621.436.01

Studies on Low Compression Ratio Diesel Engine*

(3rd Report, Relation between Combustibility and Performances)

By Noboru MIYAMOTO**, Tadashi MURAYAMA***, and Shouichi FUKAZAWA***

In a diesel engine, by a systematic controlling of the burning rate, the performances i.e., thermal efficiency, maximum pressure, maximum rate of pressure rise, combustion noise emission and maximum combustion temperature which affects the density of NOx emission, etc. may be controlled.

In this paper, the performance values, for example the indicated thermal efficiency, were calculated stepwise based on Wiebe's combustion function, and the relationships between the combustion rate and each performance value were investigated. In addition, the harmonics of engine noise were considered to be related to that of the cylinder pressure variation, hence the harmonics of cylinder pressure variation were calculated for the estimation of noise emission.

As a result of the calculations, it was clarified that when the combustion rate was controlled to a desired free value with a low compression ratio, the performances, i.e. indicated thermal efficiency, maximum pressure, combustion noise etc., could be improved simultaneously.

Thus, a theoretical guide to improving the performances of a low compression ratio diesel engine was obtained.

---

1. Introduction

. In a diesel engine, if the burning rate can be controlled at will, the control of the thermal efficiency, maximum pressure, maximum rate of pressure rise, combustion noise and NOx emission which is largely related to the maximum combustion temperature, may be possible.

· In this paper, by assuming that the burning rate obeys Wiebe's function[1], the relationship between the burning rate and performances, and that between the thermal efficiency, the compression ratio etc. were examined. Concerning the combustion noise, as previously reported in our first report[2], a correlation seems to exist between the harmonics of the cylinder pressure variation and that of the noise emission at the same engine speed. Thus, the relationships between the burning rate and the harmonics of cylinder pressure variation were examined in order to determine the relation between the combustibility and the noise emission qualitatively.

Besides, the theoretical thermal efficiency of the Otto cycle is considered to drop according as the compression ratio decreases, but the net thermal effi-

ciency was hardly varied by the compression ratio as already reported in our second report[3]. On the other hand, the problematic points, that is, the combustion noise, NOx emission in exhaust gas etc. have not been improved as yet.

In this paper, by examining the relation between the burning rate and various performances, a theoretical guide to improving the performances of a low compression ratio diesel engine was obtained.

2. Process of calculations

Considering the heat balance in the cylinder, the next equation holds:

$$C_{PI}T_IG_I' + C_{PE}T_EG_E' + Q_B' - Q_C'$$
$$- (GC_vT)' + APV' \quad \cdots\cdots\cdots\cdots\cdots\cdots (1)$$

where, the mark ′ represents the differential against the crank angle θ,

C_{PE}, C_{PI}: specific heats at constant pressure of the gas flowing out and into the cylinder respectively,

G_E', G_I': weights of gas flowing out and into the cylinder respectively during a crank angle kg/°CA,

Q_B': heat released by the combustion (burning rate) kcal/°CA,

Q_C': heat loss to the combustion chamber wall kcal/°CA,

G: weight of gas in the cylinder kg,

C_v: specific heat at constant volume of gas in cylinder kcal/kg°K,

A: heat equivalent of the work kcal/kg m,

P: gas pressure in the cylinder kg/m^2,

V': volume variation in the cylinder m^3/°CA.

From the relation between Eq. (1) and the state equation of gas $PV=GRT$, the next equation with respect to P holds:

$$P'+\kappa P\frac{V'}{V}=\frac{\kappa-1}{AV}(Q_B'-Q_C'+C_{PI}T_IG_I'$$

$$+C_{PE}T_EG_E') \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots(2)$$

By solving this equation, the cylinder pressure variation $P(\theta)$ can be obtained. In this case, J. Reisacher's equation[4] which was a function of temperature T and excess air factor λ, was used as the ratio of specific heats in Eq. (2):

$T\leqq700°K$

$K=1.448\ 84-0.159\ 0(T/1\ 000)+0.054\ 4(T/1\ 000)^2$
$-0.006\ 53(T/1\ 000)^3-0.037\ 45/\lambda-0.003\ 65/\lambda$
$(T/1\ 000)$

$T>700°K$

$K=1.383\ 81+0.151\ 83(T/1\ 000)-0.388\ 4(T/1\ 000)^2$
$+0.1813(T/1\ 000)^3-0.031\ 25/\lambda-0.012\ 5/\lambda(T/1\ 000)$

The mean gas temperature in the cylinder is expressed by the next equation,

$$T=\frac{PV}{\delta GR} \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots(3)$$

In Eq. (3), $\delta=1+1/\lambda L_0$ and $R=29.27-0.14/\lambda$[7].

The excess air factor λ was calculated from the released heat Q_B until an arbitrary crank angle, and λ was obtained from the equation $\lambda=(G_0H_0/Q_BL_0)$. Where, G_0: weight of new charge in cylinder before combustion,

H_0: lower calorific value of fuel,

L_0: stoichiometric air.

The cylinder volume $V=(\pi/4)d^2Y(\theta)+V_n/(\varepsilon-1)$ where,

$$Y(\theta)\fallingdotseq r\{1-\cos\theta+r/4l(1-\cos2\theta)\} \cdots\cdots\cdots(4)$$

d: cylinder bore,

ε: compression ratio,

r: crank radius,

V_h: stroke volume,

l: length of connecting rod.

The transferred heat Q_C' between gas and the combustion chamber wall was calculated using next equation,

$$Q_C'=\alpha(T-T_w)Ft'+0.362\{(T/100)^4$$

$$-(T_w/100^4)\}Ft' \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots(5)$$

where, α: heat transfer coefficient,

T_w: mean wall temperature of the combustion chamber,

F: heat-transfer area,

t: time,

$F=\pi/2d^2+\pi d[Y(\theta)+C]$

C: piston top clearance.

Among many expressions already reported about

α, Woschni's equation[5] was used in this paper, that is,

$$\alpha=256\ d^{-0.214}(PC_m)^{0.186}T^{-0.515}\ \text{kcal/m}^2°\text{Chr}$$

where, C_m: mean piston speed.

Further, G_E' during the exhaust stroke was calculated by the following equation.

$$G_E'=\mu_f\phi_E\sqrt{2g\frac{P}{v}}t' \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots(6)$$

where, μ: flow coefficient which is assumed to be 0.8 irrespective of the valve configuration and the valve lift,

f: flow area of the valve,

g: gravity acceleration,

v: specific volume of gas.

ϕ_E takes the following value, that is,

$$\frac{P_E}{P}>\left(\frac{2}{\kappa+1}\right)^{\kappa/(\kappa-1)}$$

$$\phi_E=\sqrt{\frac{\kappa}{\kappa-1}\left\{\left(\frac{P_E}{P}\right)^{2/\kappa}-\left(\frac{P_E}{P}\right)^{(\kappa+1)/\kappa}\right\}}$$

$$\frac{P_E}{P}\leqq\left(\frac{2}{\kappa+1}\right)^{\kappa/(\kappa-1)}$$

$$\phi_E=\left(\frac{2}{\kappa+1}\right)^{1/(\kappa-1)}\sqrt{\frac{\kappa}{\kappa+1}}$$

where, P_E: pressure in the exhaust manifold, being assumed at atmospheric pressure.

Many papers are already reported about the burning rate. In this paper, Wiebe's function[1] of the combustion was used.

Namely,

$$x=1-\exp\{-6.9(\theta/\theta_z)^{M+1}\} \cdots\cdots\cdots\cdots\cdots(7)$$

where, x: ratio of burned fuel,

θ: crank angle,

θ_z: combustion duration,

M: characteristic value of the combustion.

If the combustion efficiency is assumed to be 100%, the total heat released equals to G_0H_0/λ_0L_0. Then, the released heat Q_B is calculated as follows:

$$Q_B=\frac{G_0H_0}{\lambda_0L_0}\{1-e^{-6.9(\theta/\theta_z)M+1}\}$$

$$\therefore\ Q_B'=6.9\frac{G_0H_0}{\lambda_0L_0}(M+1)\left(\frac{\theta}{\theta_z}\right)^M$$

$$\times\frac{1}{\theta_z}e^{-6.9(\theta/\theta_z)M+1} \cdots\cdots\cdots\cdots\cdots\cdots\cdots\cdots(8)$$

The cylinder pressure variation $P(\theta)$ is desired from Eq. (2) considering the above conditions. Further, the indicated thermal efficiency η_i is calculated as follows,

$$\eta_i=\frac{A\lambda_0L_0}{G_0H_0}\oint P(\theta)dV \cdots\cdots\cdots\cdots\cdots\cdots(9)$$

Using Eqs. (2) and (9), the cylinder pressure variation P was calculated stepwise from the beginning of the compression stroke until the cylinder pressure became P_E at the exhaust stroke. And the cylinder pressure was assumed to be constant in the over-lap

period and suction stroke.

In order to examine the frequency spectrum of cylinder pressure variation, Fourier's expansion of $P(\theta)$ was obtained as follows;

$$P(\theta) = \frac{1}{2}a_0 + \sum_n \left(a_n \cos\frac{n}{2}\theta + b_n \sin\frac{n}{2}\theta\right) \cdots\cdots(10)$$

where,

$$\left.\begin{array}{l} a_n = \dfrac{1}{2\pi} \displaystyle\int_0^{4\pi} P(\theta) \cos\dfrac{n}{2}d\theta \\[3mm] b_n = \dfrac{1}{2\pi} \displaystyle\int_0^{4\pi} P(\theta) \sin\dfrac{n}{2}d\theta \end{array}\right\} \cdots\cdots\cdots\cdots(11)$$

As shown in Eq. (11), 1/2 order harmonics of engine revolution is a fundamental order in a four cycle engine. In case when S_n represents each frequency component, the relation of $S_n = \sqrt{a_n^2 + b_n^2}$ is obtained. Further, when S_n is converted to the sound pressure level, it may be referred to the cylinder pressure level i.e. *CPL*.

$$(CPL)_n = 20 \log(S_n/2.886\ 1) + 200 \quad (dB) \cdots\cdots(12)$$

The calculation of Eq. (11) was conducted by Simpson's numerical integration. Then, the precision of the integral became a problematic point when the order n became large. The interval of the integration was chosen at $1/10°$CA in crank angle. Consequently, even in the case of calculating the 360th order harmonics, a tolerable precision might be obtained by means of 10 fold integral per cycle.

3. Results of calculation and discussions

The calculation was conducted by using the dimensions of a test engine, namely $d=110$ mm, $r=75$ mm, $l=300$ mm, charging efficiency $\eta_d=90\%$ and the combustion chamber wall temperature $T_w=473°K$. Figure 1 shows the behavior of Q_B in case of $\lambda_0=1.5$ and combustion duration $\theta_z=120°$CA
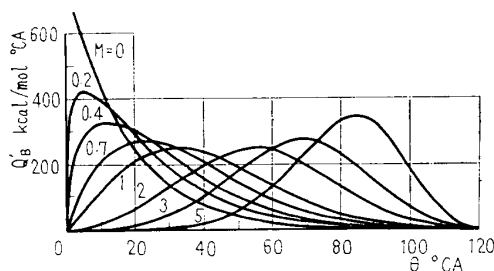
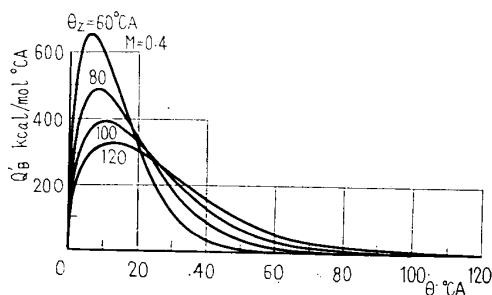Fig. 1   Effects of M on Q_B' $(\theta_z=120°$CA$)$

Fig. 2   Effects of θ_z on Q_B' $(M=0.4)$

using Wiebe's burning rate. In addition, the calculation below was conducted at $\lambda_0=1.5$ with an engine speed of $n=3\ 000$ rpm. As shown in this figure, as M decreases, a large quantity of heat release is evolved at the ignition point or thereabouts. Besides, the value of M was said to be $0.1\sim1$ in diesel combustion and $1\sim3$ in gasoline combustion. The burning rate is affected by combustion duration as well as the characteristic value of combustion M as shown in Eq. (8).

The effects of Q_B' by Q_z were also examined. Figure 2 shows the behavior of Q_B' in case of $60°$, $80°$, $100°$ and $120°$ in θ_z at $M=0.4$. And Fig. 3 shows Q_B for various θ_z at $M=1$. In each case of Fig. 2 and Fig. 3, the maximum burning rate increases with the shortening of θ_z, but large differences in burning rate when M was changed, were shown by the burning rate at the beginning of the combustion and the maximum burning rate.

As stated above, various patterns of burning rate are obtainable by changing M and θ_z. Examples of the calculated cylinder pressure $P(\theta)$ using Eq. (2) and Q_B' stated above, are shown in Fig. 4 and Fig. 5.

The ignition timing is changed in each case and Fig. 4 shows a case where a relatively rapid combustion occurs namely $M=0.2$ and Fig. 5 shows one where a smooth combustion occurs, namely $M=1$.

3·1   Effect of the ignition timing on performances

Figure 6 shows the relationships between ignition timing and performances with $M=0.2$ and $\theta_z=$
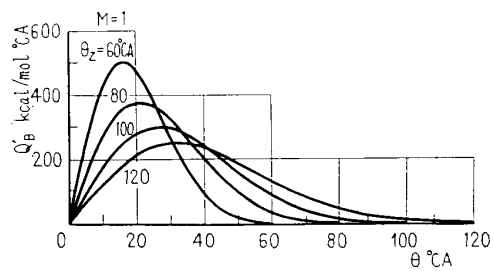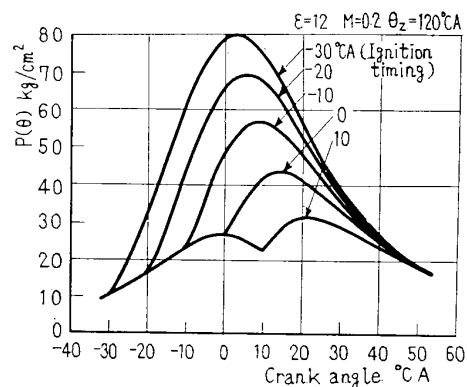
Fig. 3   Effects of θ_z on Q_B' $(M=1)$

Fig. 4 $P(\theta)$ in case of varied ignition timing $(M=0.2)$

80°CA. And the parameter given is the compression ratio.

The variation of the thermal efficiency due to ignition timing becomes smaller as the compression ratio decreases, and the ignition timing where maximum η_i is attained is not varied by the compression ratio. When the ignition timing is advanced from the optimum timing for η_i, the maximum combustion temperature T_{max} increases as the compression
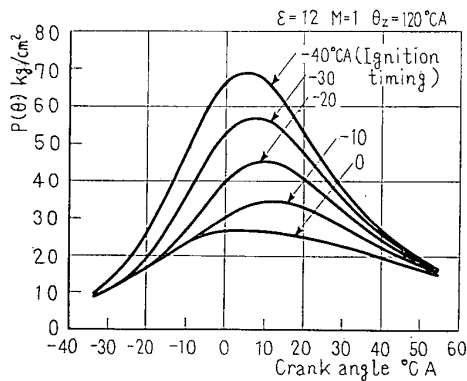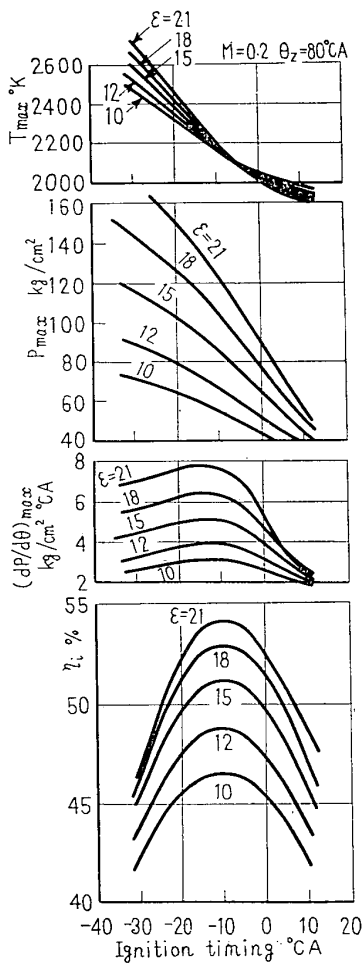

Fig. 5 $P(\theta)$ in case of varied ignition timing ($M=1$)

Fig. 6 Relation between performances and ignition timing (Effect of compression ratio)

ratio increases and as the ignition timing is advanced; but when the ignition timing is set back from that point, T_{max} decreases more rapidly as the compression ratio increases. Namely, in the case of a large retardation of ignition timing, an increased expansion work resulting from the increased compression ratio seems to reduce the maximum temperature. Figure 7 shows the relations between the performances and the ignition timing using combustion duration θ_z as the parameter and using the compression ratio at 15. From this figure, at the same ignition timing, η_i, $(dp/d\theta)_{max}$, P_{max} and T_{max} increase as θ_z decreases. And the optimum ignition timing for η_i tends to approach the top dead center as θ_z decreases. T_{max} and P_{max} increase as θ_z decreases at the same ignition timing, but the variation of T_{max} and P_{max} due to various θ_z decreases to a lesser extent in optimum ignition timing for η_i.

The frequency components of cylinder pressure variation with varied ignition timing are shown in Fig. 8 and Fig. 9. Figure 8 shows the CPL of the cylinder pressure variation given in Fig. 4, and Fig. 9 does that in Fig. 5.

It can be seen that the CPL seems to increase as the ignition timing is advanced in a lower frequency range from 500Hz (10th order of engine revolution), but in a higher frequency range than this, a definite result is not obtained. As for CPL, H. List has
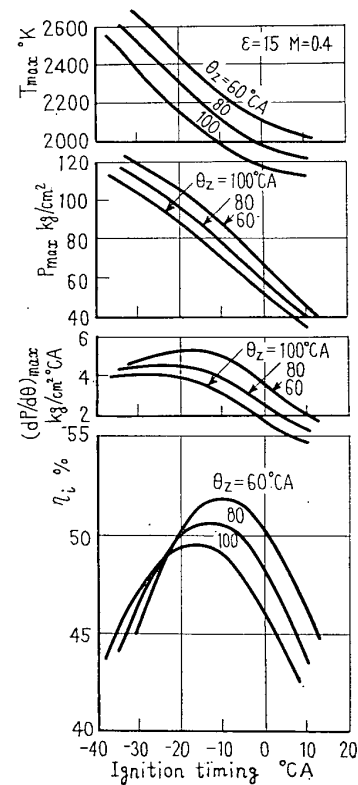

Fig. 7 Relation between performances and ignition timing (Effect of θ_z)

reported[6] that *CPL* is correlated to the values of

$$\int pdt, \quad \frac{\Delta p}{\omega}, \quad \left(\frac{dp}{dt}\right)_{max}\frac{1}{\omega^2} \quad \text{and} \quad \left(\frac{d^2p}{dt^2}\right)_{max}\frac{1}{\omega^3}, \quad \text{as}$$

the frequency becomes high. Assuming that P_{max} corresponds to $\int pdt$ and $\left(\dfrac{dp}{d\theta}\right)_{max}$ corresponds to $\left(\dfrac{dp}{dt}\right)_{max}$ in this calculation, the tendency of *CPL* in a lower frequency range shown in Fig. 8 and Fig. 9 might be adequate. In Fig. 8, *M* equals to 0.2 and in Fig. 9 *M* equals to 1, and *CPL* shown in Fig. 9 is a result of frequency analysis for a smooth $P(\theta)$. The differences between these cases are the reduction of *CPL* in a higher frequency range as *M* becomes large. In addition, the relation between *M* and *CPL* will be described later in detail. Figure 10 shows the relation of $(dp/d\theta)_{max}$, P_{max} and *CPL* in each frequency against the ignition timing. *CPL* in a lower frequency range than 500Hz seems to show the same tendency as P_{max} and $(dp/d\theta)_{max}$, but in a higher frequency range some frequency components tend to decrease as the ignition timing is advanced. In general, a method wherein the ignition timing is kept late is used to reduce the engine noise. But Fig. 10 indicates the possibility that the noise is not always reduced by later ignition timing, for example in the case where resonance frequency exists at 2500Hz
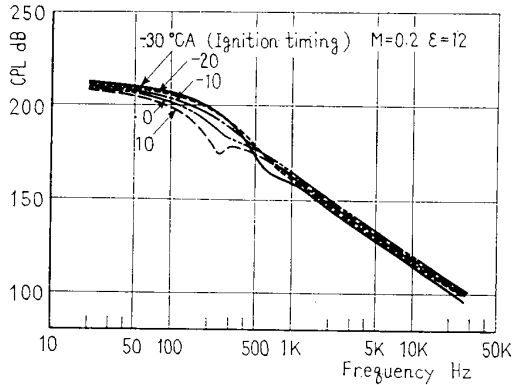
in engine structure.

3·2 Effects of *M* on engine performances

Engine performances are affected by varying the burning rate through *M*. The relationships between the ignition timing and the indicated thermal efficiency η_i at compression ratio 15 are shown in Fig. 11. As the value of *M* increases from this figure, it is noted that the optimum ignition timing for the indicated thermal efficiency tends to advance. This tendency, though it is assumed from the behavior of $Q_B{}'$ in Fig. 1, results from the retarded gravity center of the burning rate curve as the value of *M* increases.

In the case of varying the value of *M*, the performances at the optimum ignition timing for the thermal efficiency are shown in Fig. 12.

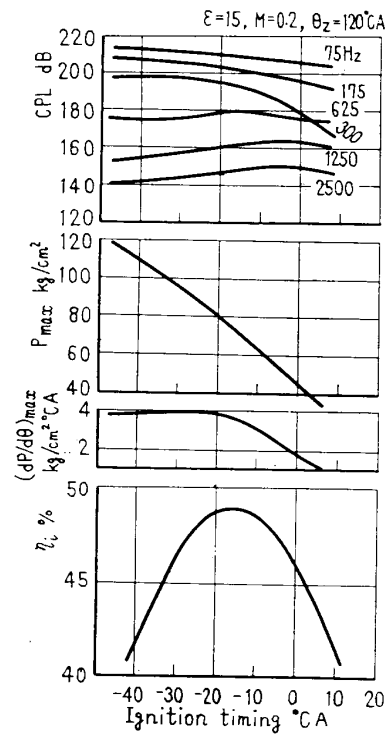In addition, Fig. 12 gives the results with the combustion duration $\theta_z = 120°\text{CA}$ and the compression

Fig. 8 Relation between ignition timing and cylinder pressure spectrum ($M=0.2$)
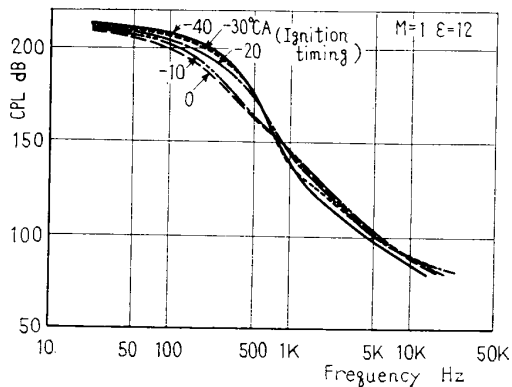
Fig. 9 Ignition timing and the cylinder pressure spectrum ($M=1$)
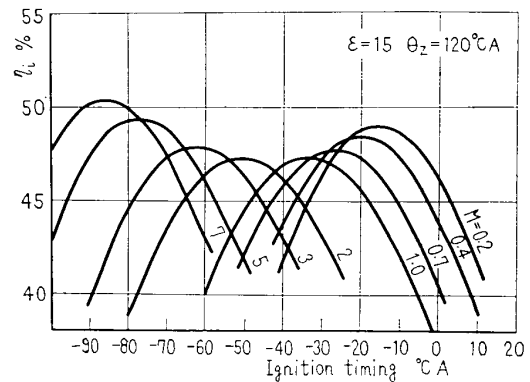

Fig. 10 Ignition timing and CPL

Fig. 11 Relation between η_i and igition timing in case of varied *M*

ratio $\varepsilon = 15$.

From this figure, the following are derived; (1) η_i indicates its minimum value in the range of $M = 1\sim2$. Thus, η_i increases by either increasing or reducing this value of M. (2) The maximum rate of pressure rise $(dp/d\theta)_{max}$ rapidly increases as the value of M decreases in the range of $M<1$, but in a range of $M>1$ the minimum pressure rise is almost unchanged. (3) The maximum pressure P_{max} is not changed largely with the variation of M. Namely, even if the burning rate varies, P_{max} is hardly changed in the case where the combustion occurs at optimum ignition timing for the indicated thermal efficiency.

This is because the constant ratio of heat must be released before top dead center to obtain maximum thermal efficiency. This ratio of heat is about 40 percent of the total released heat in this calculation.

(4) The lower frequency components of cylinder

Fig. 12 Performances and M ($\varepsilon = 15$)
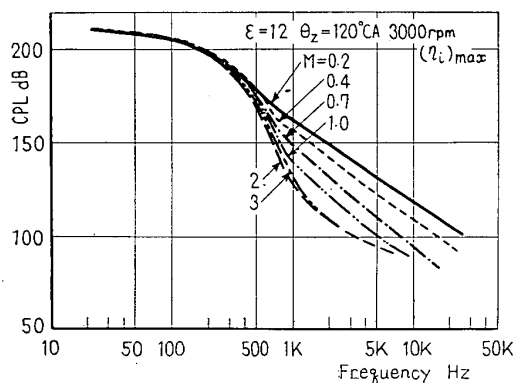
Fig. 13 Cylinder pressure spectrum with varied M (optimum ignition timing for η_i)

pressure level (CPL) are not affected by the value of M. But the higher frequency component increases suddenly as the value of M decreases from 1. In addition, in the range of $M>1$, CPL is very low and is not affected by M.

Consequently, in this calculation, a low cylinder pressure level with a comparatively good indicated thermal efficiency can be obtained by selecting M, for example $M=6$. Figure 13 shows CPL against the frequency for various M ranging from 0.2 to 3. From this figure, it is observed that as M increases, the lower frequency components are not varied, while the higher frequency components tend to reduce intensively.

3·3 Effects of combustion duration θ_z

The combustion duration θ_z as well as M affects the form of the burning rate curve. Figure 14 and Fig. 15 show the relationships between the combustion duration and the engine performances for $M=0.4$ and 1 respectively.

In every case, the ignition timing is optimized so as to attain the best indicated thermal efficiency. From these figures, the following matters are derived, namely, (1) η_i improves linearly as the combustion duration θ_z is shortened. (2) The maximum rate of pressure rise increases with the decrease in θ_z, and its extent seems to increase as the value of M becomes
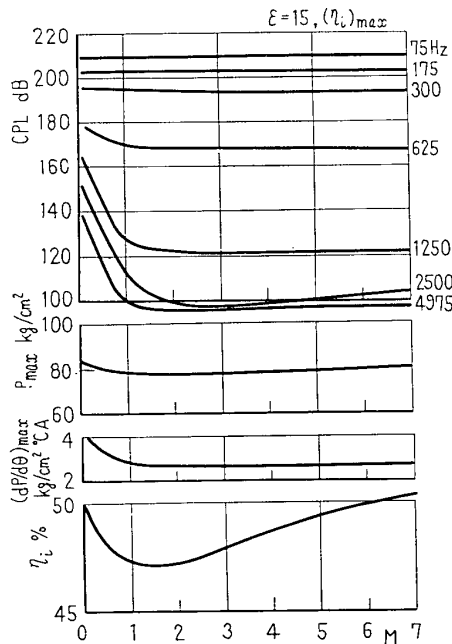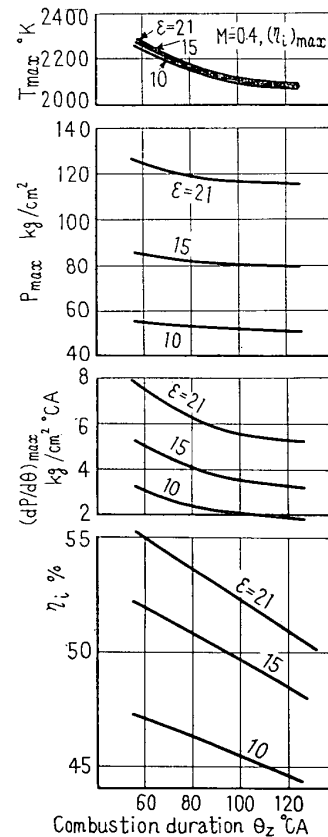

Fig. 14 Relation between performances and combustion duration ($M=0.4$)

smaller and the compression ratio higher. (3) The maximum pressure increases with a shortened θ_z.

And its extent is larger at higher compression ratios but the increment is relatively small. (4) The maximum temperature of combustion increases as the combustion duration elongates. The increase of maximum temperature owing to the increase of the compression ratio is very small at the optimum ignition timing for the indicated thermal efficiency using the same combustion duration θ_z. And this tendency appears in each case of combustion duration θ_z. In addition, the increase of maximum temperature owing to the decrease of M, namely the occurrence of sudden combustion, tends to be intensive.

Figure 16 shows the relations of η_i, $(dp/d\theta)_{max}$ and CPL against the combustion duration θ_z using M as the parameter. As is seen from this figure, as the combustion duration θ_z shortens, the cylinder pressure level is not varied in a lower frequency range, but tends to increase in a higher frequency range. On the contrary, when the combustion duration θ_z is constant, an increase in M decreases the CPL but decreases η_i in the range of $M<1$. Then, if M increases and simultaneously θ_z decreases, it becomes possible for the CPL to decrease and η_i to be improved. A detailed description will be made later.
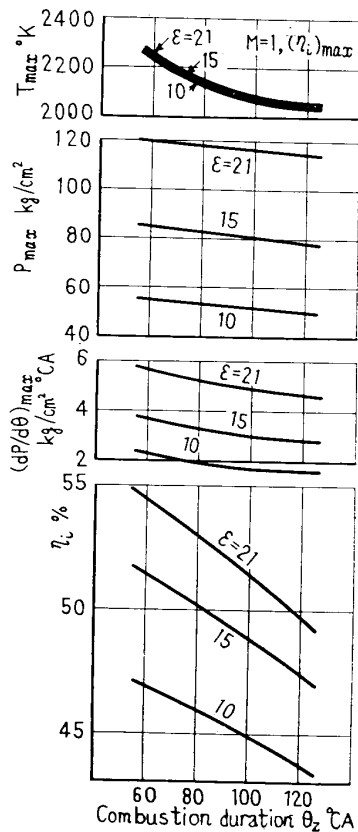
From the above results, the initial burning rate is considered to affect CPL a great deal. For example, the burning rates at $M=0.4$, $\theta_z=100°$CA in Fig. 2 and $M=1$, $\theta_z=80°$CA in Fig. 3 are similar to each other except for the lower initial rate of burning in the latter case. As for the performances of these two cases, from Fig. 16, η_i is larger and frequency components decrease in the case of $M=1$. Then, the relations of the burning characteristic value M, the combustion duration θ_z and the maximum pressure rise $(dp/d\theta)_{max}$ against the indicated thermal efficiency are shown in Fig. 17 as an example. In this case, the performances are calculated at the compression ratio 15 and the optimum ignition timing for η_i. According to this figure, as M and θ_z decrease, η_i and $(dp/d\theta)_{max}$ tend to increase. If M and θ_z can be controlled at will, it is possible to decrease $(dp/d\theta)_{max}$ with η_i maintained constant, or it is possible to increase η_i with a reduction of $(dp/d\theta)_{max}$ simultane-
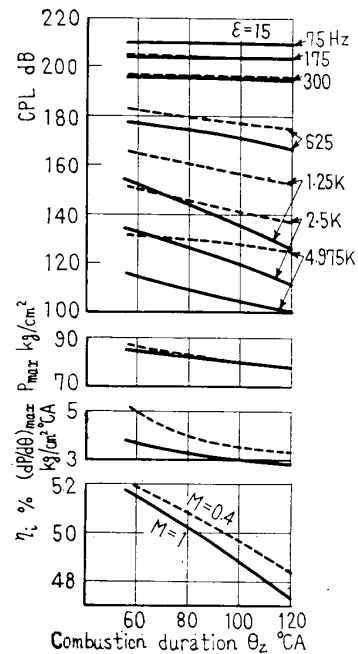

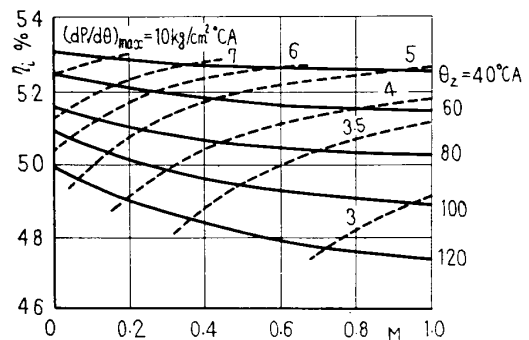Fig. 16   Relation between cylinder pressure spectrum and combustion duration

Fig. 15   Relation between performances and combustion duration ($M=1$)

Fig. 17   Relation of M, θ_z and $(dr/d\theta)_{max}$ against η_i (compression ratio $\varepsilon=15$, ignition timing is optimum on η_i)

ously. For example, if the burning rate at $M=0.2$, $\theta_z=120°$CA can be changed to the burning rate at $M=0.8$, $\theta_z=80°$CA, η_i is improved from 49 percent to 50.5 percent and $(dp/d\theta)_{max}$ from 4 kg/cm²°CA to 3.5 kg/cm²°CA. Figure 18 shows the relation of M, θ_z and CPL against η_i. In this case, the cylinder pressure level at 1 250Hz is selected, namely a 25th order component of engine revolution. In this figure, the tendency of CPL curve is similar to that of $(dp/d\theta)_{max}$ curve as shown in Fig. 17. As stated above, if the burning rate is controlled, it is possible to improve η_i and simultaneously decrease the CPL.

3·4 Effects of the engine speed

The foregoing results, including CPL and some engine performances, are calculated under the condition of engine speed at 3 000 rpm.

In this section, the effects of the engine speed on the cylinder pressure level which seems to be intensively affected by the engine speed, are examined. Figure 19 shows an example of the cylinder pressure level under condition of engine speed at 1 000, 2 000 and 3 000 rpm.

From this figure, it can be seen that each frequency component increases as the engine speed rises. Then, the noise excited by the cylinder pressure level seems to increase as the engine speed increases. In a lower frequency range than 2 kHz, the cylinder pressure
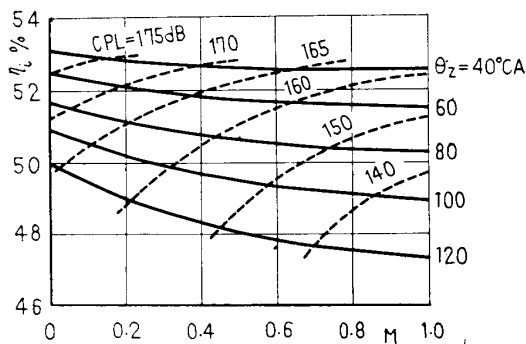

Fig. 18   Relation of M, θ_z and cylinder pressure level against η_i (compression ratio $\varepsilon=15$, ignition timing is optimum on η_i, frequency of CPL is 1 250 Hz namely 25th order of engine revolution)
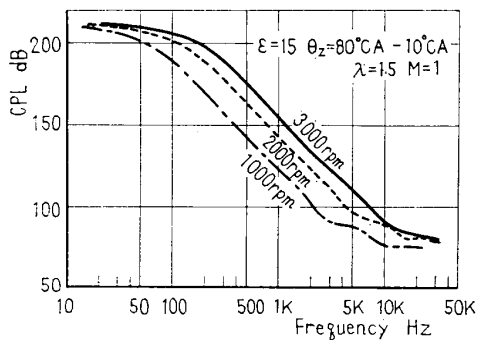

Fig. 19   Cylinder pressure spectrum in case of varied engine speed

level of the same order components of engine revolution is almost the same; namely, although the cylinder pressure level of the same order is the same when compared among different engine speeds, only its frequency is considered to increase with an increasing engine speed.

3·5 Relationship between the compression ratio and the performances

Before discussing the compression ratio, the relation between the maximum pressure and the thermal efficiency must be mentioned.

Figure 20 illustrates η_i and maximum rate of pressure rise $(dp/d\theta)_{max}$ against the maximum pressure using the compression ratio as the parameter.

In this case, the burning rate curve is assumed to be constant, namely $M=1$ and $\theta_z=120°$CA. From this figure, the maximum pressure to give the maximum value of η_i is recognized at a certain compression ratio, and its maximum pressure becomes high with an increased compression ratio. In addition, although M and θ_z were varied, the maximum pressure at which η_i took its maximum value did not vary largely. $(dp/d\theta)_{max}$ increased as the maximum pressure rose.

The relations between the compression ratio and η_i using maximum pressure as a parameter are shown in Fig. 21 and Fig. 22. And the conditions of Fig. 21 are $M=0.2$, $\theta_z=120°$CA and those of Fig. 22 are $M=1$, $\theta_z=120°$CA. In these figures, in the case where the maximum pressure is kept at constant, the optimum compression ratio for η_i is recognized and this compression ratio increases as the maximum pressure rises.
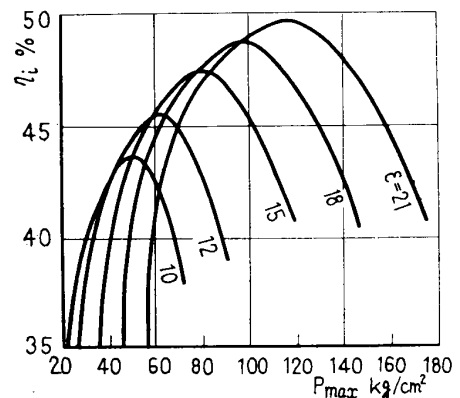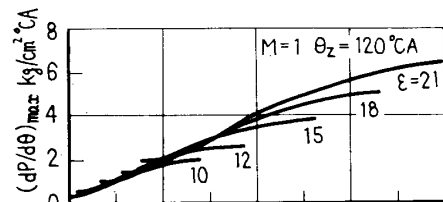

Fig. 20   Relation of η_i and $(dp/d\theta)_{max}$ against maximum pressure

The optimum compression ratio for η_i at constant maximum pressure tends to decrease as the value of M increases, namely as the curve of the burning rate becomes smoother. In general, by shortening the combustion duration, the thermal efficiency is said to increase. Then, Fig. 23 and Fig. 24 show the relation between the compression ratio and the indicated thermal efficiency by varying the combustion duration.

In this case, the value of M in Fig. 23 is 0.4 and that of M in Fig. 24 is 1. The parameters in these figures are the combustion duration θ_z and the maximum pressure rise $(dp/d\theta)_{max}$. From these figures, the following is derived; (1) if the combution duration can be shortened as the compression ratio decreases, the indicated thermal efficiency remains unchanged under conditions of constant $(dp/d\theta)_{max}$.

(2) In the case of decreasing the compression ratio and maintaining the same indicated thermal efficiency, the rate of pressure rise tends largely to decrease at large values of M. But, in a case where the value of M is small namely the burning rate suddenly increases, the extent of reduction of the rate of pressure rise is of a lesser extent or is considered to increase as compared with the case where the value of M is small. It is apparent from the results stated above that, if M and θ_z can be controlled the compression ratio may be reduced by maintaining the rate of pressure rise which seems to affect the noise emission while the indicated thermal efficiency is not varied.

4. Experimental results

The experimental results related to the calculated ones are given below. Figure 25 gives an example of the rate of heat release which was obtained by changing the combustion chamber configuration and using a direct injection type diesel engine at $\lambda_0 = 1.5$ with the compression ratio at 13.6.

As seen in this figure, the initial heat release suddenly increases and the combustion duration is shortened in the order of (A), (B) and (C).
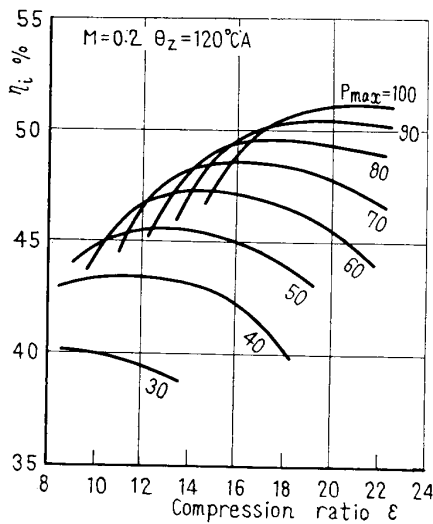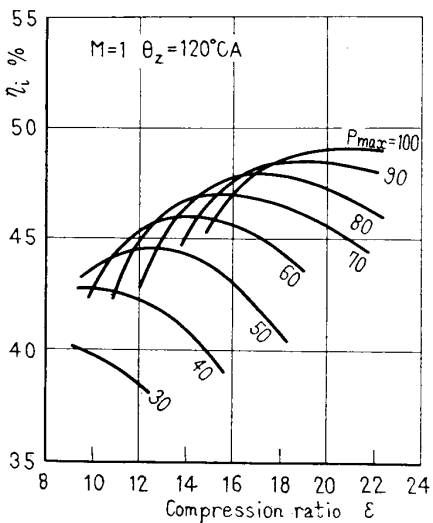

Fig. 21 η_i and compression ratio $(M=0.2)$

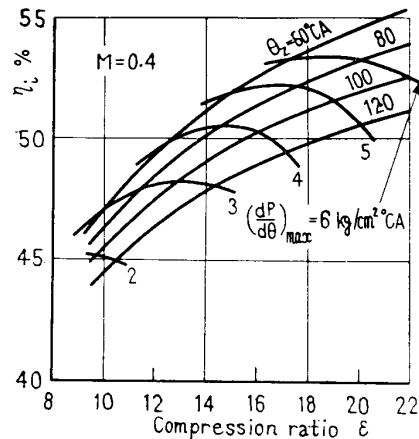Fig. 22 η_i and compression ratio $(M=1)$

Fig. 23   Relation between compression ratio and η_i in case of varied combustion duration $(M=0.4)$
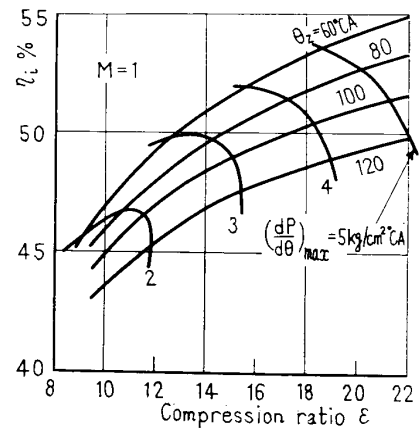

Fig. 24   Relation between compression ratio and η_i in case of varied combustion duration $(M=1)$

In each case, the injection timing is so selected as to give an optimum on the net specific fuel consumption. Figure 26 shows the plot of the net specific fuel consumption against the ignition timing in the cases of (A), (B) and (C). From this figure, the optimum ignition timing of the specific fuel consumption is advanced in the order of (C), (B) and (A), and simultaneously the extent of varying the specific fuel consumption against the ignition timing tends to decrease. These results seem to justify the relation between η_i and the ignition timing shown in Fig. 7. In addition, the form of heat release rate is actually affected by the ignition timing, and the mechanical loss is considered to exert its effect when the net specific fuel consumption is compared with the indicated specific consumption. In Fig. 26, the specific fuel consumption becomes worse in the order of (A) to (C). In this respect, the specific fuel consumption is considered to be worse with respect to the degree of constant volume when the heat release occurs suddenly, namely in the order of (C), (B) and (A). However, the experiment was conducted in order to shorten the combustion duration by decreasing the entrance diameter of the combustion chamber, so that the cooling loss increased. And this causes the specific fuel consumption to increase especially in the case of (C). Figure 27 shows the results of the frequency analysis of the cylinder pressure variation in the case of varying the heat release rate
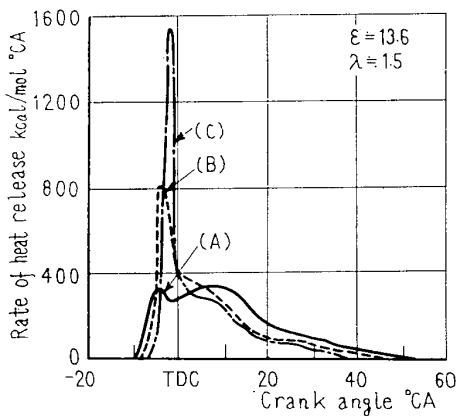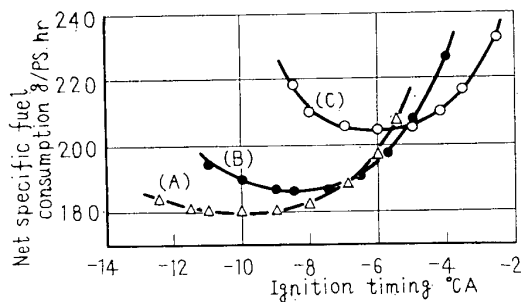
as shown in Fig. 25. In this figure, the cylinder pressure level decreases in the order of (C), (B) and (A).

Namely, as the initial heat release in the smoother *CPL* decreases, the noise in the over all range is reduced. In this figure, a large component is noticeable in the range of 2kHz.

Figure 28 shows the results of the frequency analysis of the cylinder pressure variation when the ignition timing is varied. In this figure, as the ignition timing is retarded, *CPL* decreases in all ranges, and this tendency increases in a lower range than 200Hz especially when the ignition timing is late. These tendencies are in agreement with the cases in Fig. 8 and Fig. 9. In addition, these are considered to depend on the sudden variation of the pressure gradient after the top dead center, and a large frequency component is recognized at 300Hz when the ignition timing is very late. Figure 29 shows the results of the frequency analysis when the engine speed was varied. In this case, an example was selected at a higher load range where the variation of the heat release was comparatively small. In this figure, as the engine speed rises, the frequency components especially in the lower range than 630 Hz seem to increase. In addition, at the frequency range above 630 Hz, a frequency component unrelated

Fig. 25 Comparison of heat release curve

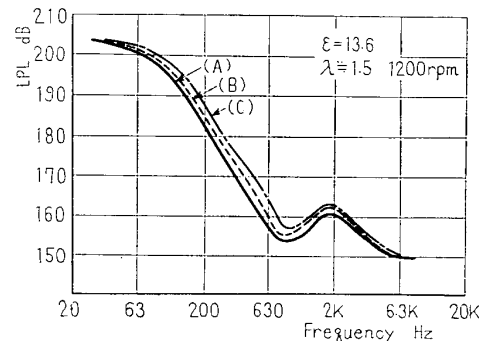Fig. 26 Relation of the specific fuel consumption against ignition timing

Fig. 27 Frequency analysis of cylinder pressure in case of varied heat release curve
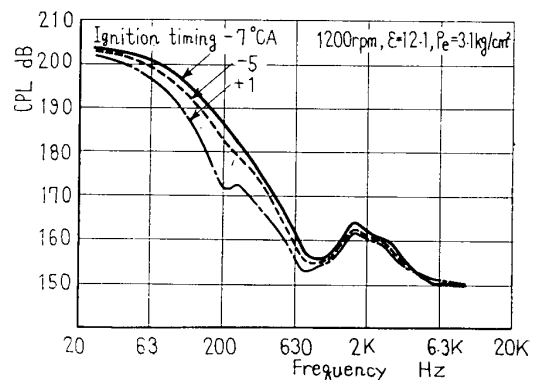

Fig. 28 Frequency analysis of cylinder pressure in case of varied ignition timing
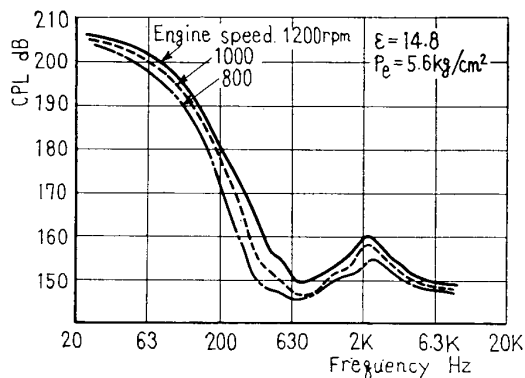
Fig. 29   Frequency analysis of cylinder pressure in case of varied engine speed

with the engine revolution appears. The frequency of the same order components of engine revolution seems to increase because of the rise of the engine speed as shown in Fig. 29 in the same manner as in Fig. 19.

As stated above, raising the engine speed results in an increase in the cylinder pressure level, which is considered an important cause to increase the noise. The combustion actually occurs in a more complicated pattern than Wiebe's burning rate. And it is necessary to take into account the disagreement of theoretical heat transfer and practical heat transfer in a test engine.

5. Conclusions

The relationships of the indicated thermal efficiency, the maximum pressure, the maximum temperature, the frequency components of the cylinder pressure variation, etc. against the burning rate, were clarified from the results of these calculations. Hence a guide to the realization of a low compression ratio diesel engine was obtained. The conclusions of this paper are as follows;

(1) If the burning rate can be controlled, for example by shortening the combustion duration, the reduction of the compression ratio may be obtaind by keeping the indicated thermal efficiency and the maximum pressure rise constant.

(2) At the optimum ignition timing for the indicated thermal efficiency, the extent of varying the maximum pressure against the variation of the burning rate curve is very small and the maximum combustion temperature increases to some extent due to the increase of the compression ratio.

But the maximum combustion temperature increases to a comparatively large extent with a decrease of the combustion characteristics number M and the combustion duration.

(3) Within a range of $0 < M < 1$ the indicated thermal efficiency decreases as the value of M increases but the frequency components of the cylinder pressure variation especially the higher frequency components decrease. In a range of $M > 1$, as the value of M increases, the indicated thermal efficiency is improved but the frequency components are hardly varied, namely they are kept at a low level.

(4) With the shortening of the combustion duration, the indicated thermal efficiency is improved linearly and simultaneously the higher frequency component of the cylinder pressure variation increases. But the latter can be largely decreased by the increase of M. Thus, it becomes possible to improve the indicated thermal efficiency and simultaneously decrease the higher frequency components, as a result the value of M increases and the combustion duration θ_z shortens.

(5) The larger the value of M becomes and the longer the combustion duration θ_z is, the more the optimum ignition timing on the indicated thermal efficiency tends to advance. However, the optimum ignition timing is not affected by the compression ratio. When the ignition timing is late, the frequency components decreases in a lower frequency range, but the chance remains of an increase in a higher frequency range.

References

(1) Sitkei, G., *Kraftstoffaufbereitung und Verbrennung bei Dieselmotoren* (1964), Springer-Verlag.
(2) Fukazawa, S., et al., *Trans. Japan Soc. Mech. Engrs.* (in Japanese), Vol. 37, No. 307 (1966), p. 1410.
(3) Miyamoto, N., et al., *Trans. Japan Soc. Mech. Engrs.* (in Japanese), Vol. 38, No. 307 (1967), p. 623.
(4) Reisacher, J., *MTZ*, Vol. 27, No. 10 (1966), p. 411.
(5) Woschni. G., *MTZ*, Vol. 26, No. 4 (1964-5), p. 128.
(6) JSME, *The Development of Modern Diesel Engine* lectured by H. List, (1963).
(7) JSME, *The Handbook of Mechanical Engineering*, (1946).

Discussion

K. Terada (Nagoya University):

(1) As one of the practical applications of this study, the optimum conditions of M, θ_z and the ignition timing are obtainable by calculation, and the engine can be operated under these conditions.

In this case, what is the basis on which to judge the similarity between the measured heat release curves and Wiebe's function?

(2) In case Wiebe's function does not approximate to the measured heat release curve, the defferences of P_{max}, $(dp/d\theta)_{max}$ etc. tend to become larger. In that case, do the authors intend to use functions other than Wiebe's function?

T. SAITO (Waseda University):

(3) It seems to be difficult to apply these results directly to the practical combustion of the engine. But, these studies are significant as providing the basic data for explaining qualitatively the relationships between the complex combustion progress and the performances.

And the results obtained in this paper can also be used to explain the Meurer combustion system well-known for the lower noise level and good performances. If the authors considered this, may I have the author's view on this point? In addition, NOx concentration in exhaust gas seems to depend simultaneously on the maximum combustion temperature T_{max} and the duration of a high temperature being maintained. The duration of a temperature above 1800°K being maintained, for example, tends to elongate as M or θ_z becomes large. May I have the author's calculated results in this respect?

T. SUZUKI (Hino Motor Co.):

(4) We have cases where, even when P_{max} and $dp/d\theta$ were almost the same, a difference in CPL at 1 000 Hz or thereabouts was recognized in the case of larger premixed combustion and the engine noise seemed to be affected by this. Append.-Fig. 5 shows the heat release curve calculated from the indicator diagrams and Append.-Fig. 6 shows the CPL spectrum.

(a) May I have the author's view about the possibility of that case and the author's experience of the calculation?

(b) Did the author examine the effects of the premixed combustion ratio, by modifying the burning rate?

(5) A large component appears at 2 kHz or thereabouts in the author's experiments (Figure 27~29). In our experiment, two components appear at 2.5 kHz and 4 kHz as shown in Append.-Fig. 6. We made an attempt at calculating the frequency across the cylinder using the equation $f_0 = C/2D$ and the two frequencies calculated by using the sonic velocity at the compression temperature; and the combustion temperature agreed generally with the experimental ones. Besides, the resonance at the compression temperature was a randum occurrence. If this big component depends on the resonance of the combustion chamber space, the compression temperature in a low compression ratio engine seems to decrease, and the low compression ratio engine seems to be a disadvantage in regard to the noise emission because the resonance frequency reduces to 1 000 Hz in an engine for an automobile. And, in the case where there is a variance in chamber configuration and combustion conditions, the resonance seems to appear at a certain diameter of the cylinder cavity (namely, D stated in the above equation is defined as cavity diameter).

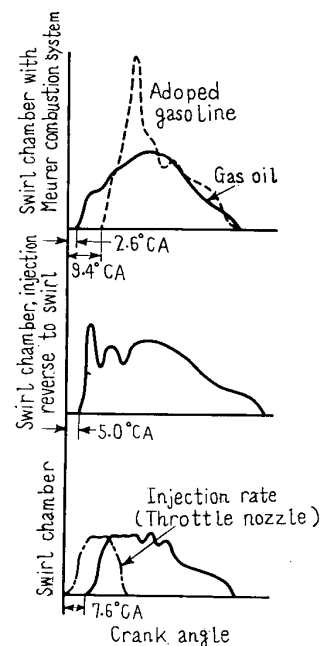(a) May I have the author's calculations in those respects stated above?

(b) What is the author's opinion about my view that a low compression ratio engine will be disadvantageous especially in regard to the noise for
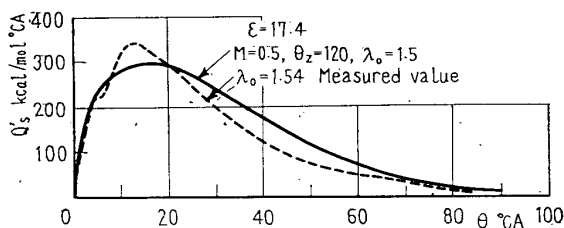
Append.-Fig. 1
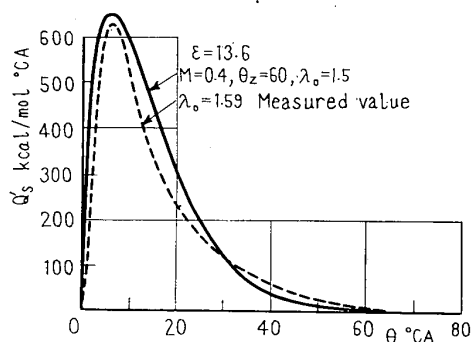
Append.-Fig. 2

Append.-Fig. 3

the above mentioned reason?

Y. Matsumoto (Komatsu Co.):

(6) The authors use Woschni's equation in heat transfer. May we use the heat transfer coefficient in turbulent flow $N_u = 0.023 R_e{}^{0.8} P_r{}^{0.4}$ in calculating the suction air quantity?

(7) The radiated heat quantity in the combustion chamber is reported to be very small even in combustion. What is the ratio of the radiated heat in these calculations? And, isn't it necessary to consider heat transfer in the exhaust manifold in calculations? What equation may we use if heat transfer is taken into consideration?

(8) May I have the calculation results in case of varied mean temperatures of the combustion chamber?

Authors' closure

(1) The standards of similarity are considered to be simultaneous agreement between the rate of heat release (by the mean square method etc.) and the degree of constant volume.

(2) A case where Wiebe's function does not

Append.-Fig. 4

Append.-Fig. 5 Combustion process at test condition of A and B

agree very well with the practical heat release appears in the experiment. In this case, functions other than Wiebe's function (for example, the kind of study reported by Prof. Matsuoka, Tokyo Institude of Technology) should be used.

In addition, examples of comparison between Wiebe's function and practical heat release are shown in Append.-Figs. 1 and 2. Fine similarity as against Wiebe's function can be obtained in the case where the tendency of two-stage combustion is smaller.

(3) The rate of heat release obtained by the Meurer combustion system is shown in append.-Fig. 3. The initial heat release becomes very smooth in comparison with that obtained by other combustion systems, and this seems to correspond to a large value of M. Besides, the Meurer combustion system has a disadvantage that the position of the maximum heat release retired comparatively. The duration of a high temperature being maintained shortens as M increases with the ignition timing maintained at constant.

On the other hand, at the ignition timing giving an optimum indicated thermal efficiency, it tends to elongate as M increases.

(4) (a) As Mr. Suzuki pointed out, it can be considered that a difference in CPL appears even when the values of P_{max} and $(dp/d\theta)_{max}$ are the same. For example, the higher frequency components seem to be contained in the pressure vibration occurring after a sudden pressure rise and in sudden pressure variation. The swirl ratio varied in Mr. Suzuki's experiments. In this case, $(dQ/d\theta)_{max}$ seems to increase as the swirl ratio becomes large. Consequently, the amplitude of the pressure vibration excited by $(dQ/d\theta)_{max}$ is considered to increase.
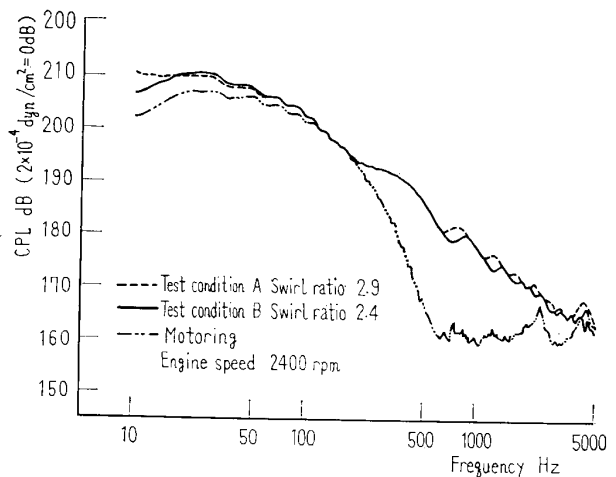
(4) (b) Calculations in these respects will be conducted in the future. But, the burning rate, in case the part of premixed burning is made to vary, is simulated by varying the value of M.
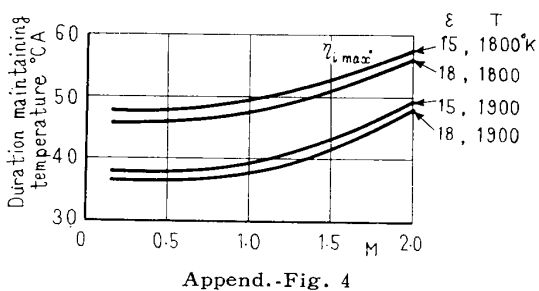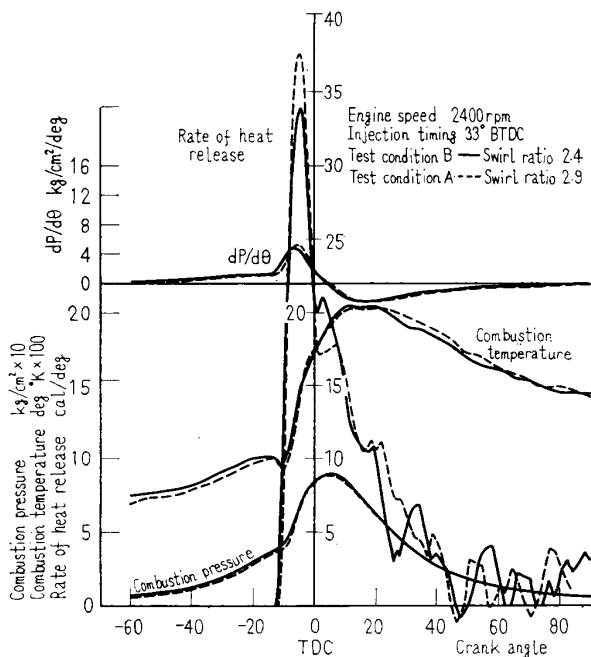
Append.-Fig. 6 Frequency analysis of combustion pressure at conditions of A and B

(5) (a) The results of the calculation on f_0 in our engine are $f_0 = 3095$ Hz(1200°K) and 3790 Hz (1800°K). And these frequency components take a high level in the higher frequency. Besides, as a 2~3 kHz vibration appear after the pressure rise in the indicator diagrams, the component of 2 kHz or thereabout seems to depend on this vibration.

But, it is not quite certain that the vibration in the indicator diagrams depends on the resonance in the combustion chamber. The authors will carry out examinations of these points in the future.

(5) (b) Mr. Suzuki's line of thought is adequate. But assuming that the gas vibration is excited by the combustion in the cylinder, *CPL* and the noise emission may be reduced by a decrease of $dp/d\theta$.

(6) Woschni's equation is based on the heat transfer of turbulent flow in the tube and its coefficients are obtained by the experiments using the tested engine. Thus, in this case, Woschni's equation seems to be preferable to that of turbulent flow in the tube.

(7) The quantity of thermal radiation differs under the conditions, and occupies some 2~3 percent of the total thermal radiation calculated by Nusselt's equation. Thus, the quantity of thermal radiation is relatively small as Mr. Matsumoto pointed out. The heat transfer in an exhaust manifold is not considered because the authors focus their attention on the state change of the cylinder gas. But, the equation of turbulent flow in the tube seems to be preferable when the necessity of its calculations occurs.

(8) Assuming that the burning rate is not affected by the wall temperature, the increase by 100°C of the mean chamber temperature results in an increase by 0.3 percent of the indicated thermal efficiency.