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An Analysis on Ignition Lag Characteristics of Divided Chamber Type Diesel Engines*

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In this paper, it was tried to analyze theoretically the ignition lag in the diesel engine with a divided chamber. At first, four influential factors on the ignition lag, namely the compression ratio, the volume ratio of divided chamber, engine revolution and a factor $K (=f_u/FC)$, were clarified. And the ignition lag characteristics were obtained in relation to these factors.

Further, the optimum factor K , namely K_{opt} , with which the ignition lag showed the shortest value, was obtained.

On the other hand, the behavior of the ignition lag in the case when the sectional area of connecting passage to the divided chamber was varied, was made clear.

1. Introduction

In diesel engines, the ignition lag is an important factor which influences the combustion process in the cylinder, output performances, the combustion noise, exhaust gas composition, the startability etc.^(1), 2) And, it may cause a serious difficulty when an increase of the engine speed is intended.

Thus, numerous works on the ignition lag have been reported hitherto. However, the majority of works of this type were carried out under the condition of the unvaried temperature and pressure. Since the ignition phenomenon under the varied temperature and pressure has been clear theoretically to some extent, the characteristics of the ignition lag in the diesel engine are here to be clarified.

While numerous works in the divided chamber type diesel engines have been done hitherto, most of them are concerned with throttling loss etc.^{3), 4), 5), 6)}

However, the ignition lag in divided chamber type diesel engines has hardly been analyzed in detail because of its intricacy. Hence, we have attempted to clarify the problems involved. Namely, equations were obtained first to determine the temperature and pressure processes in the main and divided chambers in the diesel engines. Then, by calculating the ignition lag in both chamber, the relationships between the ignition lag and the various characteristic values were obtained theoretically.

As a result of calculations, the relationships were clarified between the ignition lag and the following three factors, namely, the volume ratio of a divided chamber, the sectional area of the connecting

passage to a divided chamber and the engine speed.

Consequently, one of the guide lines for the combustion chamber specifications in a divided chamber type engine could be obtained for improving the ignition lag.

It was also attempted to clarify the ignition lag characteristics in the case when the sectional area of the connecting passage to the divided chamber was varied.

The detailed results will be described below.

2. Calculations of ignition lag in divided chamber type diesel engines

Nomenclature

A : thermal equivalent of work
C : mean piston speed
Cp : specific heat at constant pressure
Cv : specific heat at constant volume
f : sectional area of the connecting passage to a divided chamber
F : sectional area of cylinder
g : gravity acceleration
G : weight of gas
k : ratio of specific heat
l : length of connecting rod
n : engine speed
P : pressure of gas
r : crank radius
R : gas constant
t : time
T : temperature of gas
v : specific volume
V : volume of gas
β : volume ratio of a divided chamber
ε : compression ratio
θ : crank angle
μ : flow coefficient

Suffixes 1 and 2 represent the main and divided chambers respectively as shown in Fig. 1.

2.1. Pressure and temperature processes in the main and divided chambers

In this paper, since the ignition lag characteristics are to be examined qualita-

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Fig. 1 Model of combustion chamber

tively, various conditions of calculations are treated as simply as possible. Namely, the following assumptions are included in our calculations;

(1) The working fluid is the air and is treated as a perfect gas.

(2) The heat flowing into and out of the combustion chambers is negligibly small.

(3) The gas movement in the main and divided chambers, for example, eddy, turbulence etc., is neglected.

V_1 is described generally as follows,

$$V_1 = F r E \quad (1)$$

where,

$$E = 1 - \cos \theta + \frac{r}{l} \left(1 - \sqrt{1 - \frac{r^2}{l^2} \sin^2 \theta} \right) + (1 - \beta) \frac{2}{\epsilon - 1}$$

In general compression stroke, namely, $P_1 \geq P_2$, the following energy equation holds in the main chamber 1.

$$T_1 C_p dG_1 = C_v d(G_1 T_1) + A P_1 dV_1 \\ = A (k P_1 dV_1 + V_1 dP_1) / (k - 1) \quad (2)$$

Cocerning the divided chamber 2, one has,

$$T_2 C_p dG_2 = C_v d(G_2 T_2) = A V_2 dP_2 / (k - 1) \quad (3)$$

Since the total gas in two chambers is unvaried, the following equation holds,

$$dG_1 = -dG_2 = d(V_1/v_1) = -d(V_2/v_2) \quad (4)$$

And, dG_1 is given as follows, namely,

$$dG_1 = -\mu f \psi_{12} \sqrt{2g P_1 / v_1} dt \quad (5)$$

where

$$\psi_{12} = \sqrt{\frac{k}{k-1} \left\{ \left(\frac{P_2}{P_1} \right)^{\frac{2}{k}} - \left(\frac{P_2}{P_1} \right)^{\frac{k+1}{k}} \right\}} \quad \text{when} \quad \frac{P_2}{P_1} > \left(\frac{2}{k+1} \right)^{\frac{k}{k-1}} \\ \psi_{12} = \left(\frac{2}{k+1} \right)^{\frac{k-1}{k}} \sqrt{\frac{k}{k+1}} \quad \text{when} \quad \frac{P_2}{P_1} \leq \left(\frac{2}{k+1} \right)^{\frac{k}{k-1}}$$

From Eqs. (2) and (5), the following equation is obtained.

$$k P_1 \frac{dV_1}{dt} + V_1 \frac{dP_1}{dt} = -k \mu f \psi_{12} \sqrt{2g} P_1^{\frac{3}{2}} v_1^{\frac{1}{2}}$$

Putting $d/dt = n\pi/30 d/d\theta$ and $C = 2\pi r n/30$ into the above equation, we get,

$$k P_1 \frac{dE}{d\theta} + E \frac{dP_1}{d\theta} = -\frac{2k}{\pi} K \psi_{12} \sqrt{2g} P_1^{\frac{3}{2}} v_1^{\frac{1}{2}} \quad (6)$$

where $K = f\mu/FC$

In the same manner as Eq. (6), one can obtain the next equation from equations (3), (4) and (5).

$$B \frac{dP_2}{d\theta} = \frac{2k}{\pi} K \psi_{12} \sqrt{2g} P_1^{\frac{3}{2}} v_1^{\frac{1}{2}} \quad (7)$$

where $B = 2/(\epsilon - 1)$

In addition, the enthalpy which flowed out from the main chamber into a divided chamber, was assumed to be converted to the heat without any loss.

From Eqs. (4) and (5),

$$d\left(\frac{V_1}{v_1}\right) = -\mu f \psi_{12} \sqrt{2g} P_1 / v_1 dt$$

Putting Eq. (1) into above equation, one has,

$$\frac{dE}{d\theta} - E \frac{d}{d\theta} \left(\ln \frac{T_1}{P_1} \right) = -\frac{2}{\pi} K \psi_{12} \sqrt{2g} P_1 v_1 \quad (8)$$

From Eqs. (4) and (5),

$$-d\left(\frac{V_2}{v_2}\right) = -\mu f \psi_{12} \sqrt{2g} P_1 / v_1 dt$$

$$-\beta B \frac{P_2}{R T_2} \frac{d}{d\theta} \left(\ln \frac{T_2}{P_2} \right) = \frac{2}{\pi} K \psi_{12} \sqrt{2g} P_1 v_1 \quad (9)$$

Rearranging Eqs. (6), (7), (8) and (9), one has the following equation.

$$\left. \begin{aligned} k P_1 \frac{dE}{d\theta} + E \frac{dP_1}{d\theta} &= -\frac{2k}{\pi} K \psi_{12} \sqrt{2g} R P_1 T_1^{\frac{1}{2}} \\ \beta B \frac{dP_2}{d\theta} &= \frac{2k}{\pi} K \psi_{12} \sqrt{2g} R P_1 T_1^{\frac{1}{2}} \\ \frac{dE}{d\theta} - E \frac{d}{d\theta} \left(\ln \frac{T_1}{P_1} \right) &= -\frac{2}{\pi} K \psi_{12} \sqrt{2g} R T_1^{\frac{1}{2}} \\ -\beta B \frac{d}{d\theta} \left(\ln \frac{T_2}{P_2} \right) &= \frac{2}{\pi} K \psi_{12} \sqrt{2g} R \frac{T_2}{P_2} T_1^{\frac{1}{2}} \end{aligned} \right\} \quad (10)$$

Equation (10) holds when $P_1 \geq P_2$. On the other hand, when $P_1 < P_2$, the following equation (11) is obtained by putting

$$\left. \begin{aligned} k P_1 \frac{dE}{d\theta} + E \frac{dP_1}{d\theta} &= \frac{2k}{\pi} K \psi_{21} \sqrt{2g} R P_2 T_2^{\frac{1}{2}} \\ \beta B \frac{dP_2}{d\theta} &= -\frac{2k}{\pi} K \psi_{21} \sqrt{2g} R P_2 T_2^{\frac{1}{2}} \\ \frac{dE}{d\theta} - E \frac{d}{d\theta} \left(\ln \frac{T_1}{P_1} \right) &= \frac{2}{\pi} K \psi_{21} \sqrt{2g} R \frac{T_1}{P_1} P_2 T_2^{\frac{1}{2}} \\ \beta B \frac{d}{d\theta} \left(\ln \frac{T_2}{P_2} \right) &= \frac{2}{\pi} K \psi_{21} \sqrt{2g} R T_2^{\frac{1}{2}} \end{aligned} \right\} \quad (11)$$

Since the unknown variables are P_1 , T_1 , P_2 and T_2 respectively in Eqs. (10) and (11), each unknown variable is expected to be obtained by solving the four equations simultaneously.

As seen in Eqs. (10) and (11), in the case when the initial condition of calculation is constant, the behaviors of solutions $P(\theta)$ and $T(\theta)$ should be determined apparently by three factors, namely the volume ratio of a divided chamber β , the compression ratio and $K (=f\mu/FC)$.

2.2. Ignition lag

As well known, the ignition lag is affected by numerous and complicated factors. In this paper, for the purpose of grasping the basic characteristics of ignition lag, the ignition lag was assumed to be a simple function of temperature, pressure and time. Accordingly, Livengood's relation⁷⁾ on the ignition lag was used in these calculations. Namely, in the case when the ignition of fuel spray occurs under the conditions of varied pressure and temperature, the productive velocity of the activated particles

is represented as follows by making an assumption that a functional relation can be obtained between the concentration of the activated particles X and the relative time t/σ .

$$\frac{d}{dt}\left(\frac{X}{X_c}\right) = \phi\left(\frac{t}{\sigma}\right) \quad (12)$$

where σ is a function of the ignition lag under constant conditions which has been reported hitherto by a few researchers. And, X_c represents the critical concentration of activated particles for ignition, so that the ignition will occur at condition $X/X_c=1$.

Integrating Eq. (12) by representing τ' as the ignition lag in time, the following equation holds at the moment of ignition.

$$\frac{X}{X_c} = \int_0^{\tau'} \phi\left(\frac{t}{\sigma}\right) dt = 1 \quad (13)$$

By assuming that the productive velocity of the activated particles does not vary under constant conditions, Eq. (13) is represented as follows,

$$\int_0^{\tau'} \frac{1}{\sigma} dt = 1 \quad (14)$$

As for σ , Wolfer's equation was used in the calculations. namely, the next relation was given for typical fuel, namely gas oil.

$$\sigma(P, T) = 0.44 \times 10^{-3} P^{-1.19} \exp\left(\frac{4650}{T}\right) \text{ sec} \quad (15)$$

As the ignition lag is described generally in the crank angle in the internal combustion engines, τ' in Eq. (14) was replaced by τ in crank angle. Namely,

$$\int_0^{\tau} \frac{1}{\sigma(\theta)} d\theta = \frac{\pi\pi}{30} \quad (16)$$

2.3. Detailed procedures and examples of calculated results

It is quite difficult to obtain an analytical solution of Eqs. (10) and (11). Thus, numerical solution are to be obtained by advanced Euler-Cauchy methods. Besides, the relation between the calculating interval $\Delta\theta$ and the calculated value τ in ignition lag is shown in Fig. 2. As seen from this figure, when the calculating interval is smaller than 0.1° in crank angle, the calculated value becomes nearly constant.

Accordingly, in our calculations, a sufficient precision is secured by selecting the calculating intervals at 0.01° .

On the other hand, the examples of the results calculated from Eqs. (10) and (11) are shown in Figs. 3~6. As seen in these

Fig. 2 Calculating interval $\Delta\theta$ and calculated results τ

Fig. 3 Processes of pressure ($K=60 \cdot 10^{-5} \text{ s/m}$)

Fig. 4 Processes of temperature ($K=60 \cdot 10^{-5} \text{ s/m}$)

Fig. 5 Processes of pressure ($K=25 \cdot 10^{-5} \text{ s/m}$)

Fig. 6 Processes of temperature ($K=25 \cdot 10^{-5} \text{ s/m}$)

figures, in accordance with a decrease in K , the maximum pressure $P_{2\max}$ in the divided chamber decreases, but the maximum pressure $P_{1\max}$ in the main chamber tends to increase in reverse.

The crank position of the maximum pressure in the cylinder tends to advance according to a decrease in K . But, when K decreases further below $20 \cdot 10^{-5}$ s/m, its position approaches contrariwise the top dead center.

Fig. 7 Relation between maximum temperature and K in a divided chamber

Fig. 8 Relation between maximum pressure and K in a divided chamber

Fig. 9 Relation between fuel injection timing and ignition lag ($K = 74 \cdot 10^{-5}$ s/m)

Fig. 10 Relation between fuel injection timing and ignition lag ($K = 25 \cdot 10^{-5}$ s/m)

Fig. 11 Relation between fuel injection timing and ignition lag ($K = 11 \cdot 10^{-5}$ s/m)

Concerning the process of temperature, the temperature in the divided chamber after the top dead center or thereabouts, is higher than that in the main chamber.

The ignition lag is calculated by using the processes of the pressure and the temperature stated later.

The detailed results of the ignition lag will be described later.

3. Calculating results and discussions

3.1. Maximum temperature and pressure in the divided chamber

Prior to calculation of the ignition lag, the compression characteristics in the divided chamber was first treated. For a simple explanation of the compression characteristics, the maximum temperature and pressure were adopted as typical characteristics of the compression in the divided chamber type engine.

In addition, as already described in section 2.1, these typical characteristics were the volume ratio of the divided chamber β , the compression ratio and K ($=f_u/FC$).

Figure 7 shows the relationship between K and the maximum compression temperature $T_{2\max}$ as well as its crank position where the maximum temperature was obtained.

In this figure, T_c is the maximum compression temperature obtained in the single combustion chamber. Accordingly, in the case where $K \geq 200 \cdot 10^{-5}$ s/m, $T_{2\max}$ was almost the same as T_c and the crank position where $T_{2\max}$ appeared to approach the true top dead center, so that the compression characteristics in a divided chamber appeared to be the same as those of a single combustion chamber.

On the other hand, in the case where $K < 200 \cdot 10^{-5}$ s/m, as K decreased, $T_{2\max}$ tended to increase first and this tendency became remarkable as β became larger.

In this case, $T_{2\max}$ had its maximum value at a certain value of K . And, as K decreased from this value of K , $T_{2\max}$ decreased gradually. The value of K where $T_{2\max}$ had its maximum value, was varied

with β as shown by a broken line in Fig. 7. However, the maximum value of $T_{2\max}$ was obtained at $K \approx 15 \cdot 10^{-5}$ s/m or thereabouts in the range of $0.3 < \beta < 0.8$.

The maximum compression pressure in the divided chamber, as shown in Fig. 8, coincided with that in the single chamber P_c in a range where $K \geq 200 \cdot 10^{-5}$ s/m. And, as K decreased from $200 \cdot 10^{-5}$ s/m, $P_{2\max}$ decreased abruptly after showing a tendency of slight increase.

In the case where $K < 20 \cdot 10^{-5}$ s/m, $P_{2\max}$ decreased as β increased. But, in contrast, when β increased further and approached almost 1, $P_{2\max}$ tended to increase to some extent.

Consequently, it was clear that the temperature and pressure in the divided chamber were affected largely by K , and a certain value of K where the temperature and the pressure in the compression stroke reached their maximum was recognized in this section.

Thus, the ignition characteristics which are affected mainly by the temperature and pressure, will be analyzed in the following section.

3.2 Effects of injection timing on ignition lag

The injection timing is one of the important factors which affect the ignition lag. The relationships between the injection timing and ignition lag for different value of K , are shown in Figs. (9)~(11). In this case, the calculating conditions were as follows; namely, $\epsilon = 15$, $n = 2000$ rpm and $\beta = 0.3$.

From these figures, the optimum injection timing where the ignition lag was

Fig. 12 Relation between K and optimum fuel injection timing

Fig. 13 Effects of K on ignition lag

shortest, was recognized to exist in the main and divided chambers. And, two optimum injection timings in the main and divided chambers tended to separate from each other due to a decrease in K . Namely, as K decreased, the optimum injection timing in the main chamber tended to advance, but to lag in the case of further reduction of K , and finally conversed, namely a certain timing was reached before top dead center as seen in Fig. 12. In addition, this convergence corresponds to the optimum injection timing in the main chamber when the connecting passage would be closed.

On the other hand, the optimum injection timing in the divided chamber tended to lag in accordance with a decrease in K . This phenomenon seemed to result from a lag of the temperature and pressure rise in the divided chamber due to a decrease in K .

Consequently, the ignition lag was affected by the injection timing to a large extent, so much so that the description below will be conducted by using the ignition lag which would be obtained at the optimum timing in both combustion chambers.

3.3. Effects of K and β on ignition lag

Figure 13 shows the relationships among the ignition lag, K and β . In this case, the compression ratio was assumed to be 15 and the engine speed 2000 rpm.

In these figures, the real lines show

Fig. 14 Relation between ϵ and ignition lag

Fig. 15 Relation between n and ignition lag

the ignition lag in the main chamber and the broken lines also show that in the divided chamber.

In the main chamber, as K decreased, the ignition lag was shortened and finally approached that in the case of the single combustion chamber.

On the other hand, the ignition lag in the divided chamber was almost the same as that in the main chamber in a large range of K . But, as K decreased, it tended to decrease maintaining a lower level than the ignition lag in the main chamber, and took a minimum value at a certain value of K , namely K_{opt} .

After that, as K decreased from K_{opt} , the ignition lag increased.

Consequently, in a range where $K < K_{opt}$, the ignition lag in the divided chamber increased with a decrease of K . And, in the range where $K \geq K_{opt}$, the ignition lag decreased with a decrease of K , so that K_{opt} may be an important factor which determines the ignition lag characteristics in the divided chamber type diesel engine.

When the ignition lags in the main and divided chambers were compared with each other, it may be seen that the ignition lag in the divided chamber was shorter than that in main chamber in a range of larger K compared with K_{opt} .

In addition, the effects of K on ignition lag were larger with an increase in value of β .

3.4. Effects of compression ratio ϵ and engine speed n on ignition lag

As already stated, the processes of temperature and pressure in combustion chamber are determined largely by K , β and ϵ . And, the ignition lag is also affected by the engine speed n except for the factors stated above.

In this section, the effects of ϵ and n , as well as K and β , on the ignition lag were investigated. Figure 14 shows the ignition lag characteristics obtained under varied compression ratios. In this case, β is 0.3 and n is 2000rpm.

As seen in this figure, K_{opt} is rec-

Fig.16 K_{opt} and various factors

ognized for each compression ratio and their characteristics of the ignition lag show the same tendency as that stated in section 3.3. However, as the compression ratio increased, K_{opt} tended to increase slightly and the change of the ignition lag by K became small.

That is to say, the shortening of the ignition lag in combustion chamber is made more remarkable by varying K , when the compression ratio is lower.

On the other hand, concerning the effects of engine speed on the ignition lag characteristics, the ignition lag at $K = \infty$ increased linearly with an increase of engine speed, but in this case, the value of K_{opt} was almost unchanged as shown in Fig.15. Besides, the change of ignition lag with varied K became smaller as the engine speed decreased.

3.5. Effects of various factors on K_{opt}

As described above, K_{opt} seems to be the most important factor to determine the ignition lag characteristics in a divided chamber. Thus, the relationships between K_{opt} and the factors affecting K_{opt} , namely, the volume ratio of a divided chamber β and the compression ratio ϵ , are examined in this section.

In addition, in calculations for engine speed up to 4000rpm, K_{opt} is hardly affected by engine speed.

It is clear from Fig.16 that K_{opt} becomes larger at lower compression ratio and takes its maximum value at $\beta = 0.5 \sim 0.6$. And, the maximum value of K tends to move slightly in the direction of large values of β as the compression ratio increases. In the case of $\beta = 1$, the gas in the main chamber becomes zero at the top dead center at the compression end, and the whole amount of gas in the main chamber is forced to flow into a divided chamber regardless of K . Accordingly, the temperature and pressure in the divided chamber increase as K increases, and K_{opt} appears at $\beta = 1$ as seen in Fig.16.

3.6. Ignition lag characteristics in a case where the sectional area of passage to a divided chamber is varied

Fig.17 Model of variable area of passage to a divided chamber

The sectional area of connecting passage to the divided chamber f is generally constant in a common diesel engine. However, a variable area of passage has been examined by a few researchers⁽⁹⁾ for the purpose of improving the combustion noise, exhaust emission etc.

In this section, by giving a model of the variable area of passage as shown in Fig.17, the ignition lag characteristics obtained by varying the area of the passage, namely throttling, are investigated. In this model, the normal sectional area of connecting passage f is assumed to be throttled to $\delta \cdot f$ during θ_t in crank angle before and after the top dead center. The ratio of throttling δ is $0 \leq \delta \leq 1$.

Fig.18 Effects of δ and θ_t on ignition lag in a divided chamber

Fig.19 Effects of ϵ and θ_t on ignition lag in a main chamber

Figure 18 shows the relationships among the ratio of throttling δ , the duration of throttling θ_t and ignition lag τ_2 . In this case, the calculating conditions are as follows; $\epsilon = 15$, $\beta = 0.3$, $n = 2000$ rpm and $K = 75 \cdot 10^{-5}$ s/m. And, $\theta_t = 360^\circ$ corresponds to the case where the sectional area of passage f is not varied by the crank angle, so that the tendency of an ignition lag obtained by varying δ under condition $\theta_t = 360^\circ$ should be the same as that shown in Fig.13 which was calculated under $\beta = 0.3$. Thus, in the case $\theta_t = 360^\circ$, the minimum value of ignition lag is obtained at $\delta = 0.27$ which corresponds to $K_{opt} = 20 \cdot 10^{-5}$ s/m.

As shown by a broken line in Fig.18, the minimum value of ignition lag is on the curve drawn between its minimum value at $\theta_t = 360^\circ$ and the point at $\theta_t = 0$, $\delta = 1$. And, on the broken line, as θ_t decreases the minimum value of ignition lag increases and finally reaches the same value which is obtained under the condition of unvaried sectional area of passage and $\theta_t = 0$.

On the other hand, the ignition lag on the following three lines is the same as that which is obtained under the condition of unvaried sectional area of passage, namely, δ axis, the axis of $\delta = 1$ and the limit line as shown in Fig.18.

Thus, when a shortening of ignition lag in the divided chamber is intended by throttling the sectional area of passage, δ and θ_t must be selected so that the values fall between the limit line and $\delta = 1$. And, regarding the characteristics of ignition lag between the limit line and $\delta = 0$, as δ decreases and θ_t increases, the ignition lag becomes larger than that obtained by non-throttling.

In the case where $K < K_{opt}$, an improvement of the ignition lag can not be obtained by throttling. And, in the case where $K \geq K_{opt}$, an improvement of the ignition lag is possible because the line of minimum ignition lag is between $\delta = 0$ and $\delta = 1$. Furthermore, as K becomes larger than K_{opt} , the extent of improvements increases, namely, the limit line and the line of minimum ignition lag seem to approach θ_t axis.

On the other hand, the ignition lag characteristics in the main chamber are shown in Fig.19. As seen in this figure, the zone where the ignition lag increases, is not clearly recognized. And, as δ decreases and θ_t increases, the ignition lag decreases.

Besides, the ignition lag on the δ axis and $\delta = 1$, is the same as that obtained under non-throttling.

Consequently, the characteristics of the ignition lag can be determined by the value of K , even when the throttling of passage is adopted.

4. Conclusions

In this paper, for the purpose of making clear the ignition lag characteristics in a divided chamber type diesel engine, the basic equations which determined the processes of the pressure and temperature in combustion chambers, and the various characteristics values which determined the

state in chamber, were first derived. Furthermore, the ignition lag characteristics were also investigated by using the calculated processes of pressure and temperature.

On the other hand, in the case where the variable sectional area of the connecting passage to the divided chamber was adopted, its characteristics of the ignition lag were also made clear to a considerable extent.

The results of this paper may be summarized as follows;

(1) Under constant initial conditions at compression beginning, the processes of pressure and temperature in combustion chambers are determined by the following three characteristics values, namely the compression ratio, volume ratio of the divided chamber and $K (=f_v/FC)$.

(2) Both in the main and divided chambers, the optimum injection timing where the ignition lag takes a minimum value, is obtained by calculations.

As K decreases, the optimum injection timing tends to lag especially in the divided chamber.

(3) The ignition lag in the divided chamber takes a minimum at a certain value of K , namely K_{opt} . And, K_{opt} is hardly affected by the engine speed, but is determined mainly by compression ratios and volume ratio of a divided chamber.

(4) The ignition lag in the main chamber is shortened as K decreases.

(5) Concerning the effects of throttling the passage to the divided chamber on the ignition lag, the improvement of the ignition lag by throttling is obtained only under the condition $K \geq K_{opt}$.

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