Combustion Modeling and Cycle Simulation of Divided-Chamber Diesel Engines

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ABSTRACT

A new concept of "instantaneous heat release distribution" has been introduced to reveal the complex interaction between main- and pre- chamber which is a basic feature of divided- chamber engine combustion and is difficult to deal with in combustion modeling.

Based on the concept a combustion model for divided- chamber Diesel engines has been set up and a corresponding cycle simulation method developed. The results of simulation coincide well with experimental data and data analysis results. The accuracy of this simple method is thought to be satisfactory.

The method proposed here can be employed to analyse and predict the performance of divided- chamber Diesel engines.

INTRODUCTION

A number of comprehensive models for Diesel engine combustion (1,2,3,4,5) have been set up and applied to analyse and predict the behavior of engines, but most of these comprehensive models are applicable only for the direct-injection engines. As for the divided- chamber Diesel engine, partly because of the complexity of transient combustion phenomena both in main- and pre-chamber, simple and practical combustion model remains an interesting problem.

This paper discribes an investigations into the combustion modeling of divided- chamber Diesel engines. The authors' purpose is to develop a combustion model having the advantage of both simplicity and accuracy.

COMBUSTION MODEL

The review of present references (6, 7,8) on zero-dimensional model of divided-chamber Diesel engines indicates that most of the works have a common ground that is, using two pre-determined burning rate curves which are independent each other to model the combustion processes of two chambers respectively. Such a "single-chamber" criterion may simplify the calculation, and keep the heat release diagrams of two chambers straight. But in a divided -chamber Diesel engine, the two chambers are closely related during the whole combustion period due to the strong exchange of energy, mass and momentum through the connecting passage. The so-called "single -chamber" method is unable to show the interaction between two chambers. So it is difficult to get a flexible simulation of engine cycle when using the "single-chamber" method.

The method suggested by authors is a two-zone zero-dimensional method. Following assumtions have been introduced in the method. The working fluid is taken as semi-perfect gas, and the combustion process is taken to be a heat-adding process. The main- and pre- chamber are separate thermodynamic systems. In each system the homogeneous state is assumed. A total burning equation is used to discribe the engine combustion process (including two chambers). No assumptions on the combustion rate of each chamber have to be made. At the connecting passage, one-dimensional, quasi-steady, compressible gas flow is assumed.

As for Diesel engines there are two combustion forms, pre-mixed combustion and diffusion combustion, which have different heat release characteristics, thus two Wiebe fuctions (9) are used to make up the total burning equation:

$$X_t = F_b \cdot X_1 + (1 - F_b) \cdot X_2 \qquad (1)$$

$$X_i = 1 - exp(c_i \cdot \tau^{d_i}),\ i=1,2 \qquad (2)$$

F_b is the burning mode factor which presents the proportion of pre-mixed combustion.

From the total burning equation the combustion rates of two chambers could be calculated according to the mathematical discription of the relation of two chambers. Here a new concept-instantaneous heat release distributing rate η_b-has been proposed by authors, and it can be defined as:

$$\eta_b = \frac{dQ_{bp}}{d\varphi} \Big/ \left(\frac{dQ_{bm}}{d\varphi} + \frac{dQ_{bp}}{d\varphi}\right) \qquad (3)$$

The value of η_b represents the distribution between two chambers of heat released at the moment, so it should be a function of crankangle φ. Its change relating with crankangle can be determined as follows.

After injection, there should be η_b=1 from ignition to the moment the pressure of both chambers are equal. Soon afterwards because the pressure of the pre-chamber increases more rapidly than that of the main-chamber, a great deal of mixture is forced through the passage into the main-chamber due to the pressure difference of chambers. Once the main-chamber combustion starts, the value of η_b begins to decrease gradually.

It is evident that the more fuel injected into the main-chamber, the greater the transformation of heat release from pre-chamber to main-chamber, so the more rapidly the η_b is decreasing.

It is reasonable to simply assume that the relative changing rate of η_b is proportional to that of unburned fuel in pre-chamber G_p. So we can write:

$$\frac{d\eta_b}{d\varphi} \Big/ \eta_b \propto \frac{dG_p}{d\varphi} \Big/ G_p \qquad (4)$$

However, $dG_p/d\varphi$ is still unknown. To apply the assumption mentioned above the mixing concentrations in pre-chamber can be treated as uniform spatial distribution. After this manner we can substitute the relative changing rate of G_p for that of the working fluid mass in pre-chamber M_p:

$$\frac{d\eta_b}{d\varphi} \Big/ \eta_b \propto \frac{dM_p}{d\varphi} \Big/ M_p \qquad (5)$$

The mixing concentration in pre-chamber is quite uneven in fact, hence a simple function $A(\varphi)$ is adopted to improve the assumption mentioned above. So:

$$\frac{d\eta_b}{d\varphi} = A(\varphi) \cdot \eta_b \cdot \frac{dM_p}{d\varphi} \Big/ M_p \qquad (6)$$

According to the results of heat release analysis based on experimental data (see "Experimental Verification" for detail), $A(\varphi)$ takes the form of decreasing linear function:

$$A(\varphi) = 1 + K\left(\frac{\varphi_1 - \varphi}{\varphi_1 - \varphi_0}\right) \qquad (7)$$

A typical value of K is in range of 1-1.5. The form of $A(\varphi)$ means that the mixture injected into main-chamber in early period of combustion includes more unburned fuel with it.

In the model the value of η_b is calculated step by step for whole combustion process and at the same time the instantaneous heat release rates of main-and pre-chamber can be solved:

$$\frac{dX_m}{d\varphi} = (1-\eta_b)\frac{dX_t}{d\varphi} \tag{8}$$

$$\frac{dX_p}{d\varphi} = \eta_b \frac{dX_t}{d\varphi} \tag{9}$$

The schematic chart of this "heat releasing-distributing" model is shown in Fig. 1. The new feature of the model is it does not desire the assumptions on the combustion rates of main-and pre-chamber as in the conventional approaches, so it give rise to the advantage that the model can correctly take into account the interaction between two chambers.

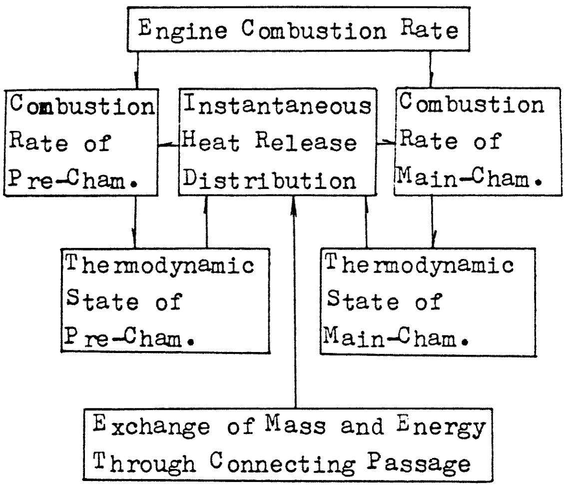


Fig. 1 Schematic chart of the model

CYCLE SIMULATION

On the ground of combustion model mentioned above a cycle simulation has been developed for divided-chamber Diesel engines. The heat transfer coefficients for the engine cylinder and pre-chamber are calculated according to Sitkei (10). The specific heats of gases in the main-and pre-chamber are calculated according to Krieger and Borman (11). The gas exchange process is modeled according to Ref. 12. The discharge coefficient of connecting passage was determined from steady flow tests, and was found to be a function of flow direction and pressure ratio. The test results are illustrated in Fig. 2. The difference of value μ at different flow directions is clear. Such a difference should be considered in model calculation.

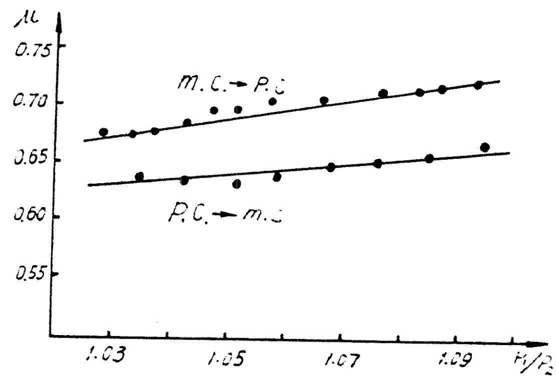


Fig. 2 Results of steady flow tests

A computer programm for engine cycle simulation is developed and applied to simulating the operating cycle of engine A (swirl-chamber type, D=95mm, S=115mm). The results of simulation are shown in Fig. 3 - Fig. 6.

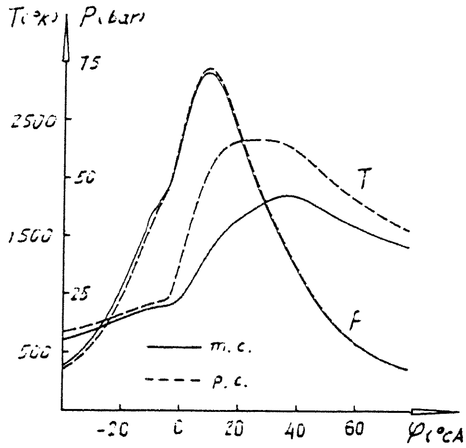
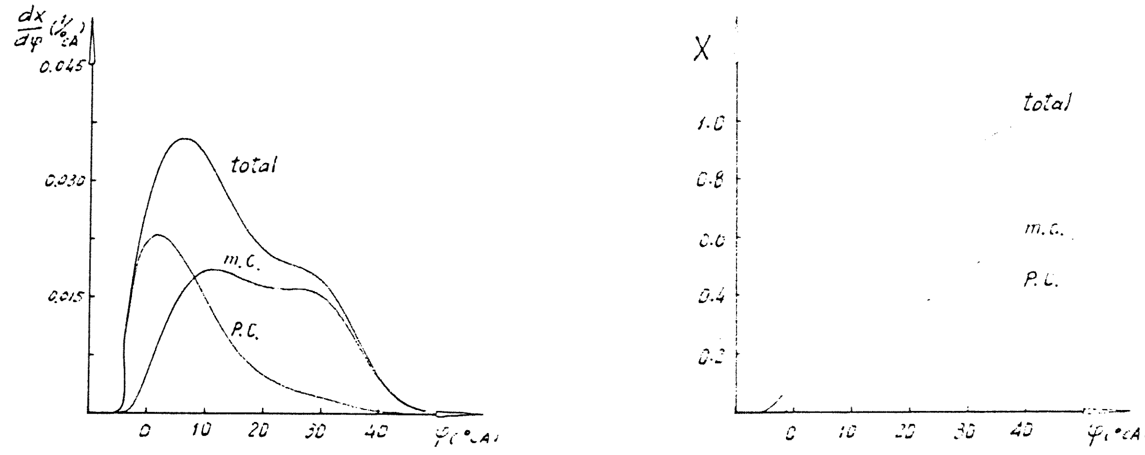


Fig. 3 Predicted pressure and mean temperature



(a) Instantaneous heat release rates (b) Cumulative heat release rates

Fig. 4 Heat release diagrams

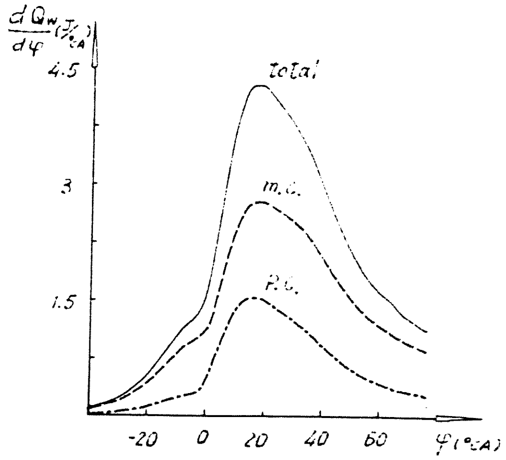


Fig. 5 Instantaneous heat transfer rates

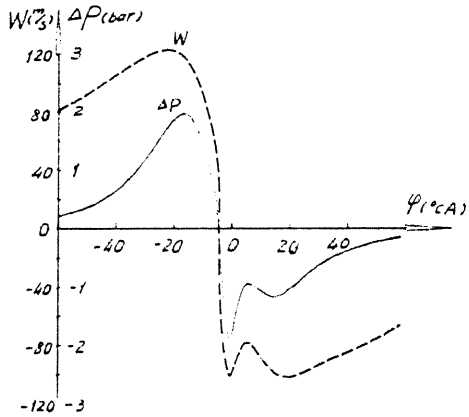


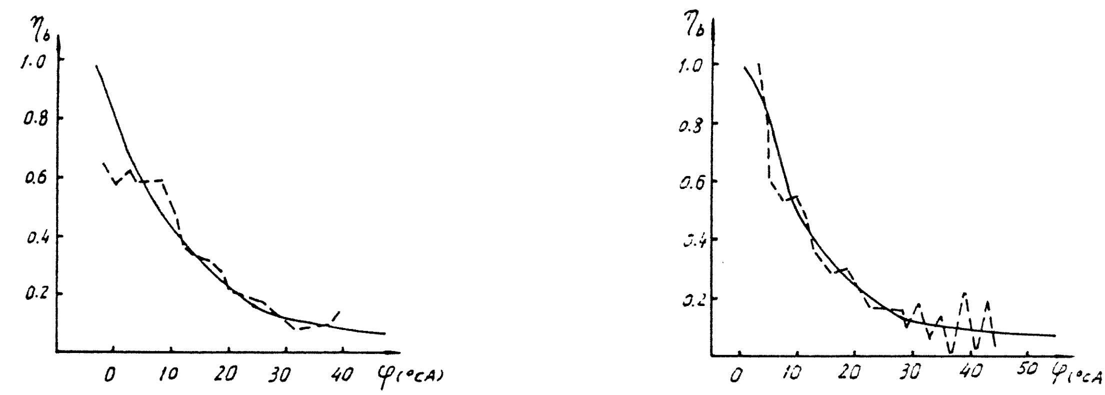
Fig. 6 Gas velocity through the passage and pressure difference of two chambers

EXPERIMENTAL VERIFICATION

For any combustion process of a divided-chamber Diesel engine there is a corresponding η_b curve. Through the heat release analysis based on measured indicator diagrams those experimental η_b curves can be got. The comparison of calculated and experimental η_b curves of engine A are shown in Fig. 7(a). In Fig.7 (b) η_b curves of engine B (swirl-chamber type, D=85mm, S=101.6mm) are compared.

The predicted η_b curves coincide well with the results of data analysis. This provides a strong support to the assumptions we have made in the model.

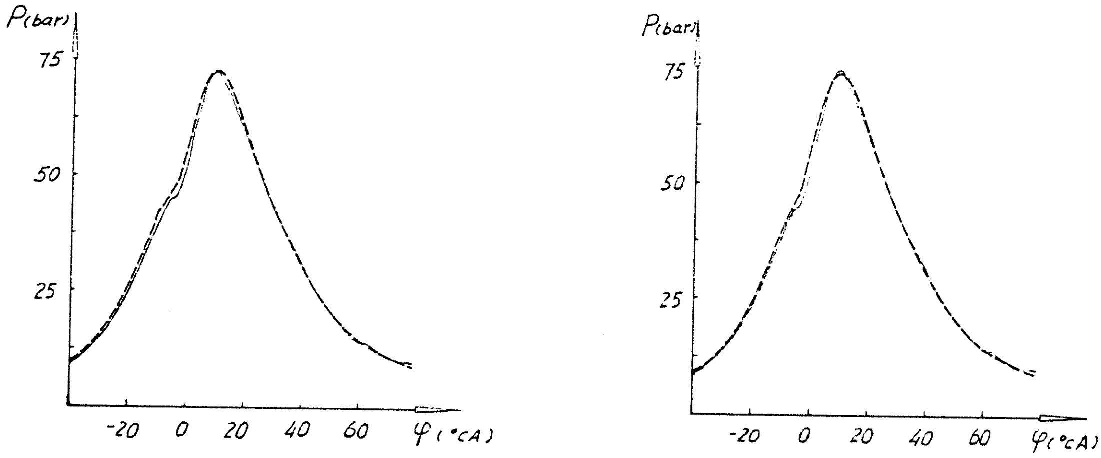
In Fig. 8 the predicted pressure diagrams are compared with experimental data. The comparison indicates that the accuracy of this simple method is quite good. The reason of this may be conceivable that the important factor, the close relation and interaction of two chambers, is taken into account correctly in the model.



(a) Engine A

(b) Engine B

Fig. 7 Comparison of η_b curves
(— modeling --- data analysis)



(a) main-chamber

(b) pre-chamber

Fig. 8 Comparison of predicted and measured pressure diagrams

PARAMETRIC STUDY

The cycle simulation is applied for some parametric studies in order to investigate the effects, potentiality and limitation of the method as a prediction method. More results and detailed discussion will be given in another paper.

Rv and Rf are two important geometric parameters of divided-chamber engines which represent the volume of pre-chamber and the cross section area of passage. Their effects on the behavior of engine are studied using the model. The variation of some interesting parameters is shown in Fig. 9 and Fig. 10.

Among the results, Rb is the proportion of heat released in pre-chamber to total heat released. q_w is the proportion of heat transfered to heat released. q_c is the proportion of cumulative gas kinetic energy through passage to heat released.

Gas velocity when it passes through the passage has intensive influence on the air motion in pre-chamber (in com-

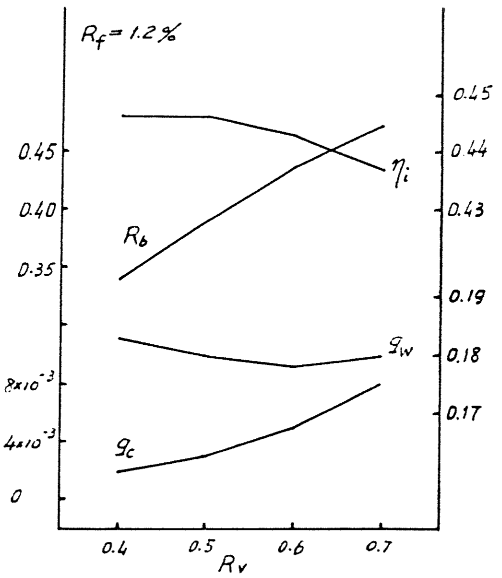


Fig. 9 The effects of Rv

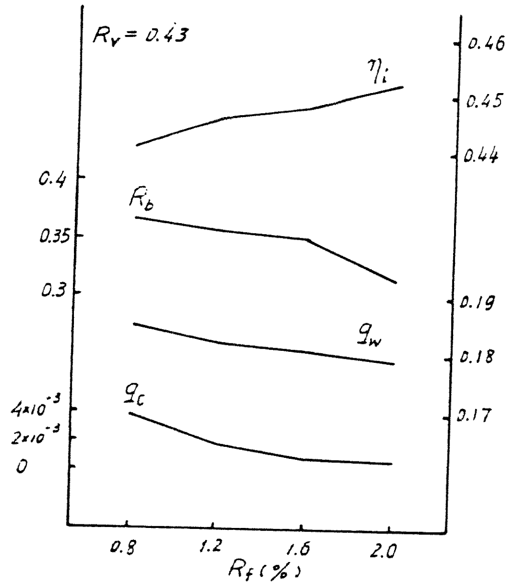


Fig. 10 The effects of Rf

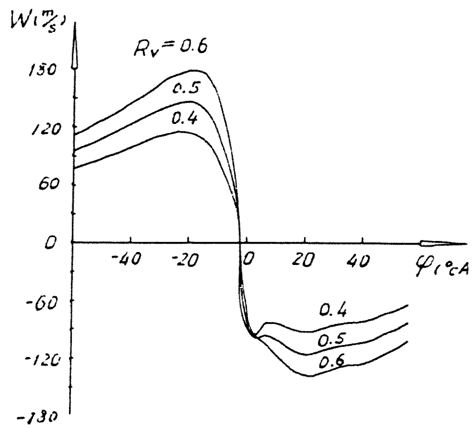


Fig. 11 Predicted W with different Rv

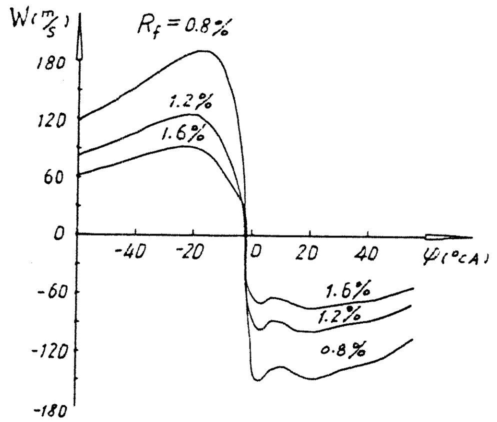


Fig. 12 Predicted W with different Rf

pression stroke) and main-chamber (in expansion stroke), therefore greatly affects the air-fuel mixing and combustion. Calculated W curves with different Rv and Rf are given in Fig. 11 and Fig. 12.

The results shown in Fig. 9 - Fig. 12 are examples of the application of the model. We should remember that the total burning equation in the model is not changed in these cases. In this way we can examine the effects of each parameter separately. When a wide range of engine operating conditions are concerned, some rational relations between burning equation parameters (Fb, c_1, d_1) and engine condition should be introduced in order to get a wide range simulation.

CONCLUSION

In the combustion process of a divided-chamber Diesel engine the intantaneous heat release distributing rate η_b is an important parameter. The expression of η_b is brief, concise and to the point. Its change relating with crankangle reveals the complex interaction between main-and pre-chamber which is a basic feature of divided-chamber Diesel engine combustion and the difficult point in combustion medeling. We suggest the application of the concept of the distributing rate η_b to heat-release analysis and combustion modeling of divided-chamber Diesel engines.

On the basis of the releasing-distributing approach a two-zone, zero-dimensional combustion model has been proposed and corresponding cycle-simulation method developed. The results of calculations coincide with experimental data. The accuracy of this simple predictive method is thought to be satisfactory. Further work is to examine and improve the model on the basis of more experimental data obtained from wider range of engine operating conditions.

NOMENCLATURE

$A(\varphi)$ = linear function in Eq. 6

c_i = coefficient in total burning equation

CA = crankangle

D = cylinder bore, m

d_i = index in total burning equation

Fb = burning mode factor

Gp = unburned fuel mass in pre-chamber,kg

K = constant in Eq. 7

Mp = working fluid mass in pre-chamber,kg

m.c.= main-chamber

P = pressure, bar

P1 = upstream pressure, bar

P2 = downstream pressure, bar

p.c.= pre-chamber

Qbm = heat released in main-chamber, J

Qbp = heat released in pre-chamber, J

q_c = percentage gas flow loss, relative to total heat added

q_w = percentage heat transfer loss, relative to total heat added

Rb = ratio of pre-chamber combustion heat to total combustion heat

Rf = ratio of the cross-sectional area of the passage to that of the piston

Rv = ratio of pre-chamber volume to total clearance volume at Top-Dead-Center

S = piston stroke, m

T = temperature, ^oK

W = gas velocity through the passage, m/s

X1 = mass fraction burned of pre-mixed combustion

X2 = mass fraction burned of diffusion combustion

Xt = total mass fraction burned

Δp = pressure difference of two chambers, bar

η_b = instantaneous heat release distributing rate

η_i = indicated thermal efficiency

μ = discharge coefficient of the passage

τ = time, s

φ = crankangle, ^{o}CA

φ_o = crankangle of combustion beginning, ^{o}CA

φ_1 = crankangle of combustion end, ^{o}CA

Subscripts

b = burning

c = gas flow loss

f = area

m = main-chamber

p = pre-chamber

t = total

v = volume

w = heat transfer

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