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A Study on Low Compression Ratio Diesel Engines*

(5th Report, Effect of Intake Air Temperature on Burning Rate
and Engine Performances)

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The control of the burning rate is very important in a common diesel engine, and is also an important problem especially in a lower compression ratio diesel engine for further improving the noise emission, thermal efficiency etc.

In this paper, for the purpose of controlling the initial burning rate, it was tried to investigate experimentally and theoretically the relationship between the intake air temperature and the initial burning rate.

And the effects of the intake air temperature on engine performances, startability and NO_x emission, were also examined.

As to the burning rate, the initial maximum burning rate almost the same as the experimental one could be obtained by calculations using the evaporating theory of droplets and the thermal reaction theory. And the possibility and the extent of controlling the initial burning rate by an increase of the intake air temperature, were made clear.

1. Introduction

One of the factors which has the greatest influence on the performance of a diesel engine is the burning rate in the cylinder.

Especially in a low compression ratio diesel engine, the control of the burning rate is important for the improvements of thermal efficiency, NO_x emission and the pressure rise related to noise emission.

In this paper, the relationship between the intake air temperature and the initial burning rate was investigated with the express purpose of controlling the initial burning rate. The effect of intake air temperature on the engine performance, startability and NO_x emission were also examined in this paper.

Regarding the burning rate, the calculation results of burning rate using the evaporation theory of droplets and thermal reaction theory, show almost the same initial maximum burning rate as that obtained experimentally. Thus, the possibility of controlling the burning rate by intake air temperature was confirmed experimentally and theoretically.

As a result of experiments, a considerable improvement of the noise emission was accomplished by a decrease of the initial maximum burning rate resulting from an increase of intake air temperature. The relation between the intake air temperature and startability was also confirmed, which may be considered to hold for a low com-

pression ratio diesel engine.

The experimental results will be described in detail below.

2. Experimental apparatus
and procedures

The test engine was a four-cycle, direct injection type diesel engine, and its main specifications are as shown in Table 1.

Engine type	Water cooled, single cylinder and vertical type
Bore/Stroke	110 ϕ mm/150 mm
Rated horsepower	10 PS/1200 rpm
Injection pump	Bosch NPE 1A70B
Injection nozzle	Bosch DN4SD123
Nozzle opening pressure	200 kg/cm ²

Table 1 Specifications of test engine

The operating conditions of the test engine were as follows; revolution 1200rpm, cooling water temperature $70 \pm 5^\circ\text{C}$, lubricating oil temperature $35 \sim 40^\circ\text{C}$.

The combustion chamber configurations were of the BIP type and the compression ratios were adjusted by varying the combustion chamber volume in the piston with the piston top clearance maintained at constant value. The intake air temperature was also adjusted with a 2-kw electric heater inserted into the intake manifold.

In addition, the automotive gas oil (specific gravity $0.825/15^\circ\text{C}$, cetane number 55) was used in test.

The measurements of NO and NO_2 densities were carried out by using an NDIR gas analyzer, and the sampling position of test gas was in the exhaust manifold, at a distance of 100mm from the exhaust valve.

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An indicating sound pressure meter (B and K 2203/1613) was used for measuring the noise emission and the measuring position was on the opposite side of the fly wheel, 1m apart from the engine center and at 1m height from floor. In addition, the calculation of the rate of heat release was done using the same computing programs as already reported in our previous report (1).

3. Experimental results and discussions

3.1 Effect of intake air temperature on engine performances

The relationships between the compression ratio and the intake air temperature were first examined, and the results are shown in Figure 1. With an increase in

Fig. 1 Relations between intake air temperature and performances

the intake air temperature, the ignition lag was shortened, and both the maximum pressure rise and the engine noise were improved. These tendencies became intense when the compression ratio was lower. Namely, extensive improvement of engine noise resulting from an increase in intake air temperature were made especially at lower compression ratio, while at the compression ratios higher than 17.4, no reduction in engine noise was obtained. A large reduction of engine noise at low compression ratios seems to depend mainly on the decrease of maximum pressure rise, as may be surmised from the close correlation between the maximum pressure rise and the noise emission as reported previously (1).

In addition, since the ignition lag is considered to affect the pressure rise to a large extent, the phenomenon of ignition lag when the intake air temperature was varied, will be discussed further below.

In this paper, a mathematical relation on the ignition lag reported by Livengood (2), namely $\int_0^{\sigma} \frac{1}{\sigma} dt = 1$, was adopted for cal-

culating the ignition lag in a cylinder where the pressure and temperature were varied with time.

Fig. 2 Effect of intake air temperature on ignition lag

Regarding these calculated results, the full line in Figure 2 shows the calculated values and some points in this figure show the experimental results. In the figure, the ordinate represents the ratio of the ignition lag at all intake air temperatures against that at 20°C. As seen in the figure, the experimental values are in comparatively good agreement with the calculated ones. Thus, when the intake air temperature was varied, it is possible to predict the ignition lag to a certain extent by using Livengood's relation on ignition for each intake air temperature and compression ratio.

On the other hand, with an increase in the intake air temperature, the specific fuel consumption tended to worsen at each compression ratio.

Fig. 3 Relations between excess air factor and specific fuel consumption

In this respect, the relationships between the net mean effective pressure and the specific fuel consumption, and between excess air factor and the indicated specific fuel consumption are illustrated in Figure 3.

As seen in this figure, the net specific fuel consumption becomes worse at a

higher load range, and then the maximum power output, namely the smoke limited power output, tends to decrease in accordance with the intake air temperature increase.

The indicated mean effective pressure P_i is given by the following equation;

$$P_i = \frac{\gamma_n}{A} \left(\frac{H_u}{L_0} \right) \frac{\eta_d \eta_i}{\lambda_0}$$

where, γ_n : the specific weight of air at standard conditions,

A : the heat equivalent of work,

H_u : the calorific value of fuel,

L_0 : the stoichiometric air,

λ_0 : the excess air factor,

η_d : the charging efficiency,

η_i : the indicated thermal efficiency.

$\frac{\gamma_n}{A} \left(\frac{H_u}{L_0} \right)$ is generally considered to be a constant value; thus the following equation may be obtained;

$$P_i = \text{const.} \frac{\eta_d \eta_i}{\lambda_0}$$

By further examining the relations between the excess air factor and the indicated specific fuel consumption, the degree of worsening of the indicated specific fuel consumption was ascertained to be very small, although a slight tendency of worsening was recognized. Thus, the reduction of P_i seems to depend mainly on the decrease of η_d in the case when the intake air temperature increase, because η_i , as mentioned above, does not seem to be largely affected by the intake air temperature.

The net thermal efficiency inevitably decreases owing to the mechanical efficiency η_m caused by a reduction of P_i .

Consequently, the aggravation of the maximum power output and the net specific fuel consumption in the heavier load range, arising from an increase of the intake air temperature, seemed to depend mainly on the reduction of the indicated mean effective pressure caused by a decrease of the charging efficiency, and further on the reduction of mechanical efficiency.

Fig. 4 Effects of intake air temperature on rate of heat release

As shown in Figure 4, the forms of heat release show a considerable change, especially, at lower compression ratios with an increase of air temperature. Namely, the rate of the initial maximum heat release decreases, and the duration of heat release

becomes prolonged simultaneously. Accordingly, as shown in Figure 5, the degree of constant volume decreases especially at lower compression ratios with an increase in intake air temperature.

Fig. 5 Relations between intake air temperature and degree of constant volume

And this reduction of the degree of constant volume is considered to be one of the reasons for aggravation of the net specific fuel consumption.

In our previous experiments, the relations between the intake air temperature and engine performances were investigated by using a pre-chamber type diesel engine.

And, the results described in this section agree partially with those obtained previously (3).

3.2 Effects of intake air temperature on burning rate

A change of the heat release due to an increase of the intake air temperature affects the engine performances to a large extent as already reported above. In this section, we present a comparison made between the theoretical burning rate and the experimental rate.

Prior to calculations of the burning rate, the following assumptions were made ;

(1) The thermal reaction occurs simply between two materials, namely the fuel and oxygen.

(2) The fuel and oxygen are distributed uniformly in a combustion chamber.

(3) The fuel taking part in the thermal reaction is completely evaporated.

(4) The fuel spray is a gathering of single droplets of a certain diameter.

In the combustion process, a change of fuel concentration is represented by the following equation.

$$\frac{dC_f}{dz} = -\phi C_f C_o + \frac{dC_{fe}}{dz} \quad (1)$$

Where, ϕ : the constant of reaction rate,

C_f : the fuel concentration,

C_o : the oxygen concentration,

z : the time, and

C_{fe} : the concentration of evaporated fuel.

Defining σ and x as the ratios of the evaporated fuel and the burnt fuel to the entire amount of injected fuel respectively;

$$\sigma = (Q_e(z)/B_0), \quad x = B(z)/B_0$$

Thus, the next equation holds,

$$C_f = \frac{(\sigma - x)B_0}{M_f V}, \quad C_0 = \frac{0.232(\lambda_0 - x)L_0 B_0}{M_0 V} \quad \dots (2)$$

Where, B_0 : the whole amount of fuel injected,

$Q_v(z)$: the amount of evaporated fuel at time z ,

$B(z)$: the amount of burnt fuel at time z ,

M_f : the molecular weight of fuel,

M_0 : the molecular weight of oxygen,

V : the volume of combustion chamber,

L_0 : the stoichiometric air.

On the other hand, ϕ is assumed to be expressed by the relation of Arrhenius in consideration to the effects of higher pressure (4) in a combustion chamber.

$$\phi = k \left(\frac{P_c}{P_0} \right)^2 \exp \left(-\frac{E}{RT_c} \right) \quad \dots (3)$$

And, C_f is represented by the following equation,

$$C_f = \frac{\sigma B_0}{M_f V} \quad \dots (4)$$

From equations (2), (3) and (4), the equation (1) is represented as follows;

$$\frac{dx}{dz} = \frac{0.232 L_0 B_0 k}{M_0} \times \left(\frac{P_c}{P_0} \right)^2 \exp \left(-\frac{E}{RT_c} \right) \frac{(\sigma - x)(\lambda_0 - x)}{V} \quad \dots (5)$$

where, E : the activation energy,

R : the gas constant,

k : the coefficient of reaction,

P_0 : the standard pressure,

λ_0 : the excess air factor for the entire amount of fuel injected,

P_c : the pressure in the combustion chamber,

T_c : the temperature in the combustion chamber.

The burning rate $B_0 H_u (dx/dz)$ can be calculated by solving equation (5).

On the other hand, σ is a variable affected by the amount of evaporated fuel, hence σ may be obtained by calculating the evaporating process of a single droplet as mentioned below.

The amount of evaporated fuel dG at the surface of droplets is expressed as follows;

$$dG = \beta f_d (P_s - P_{s0}) dt \quad \dots (6)$$

where, β : the coefficient of mass convection,

f_d : the surface area of droplet,

P_s : the saturated pressure of fuel vapor,

P_{s0} : the pressure of fuel vapor around a droplet.

Putting $\beta = D_p S_h / D$, $f_d = \pi D^2$ and $P_{s0} = 0$ into equation (6), the next equation is obtained;

$$dG = \pi D D_p P_s S_h dt \quad \dots (7)$$

where, D_p : the coefficient of diffusion,

S_h : Sherwood number,

D : the diameter of a droplet.

From the relation $dG = -d \left(\frac{\pi}{6} D^3 \gamma_f \right)$, the next equation holds;

$$D^2 = D_0^2 - \int_0^t Y dt \quad \dots (8)$$

$$\text{where, } Y = \frac{4 D_p P_s S_h}{\gamma_f}$$

From the equations (7) and (8), one has,

$$dG = \frac{\pi}{4} \gamma_f Y \sqrt{D_0^2 - \int_0^t Y dt} dt$$

Since the initial weight of a droplet $G_0 = (\pi/6) D_0^3 \gamma_f$, the next equation is obtained,

$$d \left(\frac{G}{G_0} \right) = \frac{3}{2} \frac{Y}{D_0^3} \sqrt{D_0^2 - \int_0^t Y dt} dt$$

That is,

$$\rho(\tau) \equiv \frac{G}{G_0} = \frac{3}{2} \frac{1}{D_0^3} \int_0^\tau Y \sqrt{D_0^2 - \int_0^t Y dt} dt \quad \dots (9)$$

Putting $\rho(\tau) = 1$ when $\tau \geq t$, in this case, t is obtained from the relation of $\int_0^t Y dt = D_0^2$.

Defining $IR(T)$ as the injection rate, $Q_v(z)$ may be obtained by $\tau = z - T$, which is derived from the equation (9) and the relation shown in Figure 6.

$$Q_v(z) = \int_0^z IR(T) \rho(z - T) dT$$

That is,

$$\sigma(z) = \frac{1}{B_0} \int_0^z IR(T) \rho(z - T) dT \quad \dots (10)$$

In order to calculate the burning rate, it is assumed that the pressure and the temperature in the combustion chamber were uniform in the cylinder and varied under the first law of thermodynamics.

Fig. 6 Relations between $IR(T)$ and $Q_v(z)$

Using equation (5) and (10) with the first law of thermodynamics, one has,

$$B_0 H_u \frac{dx}{dz} - \frac{dQ_c}{dz} = \frac{d}{dz} (G C_v T_c) + A P_c \frac{dV_c}{dz} \quad \dots (11)$$

Putting the relation $P_c V_c = G C_v R T_c$ into the equation (11), one has,

$$\frac{dP_c}{dz} + k P_c \frac{dV_c}{dz} = \frac{\kappa - 1}{A V_c} \left(B_0 H_u \frac{dx}{dz} - \frac{dQ_c}{dz} \right) \quad \dots (12)$$

where, Q_c represents the cooling heat, which was calculated by using the coefficient of heat convection reported by Woschni (5).

Further, the next relations were used in the calculations;

$$S_h = 2(1 + 0.3 S_c^{1/3} R_c^{1/2})$$

$$D_p = 0.121 \times 10^{-4} \left(\frac{T_c}{273} \right) \frac{1}{P_c} \text{ m/h}$$

$$P_c = 6.0 \times 10^7 \exp \left(\frac{-4.15 \times 10^3}{T_f} \right) \text{ kg/m}^2$$

The surface temperature of a droplet was calculated under the assumption that the temperature inside a droplet was uniform.

$$C_{df} G_f dT_f = \pi D^2 \alpha (T_c - T_f) dz - l_i dG$$

where, C_{df} : the specific heat of a fuel droplet,

l_i : the amount of heat needed for evaporation.

From the relation $\alpha = \frac{\lambda_d}{D} N_u$, one has,

$$C_{df} G_f \frac{dT_f}{dz} = \pi \lambda_d D N_u (T_c - T_f) - l_i \frac{dG}{dz} \quad \dots (13)$$

where, $N_u = 2(1 + 0.3 P_r^{1/3} R_e^{1/2})$

In calculation, it was assumed that the relative velocity between the fuel droplets and the air was zero, that is $Re=0$.

And, the results reported by Sitkei (6), were used in calculations as C_f and l_i .

Further, k and E were assumed to be $2.5 \times 10^{11} \text{ m}^3/\text{kmols}$ and $4 \times 10^4 \text{ kcal/kmol}$ respectively. The numerical calculations by using equation (5) were carried out under consideration of equations (10), (12) and (13).

The calculated results are shown in Figures 7 and 8.

Fig. 7 Effects of intake air temperature on burning rate

Fig. 8 Effects of intake air temperature on burning rate

In these cases, calculations were carried out under the conditions of $D_0 = 30 \mu$, the initial temperature of droplet $T_f = 50^\circ \text{C}$, $\lambda_0 = 2.5$ and under the assumption that the engine dimensions needed for the calculations are the same as those of the experimentally tested engine.

On the other hand, the rate of fuel injection $IR(T)$ was measured by a long tube method reported by W.Bosch. And the weight of gas in cylinder G_a was the experimental values under all experimental conditions where the intake air temperature was varied.

As a result of these calculations, the two-stage combustion which was generally observed in a diesel combustion, was clearly obtainable at the burning rate calculated.

Further, as the intake air temperature increased, the ignition period was advanced and the initial maximum burning rate decreased as shown by a broken line in Figures 7 and 8. The behavior of the initial maximum burning rate calculated in this paper is in good agreement with that in the experimental results which were obtained at the time when the intake air temperature was varied.

Besides, since the initial maximum burning rate, as already reported in our previous report, largely affects the engine noise, the reduction of the initial maximum burning rate is a most important problem, especially for the lower compression ratio diesel engine.

Fig. 9 Experimental and calculated values of initial maximum burning rate

A comparison between the initial maximum burning rates obtained by experiments and those calculated in this section is made in Figure 9. In this figure, the abscissa shows the initial maximum burning rate calculated and the ordinate shows the experimental values.

And it may be seen from this figure that the calculated values tend to be larger than the experimental ones in the case where the initial diameter of droplet D_0 is assumed to be comparatively small. On the other hand, in a range of the initial maximum burning rates lower than $1000 \text{ kcal/kmol}^\circ \text{CA}$, which is the most common range of burning rate in a diesel engine, the calculated results of the initial maximum burning rate agree with the experimental results for each compression ratio under the assumption of $D_0 = 30 \mu$. Thus, it is clear that the initial maximum burning rate could be predicted to some extent by using the method mentioned

in this section.

Under the general conditions of engine running, the relation $dP/d\theta \propto dQ/d\theta$ is considered to hold between the initial maximum burning rate $dQ/d\theta$ and the maximum pressure rise $dP/d\theta$, so that the behavior of $dP/d\theta$ arising from an increase of intake air temperature may be explained by the initial maximum burning rates obtained from the calculations mentioned in this section.

A comparison between the experimental pressure rise and the initial maximum burning rates calculated is made in Figure 10.

Fig. 10 Effects of intake air temperature on initial maximum burning rate and maximum pressure rise

It is seen from this figure that a change of pressure rise arising from an increase of intake air temperature exhibits the same tendency as the initial maximum burning rates calculated. That is, the experimental behavior of the pressure rise owing to an increase of intake air temperature, can be also explained clearly by the burning rate calculated in this section.

3.3 Effects of intake air temperature on NO emission and startability

As stated in the previous section, the burning rate can be controlled for practical purposes to some extent by increasing the intake air temperature. Further, the effects of intake air temperature on NO emission and startability were examined in this section. The density of NO emission tends to increase with an increase in the intake air temperature, but it decreases at higher load range as shown in Figure 11.

On the other hand, the density of NO₂ emission decreases at each load range according as the intake air temperature increases. However, the density of NO₂ emission was still quite lean compared with that of NO emission, hence no description regarding NO₂ emission will be made here.

The data shown in Figure 11 were rearranged again by using the excess air factor for the purpose of a detailed examination of the experimental tendency of NO density, and its results are shown in Figure 12.

According to this figure, the maximum density of NO emission is obtained in the range of excess air factors approximately between 1.3 and 1.5 regardless of the load ranges.

Consequently, the density of NO emis-

sion in the case when intake air temperature increases seem to depend largely on the excess air factor. However, it can be seen from Figure 12 that the density of NO emission increases as the load range decreases even in the case when the excess air factor is the same. This phenomenon seems to take place follows; the combustion temperature became higher in the lower load range with an increase of the intake air temperature under the same excess air factor.

One of the important problems in a lower compression diesel engine is its startability. Thus, startability was examined in a case where the intake air temperature was varied at the moment of starting.

Fig. 11 Relations between intake air temperature and NO_x

Fig. 12 Relations between excess air factor and NO

Fig. 13 Effects of intake air temperature on number of motoring cycle for starting

Figure 13 shows the relationships between the intake air temperature and the number of motoring cycle for starting under conditions where the engine body temperature was set at 5°C and the engine revolution was set at 200 rpm. In this case, the number of motoring cycles for starting means the number of cycles required for the continuous combustion at the beginning of starting.

Thus, with a smaller number of motoring cycles for starting the startability improves. It can be ascertained from this figure that the startability tends to improve as the compression ratio and the intake air temperature become higher. Further exami-

Fig. 14 Relations between number of motoring cycles for starting and temperature at compression end
 nation was carried out for investigating the relations between the calculated temperature at the compression end and the number of motoring cycles for starting. A simple relationship between them was found without regard to the compression ratios as shown in Figure 14.

Consequently, it was made clear that the same startability as that of a higher compression ratio diesel engines could be obtained at a lower compression ratio by keeping the temperature at the compression end the same as that of a higher compression ratio diesel engine.

4. Conclusions

Elucidation of the effects of intake air temperature on the burning rates and the engine performances provided a guide for improving the performances of a low compression ratio diesel engine. The results of this paper may be summarized as follows:

(1) The maximum power output and the minimum specific fuel consumption dropped mainly with a decrease of the charging efficiency arising from an increase of the intake air temperature.

However, the ignition lag was shortened and the noise emission was improved to a considerable extent by an increase of the intake air temperature.

(2) The change of ignition lag caused by an increase of the intake air temperature could be explained clearly by the relation reported by Livengood.

(3) The initial maximum burning rate which has a close correlation with the engine noise was found to be controlled to a certain extent by an increase of the intake air tem-

perature. On the other hand, the burning rates could be obtained qualitatively by calculations using the evaporation theory of fuel droplets and thermal reaction theory.

Further, the behavior of the initial maximum burning rate and the maximum pressure rise in the case where the intake air temperature was varied, could be explained theoretically to a certain extent.

(4) With an increase of the intake air temperature, the following general tendency of NO emission was recognized; namely, an increase in NO emission was seen in the range of excess air factors larger than 1.3 or 1.5, and a decrease in a range of smaller excess air factors.

(5) In a low compression ratio diesel engine, the same startability with a higher compression ratio could be obtained by keeping the temperature at the compression end the same as the higher compression ratio.

- Appendix -

According to Livengood's relation (2), the next equation should hold regarding the ignition lag τ in time;

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

σ is an equation indicating the ignition lag under the conditions where the pressure and the temperature were held constant. And in this case, Wolfer's equation was used for σ .

$$\sigma = CP_c^{-n} \exp\left(\frac{E}{RT_c}\right) \text{ ms}$$

where, $C=0.44$, $n=1.19$, $E/R=4650^\circ\text{K}$

Assuming P_c and T_c vary adiabatically, one has,

$$P_c = P_0 \left\{ \frac{V_0}{V(t)} \right\}^{\kappa}, \quad T_c = T_0 \left\{ \frac{V_0}{V(t)} \right\}^{\kappa-1} \quad (15)$$

Putting $A = CP_0^{-n} V_0^{-n(\kappa-1)}$ and $B = -E/RT_0 V_0^{\kappa-1}$ into equation (14), the next equation holds,

$$A \int_0^{\tau} V(t)^{-n\kappa} \exp\{BV(t)^{1-\kappa}\} dt = 1 \quad (16)$$

As $V(t)$ can be obtained from the engine mechanism, the ignition lag may be calculated by numerical calculations. In this case, T_0 was determined by the pressure P_0 at the beginning of the compression stroke and air quantity which flowed into the cylinder.

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