

Table 1. Specification of tested engine

| ITEM | SPECIFICATIONS |
|---|---|
| ASPIRATION | TC |
| COMPRESSION RATIO | 17:1 |
| BORE x STROKE, mm | 121 x 150 |
| RATED OUTPUT, kW/rpm | 188 / 2200 |
| PEAK TORQUE, Nm/rpm | 892 / 1400 |

The cylinder pressure was measured by a piezo-quartz pressure transducer (AVL-8QP3000) and AVL Indicating System equipped with amplifier(AVL type 3059), control unit(AVL type 4004) and 4-channel oscilloscope (Tektronix type 5103N).

## 'M' COMBUSTION SYSTEM

The 'M' system is a kind of direct injection combustion system(5). However, the 'M' system does not distribute the fuel in the air initially, but sprays it with a single-hole nozzle onto the oil-cooled wall of the spherical combustion chamber, where it spreads to form a thin film as shown in Fig.1(6).

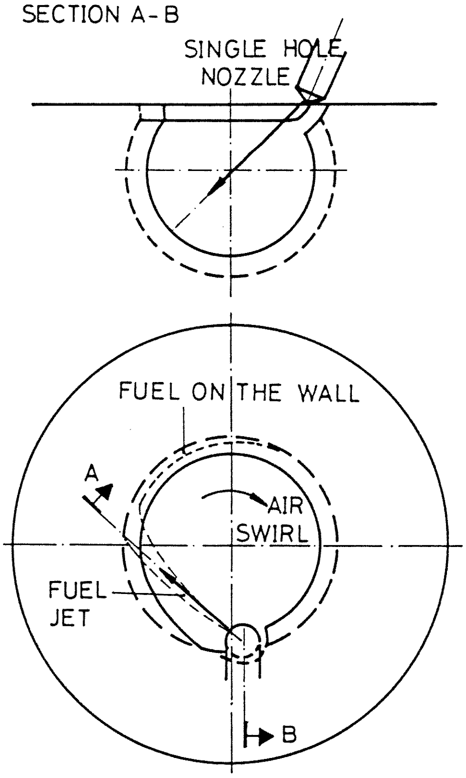



Fig. 1 'M' combustion system

The fuel is injected in the direction of a high-speed air swirl which is produced during the suction stroke of the engine by the helical inlet ports. The orientation of the fuel jet parallel with the air swirl reduces the relative velocity between the air and the fuel jet which, coupled to a short distance between nozzle and combustion chamber wall, helps to prevent the fuel jet from being broken up to any great extent. This means that during injection only a relatively small portion of the fuel is mixed with the air and prepared for ignition. Therefore, independent of the temperature and stress levels of the engine, there is always only a very small amount of air/fuel mixtures ready for the reactions that precede combustion. As a result, the initial rate of heat release is slowed down considerably and a quiet, controlled combustion cycle achieved.

THERMODYNAMICS OF COMBUSTION PROCESS

Combustion in a diesel engine can be considered to commence at the dynamic injection point, when the liquid fuel is injected into the engine. The injected fuel is broked down into droplets and distributed in jets, throughout all or part of the combustion chamber, mixed with air, evaporated and diffused to produce a gaseous mixture for burning. These process are referred as ignition delay and the length of the delay period is of considerable importance since it affects the development of the processes following ignition and hence the rate of pressure rise, peak cylinder pressure, engine break mean effective pressure and exhaust emission, etc. During the delay period a considerable amount of fuel are injected into the combustion chamber and most of them are prepared for burning, rapid combustion are taken place immediately after the beginning of ignition.

Once the prepared fuel accumulated during the delay period has been consumed, the rate of burning falls to a value that can be maintained by the preparation of fresh fuel. The major controlling factor is the need for the fuel to find oxygen. Fuel may well be injected into the burning mixture so that combustion is in part regulated by the injection process as well as by the mixing and diffusion processes.

Finally after all the fuel has been injected, combution continues at a diminishing rate as the fuel and oxygen are consumed. This process and the previous combustion process are both characterized by the diffusion combustion, with production and combustion of carbon particles and a high rate of heat transfer by radiation.

COMBUSTION MODEL

The simplest single-zone combustion model specifies the heat release pattern in advance so that the cycle calculation merely involves adding energy to the cylinder contents in accordance with this pattern at the appropriate points in the calculations. This approach considers the cylinder contents to be a homogeneous mixture of air and gases which are always in thermodynamic equilibrium and dealt as ideal gases. Together with this assumption, in the present work, dissociation is neglected and only four gases are considered; oxygen, nitrogen, water vapor and carbon deoxide.

Wiebe's Combustion Function

Probably the most widely used heat release model is based on the Wiebe's combustion function. Wiebe specifies a function to represent the combustion curve, for the cumulative fuel burnt, as a fraction of the total fuel injected,

$$x = 1 - \exp(-a \cdot y^{m+1}) \qquad (1)$$

or in differential form (rate of fuel burning)

$$\frac{dx}{dy} = a \cdot (m+1) \cdot y^{m} \cdot \exp(-a \cdot y^{m+1}) \qquad (2)$$

The variable x represents the fraction of the mass of fuel burnt relative to the fuel injected, the variable y the time relative to the duration of combustion. The parameter a and m characterize the shape of combustion curve. Therefore, the heat release pattern is defined by the total amount of fuel injected, the start of combustion, the combustion duration and the shape parameters of Wiebe function. The start of combustion may be defined by the start of fuel delivery, the injection delay and the ignition delay. The start of fuel delivery and the injection delay is known from the geometric data of injection system. But the ignition delay is related to the cylinder gas temperature, pressure and fuel properties, so it is customary to calculate it according to a semiempirical expression of the Wolfer type (7):

$$\delta = b_1 \cdot \exp(b_2/Tm) \cdot P_m^{b_3} \qquad (3)$$

where $P_m$ and $T_m$ are the mean cylinder pressure and temperature during the delay period respectively, and $b_1$, $b_2$ and $b_3$ are empirical constants.

The combustion duration is an arbitrary period in which combustion must be completed. The actual point at which combustion ceases has little real significance since the combustion rate decays exponentially to almost zero long before combustion truly stops. Wochni and Anisits (8) found that the combustion duration was influenced by equivalence ratio and engine speed, for the medium speed diesel engine.

Parameter a in the Wiebe function can be considered as a combustion efficiency term, since the fraction of the cumulative fuel burnt to the total fuel injected is only dependent upon the parameter a. Therefore, a may be chosen such that all the fuel is burnt at the end of combustion (for example, x=0.999 at y=1 if a=6.9).

Then one may easily have a good approximation of a heat release pattern by varying the shape parapeter m. Fig. 2 shows us the burning rate curves for various values of m suggested by Wiebe. However, the heat release shape defined by the Wiebe function differs markedly in the rapid heat release at the onset of ignition found in the measured heat release curve on most types of high speed diesel engines. Generally, the combustion may by considered as the combination of the premixed and diffusive combustion. So, Murayama et al.(9) and Watson et al.(10) suggested that the heat release model composed of two heat release function, one for premixed and the other for diffusion combustion, produced the closer approximation of the heat releas shape for diesel engine combustion to that measured in practice, than would be the case with a single Wiebe function. Murayama et al. used two Wiebe functions, applied the premixed and the diffusion burning rate respectively,

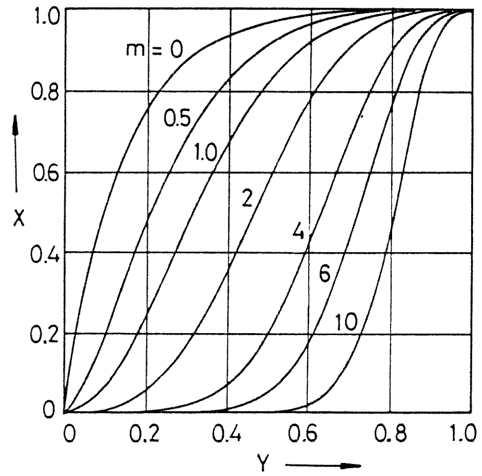
$$X_p = 1 - \exp(-6.9 y_p^{m_p+1}) \qquad (4)$$

$$X_d = 1 - \exp(-6.9 y_d^{m_d+1})$$

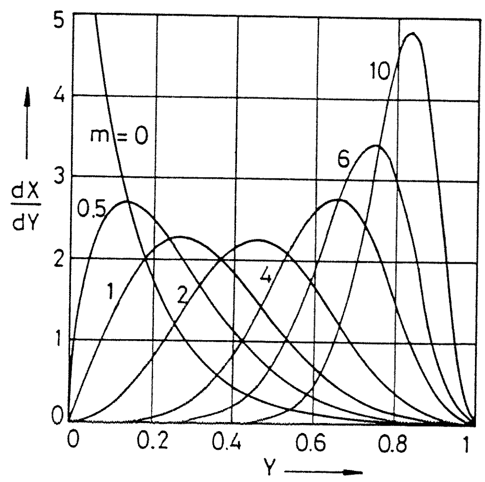
But Watson et al. used one Wiebe function for a diffusion combustion only and a simple algebraic expression for the premixed combustion, rather than a Wiebe function.

$$X_p = 1 - (1-y^{c_{p1}})^{c_{p2}}$$

$$X_d = 1 - \exp(-c_{d1}\, y^{c_{d2}}) \qquad (5)$$




a) Cumulative fuel burnt




b) Fuel burning rate

Fig. 2 Fuel burning rate curves for various values of m suggested by Wiebe

<u>Whitehouse-Way's Heat Release Model</u>

Whitehouse and Way have presented an alternative empirical model, based on elementary combustion principles. Fuel is injected into the engine in liquid form; before it can be burned, it must be heated and mixed with a sufficient quantity of oxygen for burning. These two physical process are referred to collectively as preparation. The prepared fuel may then be burn at a rate given by chemical kinetic equations.

Assuming that the rate of preparation of the fuel is proportional to the total surface area of the fuel droplets and allowing for the effect of oxygen availability on preparation, Whitehouse et al. suggest that the preparation rate may be given by

$$P = K \cdot Mi^{1-v} Mu^{v}\ Po_2^{w} \qquad (6)$$

where K, v and w are empirical constants, Mi is the total mass of fuel injected up to the time under consideration, Mu is the yet unprepared mass of fuel and $Po_2$ is the partial pressure of oxygen.

At the high temperatures corresponding to the main period of combustion, the time taken by the burning of the prepared fuel is negilgible compared with the preparation time. Therefore, for most of the burning period, the fuel is assumed to burn as rapidly as it is prepared. However, at the beginning of the burning period, since the temperature is too low for rapid burning the fuel can not burned immediately after prepared. A reaction rate equation based on the Arrhenius equation is used to link the preparation rate to the burning rate, during the early 'premixed' part of diesel combustion.

$$R = \frac{K' \cdot Po_2}{N \cdot \sqrt{T}} \cdot \exp(-act/T) \cdot \int (P-R)\, d\theta \qquad (7)$$

where K' and act are empirical constants, N is the engine speed and $d\theta$ is the step size of crank angle.

Fig. 3 illustrates the results of calculations using the Whitehouse-Way's heat release model. It will be noted that although no delay period has been referred to as such, the initial period of low or negative heat release is effectively inidentical to the well-known delay period.

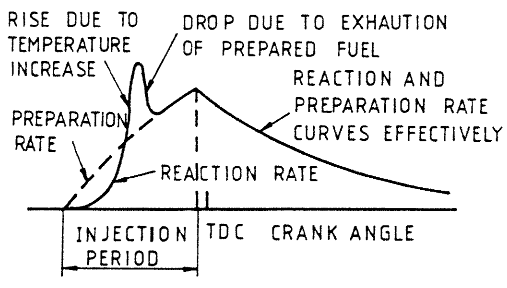



Fig. 3 Heat release rates calculated by Whitehouse-Way's model

## CALCULATION RESULTS

In the first step of the combustion modelling, the fuel burning rate was calculated from the measured cylinder pressure diagram, using the instantaneous energy balance of the cylinder contents, that is, the first law of thermodynamics (11), under the assumptions already mentioned in this paper. The results of the rate of heat release show the slow and smooth burning without any peak at the start of combustion (Fig. 7).

The combustion model was made by using the Wiebe's combustion function and the Whitehouse-Way's heat release model. In case of using the Wiebe function, the model composed of two Wiebe function was compared with the single Wiebe function model, in order to obtain a fine approximation of initial combustion stage. Therefore 3 types of heat release model were made and comapred with each other.

For each model, the effect of the variation of the model has been studied. The value of combustion duration in Wiebe function was determined from the curve of accumulated heat release. And a in equation (1) and (2) was chosen as 6.9 so that the combustion efficiency could be reached to 99.9% at the end of combustion. So, the single Wiebe function model may be decided by shape parameter, m only, and the two Wiebe function model by 4 parameters, namely shape parameters, $m_p$ and $m_d$, premixed combustion duration and the portion of the premixed combustion among the total injected fuel. Fig. 4 and 5 show us the effects of variation of parameters in Wiebe function. As the value of shape parameters $m_p$ and $m_d$, that is, the characteristic value of combustion, becomes smaller, the rate of heat release in the initial combustion stage is increased so that the cylinder pressure rises more rapidly and the peak cylinder pressure is increased also. The duration of premixed combustion is very important to the shape of the rate of heat release. If the smaller values of the pre-mixed combustion are chosen, the initial rapid combustion will be distinguished more markedly.

For the Whitehouse-Way's model, a fixed value of w in Equation (6) was used, according to the results in the paper by Whitehouse and Way. K' and act in equation (7) were also considered as constants that should be dependent only on the fuel and not at all on the engine. So, the effects of the variation of K and v were only studied and the results depicted in Fig. 6. As the preparation coefficient K increases, the fuel preparation rate is increased and the rate of heat release and the cylinder pressure in the initial combustion stage are increased very significantly. But the variation of v results in the reverse effect of K.

From the results of the variation of model coefficients the appropriate values of parameters for each model were chosen by judging the results of cylinder pressure and the shape of the rate of heat release,

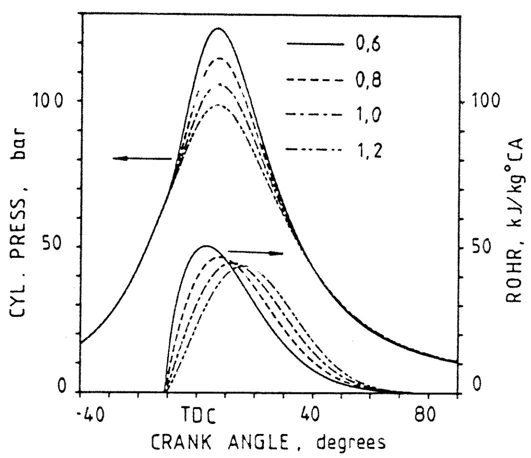
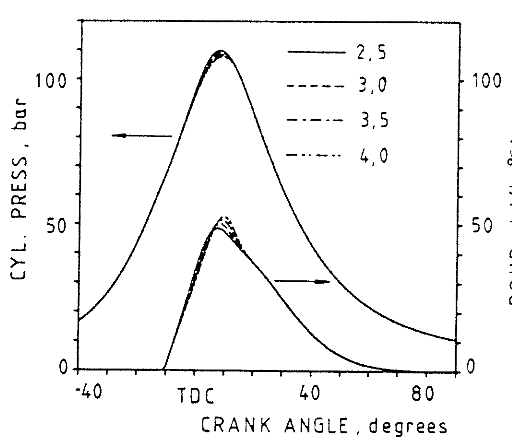
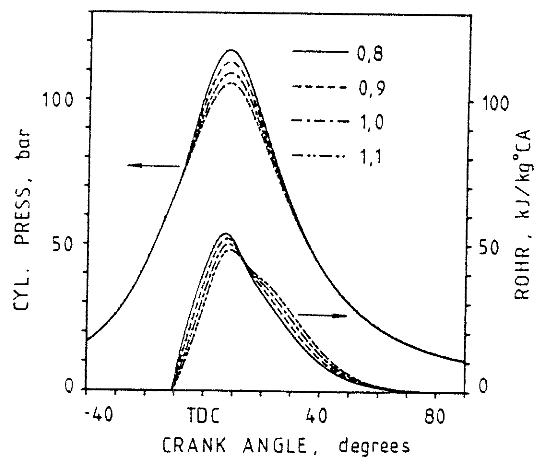


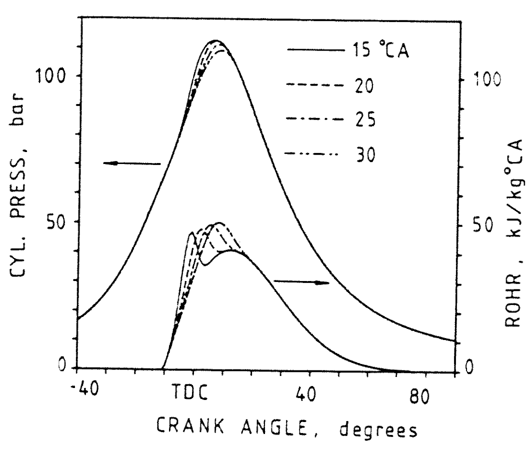

Fig. 4 Effect of shape factor m on the cylinder pressure and the rate of heat release in the single Wiebe function model



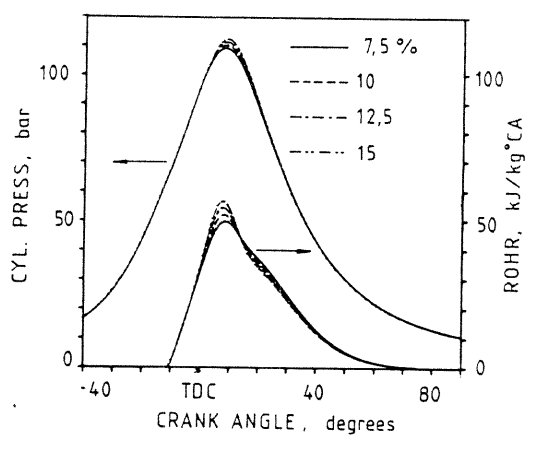

a) Shape factor, $m_p$




b) Shape factor, $m_d$




c) Premixed combustion duration




d) Faction of premixed combustion

Fig. 5 Effect of parameters on the cylinder pressure and the rate of heat release in the two Wiebe function model instead of the rates of heat release itself. In Fig. 7, the calculation results from the three different heat release models and the measured cylinder pressures were dipicted and compared with each other by the cylinder pressure and the rate of heat release. Since the data obtained from the experimentation showed the smooth rise of cylinder pressure and the rate of heat release at the initial stage of combustion, it was possible to obtain somewhat close approximation by a single Wiebe function model, though the results of two superimposed Wiebe function model were more accurate. In case of Whitehouse-Way's model, the rate of heat release in the delay period were calculated higher than the experimentation, so the pressure in the early stage of combustion were increased as compared with the other two models and the experimentation. This differences are considered as the results from the choice of the values of w, K' and act in Equation (6) and (7) as the fixed values independent on the engine. Therefore, in order to obtain the better approximation by Whitehouse-Way's model, it is necessary to take account the effects of the engine shape and operating conditions on the values of w, K' and act.

SUMMARY

In summary, following results have been obtained through the engine combustion simulation for the 4-stroke, direct injection and turbocharged diesel engine having 'M' type combustion chamber.

1. The combustion model using two-superimposed Wiebe functions that characterize the premixed and diffusion combustion respectively have the better prediction results than the one using the Whitehouse-Way's heat release model or the single Wiebe function.

2. In case of the two Wiebe function model, the value of shape parameter, $m_p$ and $m_d$, are in the range of 3.0-3.5 and 0.9-1.2, respectively, for the test engine.

3. If the effect of the engine shape and operating conditions of the constants, w, K' and act, in the Whitehouse-Way's model will be studied furthur, the prediction results by the Whitehouse-Way's model will become more accurate than the one obtained in present work.

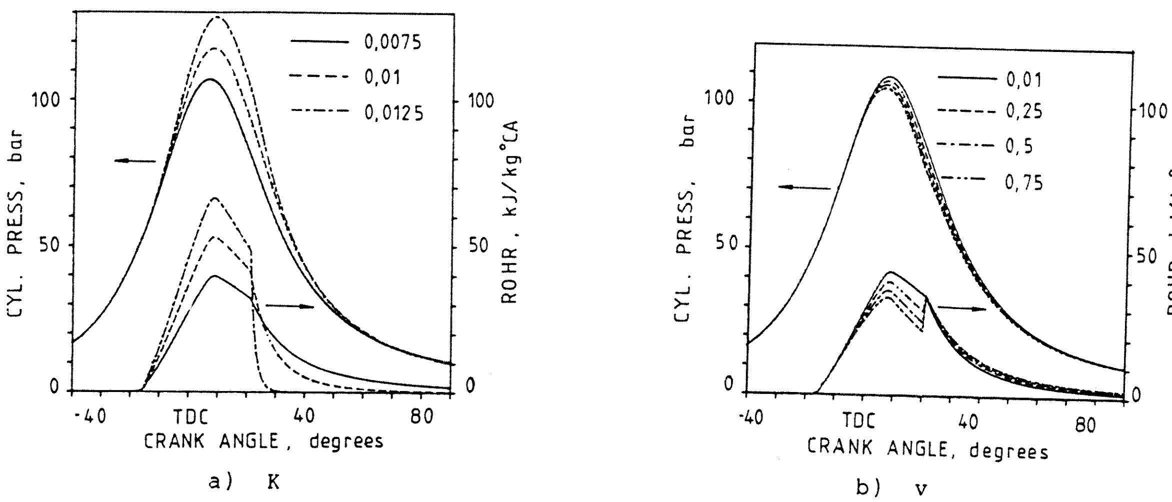
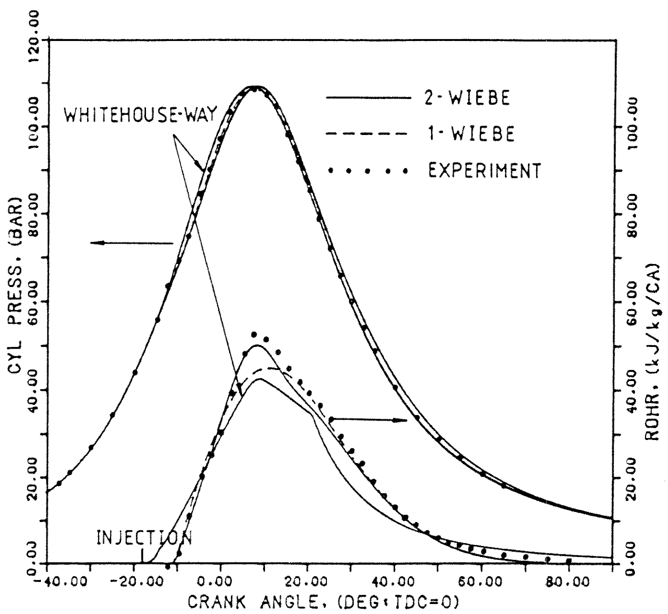
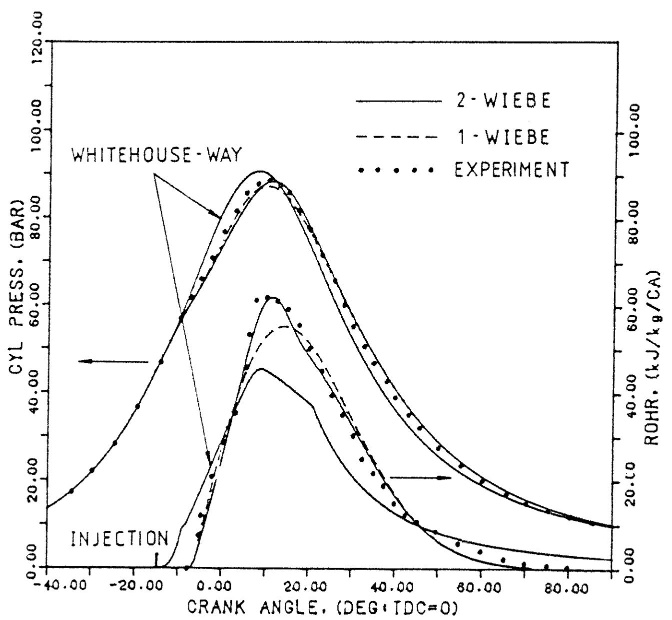



Fig. 6 Effect of parameters on the cylinder pressure and the rate of heat release in the Whitehouse-Way's model




a) 2200rpm, full load




b) 1400rpm, full load

Fig.7 Comparison of experimental and predicted results for test engine at rated speed and peak torque speed

## NOMENCLATURE

a = Combustion efficiency coefficient
act = Index in reaction rate equation, K
b = Empirical constants in the ignition delay equation (i=1,2,3)
$c_1, c_2$ = Shape factors in fuel burning rate equation (5)
K = Coefficient in preparation rate equation, bar
K' = Coefficient in raction rate equation, K/bar·sec
M = Mass of fuel, kg
m = Shape factor in Wiebe function
N = Engine speed, rpm
P = Rate of preparation of fuel, kg/°CA
$P_{O_2}$ = Partial pressure of oxygen in cylinder, bar
p = Cylinder gas pressure, bar
R = Rate of fuel burning (reaction), kg/°CA
T = Cylinder gas temperature, K
v, w = Indexes in preparation rate equation
x = Fration of fuel burnt relative to the total injected fuel
y = Time from ignition, relative to the duration of combustion
$\delta$ = Ignition delay, msec
$\theta$ = Crank angle, degrees

Subscripts
d = Diffusion
i = Injection
m = Mean value during ignition delay period
p = Premixed

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