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A Study on Ignition and Combustion of a Diesel Spray

by Means of a Rapid Compression Machine

By

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A free-piston type rapid compression machine has been developed to simulate combustion process occurring in a diesel engine. A light piston is pneumatically shot and gets hammered in a stop ring at the compression end. A satisfactory performance has been achieved by this device, permitting a near-constant elevated pressure and temperature sufficient to explore ignition of a fuel spray and the subsequent combustion. From a series of tests carried out on this machine, it has been shown that the ignition lag exhibits little or small dependence on pressure and temperature at some elevated pressures and temperatures. It has been also pointed out that in the so-called diffusive burning period, combustion proceeds with self-accelerating rate of heat release in the case when much fuel is allotted to this period, while the rate declines monotonically with time in the contrary case.

1. Introduction

In recent years, the diesel combustion has been studied intensively to meet the demand for high power and high efficiency of engine compatible with reductions of noise and of exhaust emissions. However, the combustion occurring in a diesel engine is quite a complex phenomenon in which physical and chemical processes, such as spray development, fuel-air mixing, ignition, and combustion, proceed complicatedly, all taking place in a very short time. The lack of detailed basic knowledge of them makes it almost impossible to meet the combustion system to several requirements.

In modern techniques for studying the diesel combustion it is important to measure the cylinder pressure, make high-speed photography, and conduct direct gas-sampling and analysis. These methods are quite useful, but their application to an existing engine system is usually hampered by several limitations. This arises not only from some hardware problems but also from the inability to separate a single factor from mutually dependent parameters. For the purpose of fundamental studies on the diesel combustion, it is therefore advantageous to employ a single shot burning device compared to the ordinary engine. A constant-volume bomb suits this purpose and makes it possible to simulate the real progress by injecting a fuel into a compressed air which is preheated to a desired temperature(1) ~ (3). This device is relatively

simple, but a strong heating is necessary to realize sufficient levels of pressure and temperature. For this reason, there exists a practical limitation in attainable temperature. An alternative method is to use a rapid compression machine which attains a high pressure and temperature by compressing air using a piston. Although this scheme does not require a strong heating unlike the bomb method, a special device is needed to drive and stop the piston accurately and quickly. Usually, the compression is made either hydraulically or mechanically; Pischinger's device(4) and Rogowski's device(5)~(6) are famous as the former group. A mechanical system has been employed by several other investigators; Zigan(7) and Tsuda(8) have fabricated their machines using a link mechanism, and Ogasawara(9) has employed a freely falling plane-cam to drive the piston. Since a fast action is needed, the mechanical rapid compression machine should have a relatively large dimension and a complex hardware.

In the present study, a new type rapid compression machine has been constructed to enable an investigation on basic aspects of the diesel combustion including ignition, in the same manner as in the earlier studies described above. Basically, this machine, operated pneumatically, consists of a long cylinder and a light freely moving piston. To compress air in the cylinder, the piston is pneumatically shot into a stop ring, finally attaining a constant-volume condition. This simple principle not only obviates elaborate design in the hardware but also guarantees an excellent reproducibility of the obtained results.

2. Design of the rapid compression machine

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Fig. 1 Experimental set-up of rapid compression machine and injection system

2.1 Machine body

For the purpose of studying the diesel combustion, a hot compressed air of a pressure higher than 30 atmospheres and a temperature higher than 700 K is required. Unless a sufficient piston speed is ensured, it is difficult to realize this condition owing to heat losses during compression. It is therefore recommended to make the piston speed as high as possible to minimize the losses. However, a fast compression gives rise to a difficulty in bringing the piston to a halt. In other words, absorbing a shock constitutes one of the central problems in designing this kind of device, and in fact the most modern compression machines devised to date are complicated in structure.

For this reason, it has been decided to use a very light free-piston to obtain a sufficient compression speed and at the same time to alleviate a strong impulse acting on the stopping device. This piston is shot pneumatically by a compressed air and is rammed into a tapered ring for fixing by means of a plastic deformation as well as of hooping effect. In Fig. 1 a schema of the machine is shown together with an outline of the fuel injection system. The main body consists of several cylinders connected in series each having an inner diameter of 50 mm and a length of 300 mm. A free piston is placed on the far left side of the cylinder paneled by an orifice plate and partitioned by a mylar diaphragm. A driver chamber is installed on the orifice plate and is connected to a pressure reservoir by a ball valve. To operate the device, the driver chamber and the pressure reservoir are first filled with nitrogen whose pressures are held below and above the burst pressure of the diaphragm (around 12 kgf/cm²) respectively. Once the valve is opened, the diaphragm bursts and the piston is driven. The piston accelerates toward the combustion chamber, but it is decelerated by a pressure drop behind due to a throttling effect of the orifice

Fig. 2 Combustion chamber and piston

that has a 36% area of the cross-section of the cylinder.

In Fig. 2 the details of the piston and the combustion chamber are shown. The piston is composed of a demountable head and a skirt both made of an aluminum alloy, total weight being only 110 grams. On account of the damage at each run, the piston head has to be replaced by a new one at every experiment. Nevertheless, the piston does not secede from the stopper-ring and well seals the chamber even at a pressure higher than 100 kgf/cm², as will be exemplified later. The inner wall of the cylinder liner has been polished finely, and an O-ring fitted on the piston head serves to seal the cylinder during compression.

2.2 Combustion chamber and the injection system

The combustion chamber has a wedge form as is shown in Fig. 2 and a compression volume of 80 cm³. A fuel injection nozzle is placed in the rear side of the chamber on the axis of the cylinder in the direction opposed to the piston head. This chamber simulates the diesel combustion process in a quiescent chamber. In the present report, a Bosch DN4S1 type pintle nozzle was employed, and the fuel used was a diesel fuel of a specific weight 0.830. In order to attain a single injection at a desired

timing, an electrically controlled injection system was devised. This system consists of an electro-magnetic servo-valve and an accumulator, as is shown in Fig. 1. A pressurized fuel is stored in the accumulator at pressures around 180 kgf/cm² by using a standard Bosch fuel pump. Once the electro-hydraulic servo-valve (Shimadzu 2F-06 type) is triggered by a timing pulse, the accumulator communicates with an injection pipeline, and thus injection is performed. The pipeline has an inner diameter of 1.6 mm and a length of 650 mm. The opening pressure was selected to be 120 kgf/cm². The control signal for the servo-valve was generated upon reception of a pressure signal of the combustion chamber. This pressure signal was delayed by a preset time and formed, by using three monostable multivibrators, to give proper delay time and pulse width.

At the top of Fig. 3, the waveform controlling the servo-valve is sketched in relation to the needle lift curve; t_s and t_n are the durations of positive and negative pulses, respectively, t_d the inherent lag between the application of the pulse and the beginning of injection, and t_{inj} the injection period. In the case of a single pulse, case (a) in Fig. 3, the fuel quantity V_{inj} is controllable only in a narrow range of t_s . In addition the injection lag t_d largely changes with t_s . To remove these two drawbacks, in the case (b), a negative pulse of $t_n = 6$ ms is added to normal positive pulse. This reduces pressures in the nozzle and the pipeline quickly down to the atmospheric pressure when injection is ended, similarly to the retraction of the ordinary diesel jerk pump. In this case, the lag t_d is not largely affected by t_s , and the fuel quantity V_{inj} is almost proportional to t_s . Moreover, the valve closes very quickly. Thus this latter scheme has been adopted throughout the study.

2.3 Basic characteristics of the rapid compression machine

Fig. 3 Injection characteristics

In Fig. 4 is given an example of the records of the pressure in the combustion chamber p and of the needle lift h , attained during a fired run on the devised compression machine, under the condition of initial pressure $p_o = 1$ kgf/cm², compression ratio $\epsilon = 19.5$, and fuel quantity $V_{inj} = 0.107$ ml. Pressure inside the chamber was measured by a piezo-electric transducer (AVL 12QP-250 type) and a charge amplifier (AVL TA2/C type). The needle lift was detected photo-electrically using a light-emitting diode and a photo-diode. These signals were recorded on digital wave memories of three channels each having one kilowords (two channels are 8 bit NF Circuit Block WM-812 type, and the other 10 bit home-made one). Reproduction was conducted after each run on an X-Y recorder.

As can be seen in Fig. 4, pressure in the chamber pressure increases exponentially with time as the piston compresses the air charge, and the pressure rise is finally interrupted when the piston stops instantly. A strong impact by the piston causes a transient oscillation which is damped quickly. Simultaneously, pressure decreases gradually due to heat losses to the wall. In this case, a fuel is injected at 11 ms after the halt when the pressure oscillation is damped. After an ignition delay, the pressure rises quickly, just as in ordinary diesel combustion. The upper trace in Fig. 4 reproduces the pressure change around the ignition delay period, displayed on magnified time- and pressure-scales. This well suits studying the state of ignition and the ignition process in detail.

Before tests, basic characteristics of the machine were investigated by varying the compression ratio ϵ . Figure 5 summarizes the obtained results showing the effect of ϵ on compression pressure p_c , compression temperature T_c , and polytropic index m , for the initial cylinder pressure $p_o = 1$ kgf/cm² and the initial temperature $T_o = 300$ K. The polytropic index m was given by

$$m = \log(p_c/p_o) / \log \epsilon \quad (1)$$

The compression pressure p_c was estimated by centering the oscillatory pressure

Fig. 4 Example of pressure record

curve. Within the range of the test, m is shown to be 1.33 ± 0.01 for the compression ratio ranging from 19 to 27. This value is considered to be high judging from the fact that a small piston diameter is used. This is clearly due to high piston velocities of the order of 100 m/s in average. The compression temperature T_c was calculated, assuming a uniform temperature distribution over the chamber, by

$$T_c = T_o (p_c/p_o)^{(m-1)/m} \quad (2)$$

At the compression ratio of $\epsilon = 19$, this temperature amounts to 750 K which may suffice for an autoignition of the fuel sprayed.

3. Ignition process of the spray

3.1 Ignition delay

Experiments were first carried out on the rapid compression machine to elucidate the ignition process of a fuel spray. It has been known that the compression ignition process is featured by a rapid pressure rise due to an explosive burning of a combustible mixture which accumulates during the ignition delay

Fig. 5 T_c , p_c , and m versus compression ratio ϵ

Fig. 6 Effect of temperature on ignition delay τ_1

period. This rapid pressure rise is responsible for the so-called diesel knock in an actual engine system. Among a lot of studies conducted on ignition delay of the spray, only a few are known to be of millisecond range in which the existing engines normally operate.

In the present study, special attention is paid to this range. According to the magnified pressure curve shown in Fig. 4, in the case of a short delay, it is not easy to determine the start of ignition precisely because pressure rises very slowly. Moreover, this faint pressure change is hidden in a pressure drop caused by heat absorption of the spray. For this reason, the start of ignition is assumed here as the time elapsed to reach minimum pressure, and thus the ignition delay τ_1 is defined as the time from the start of injection to this ignition time. Experiments were performed for various pressures and temperatures at a fixed fuel quantity of $V_{inj} = 0.107$ ml and a fixed injection period of $t_{inj} = 6.5$ ms. The obtained results are expressed in terms of initial pressure and temperature at injection time, p_i and T_i respectively.

Figure 6 gives an Arrhenius plot of the results obtained at a fixed initial air pressure $p_i = 42$ kgf/cm². From this figure, it is first noted that the dependence of τ_1 on temperature differs between high and low temperature ranges around 710 K. This may be attributed to the fact that at high temperatures a time is required to form a combustible mixture, while at low temperatures chemical processes play a dominant role before reaching ignition. If we use a familiar form of

$$\tau_1 = a e^{b/T} \quad (3)$$

then the activation temperature b is 1210 K for the higher temperature range and 5560 K for the lower temperature range. The latter value is not remote from both Wolfer's result⁽¹⁾ of 4650 K and Fujimoto's result⁽³⁾ of 5130 K. In this figure, the plots of dark circle represent data gained at very high temper-

Fig. 7 Effect of pressure on ignition delay τ_1

atures by using a 21 % oxygen containing argon in place of normal air. It is noted that these plots seem to correlate well with air data.

In Fig. 7 the influence of pressure on τ_1 is given. It is shown that τ_1 decreases rapidly with pressure and finally reaches a constant value above a certain pressure. A similar result has already been reported on a bomb test using a multi-hole nozzle⁽³⁾.

3.2 Initial process of heat release

To have a better insight into the ignition process of a fuel spray, a faint heat release during the initial stages of ignition has been investigated to some detail, by means of the hot-motored technique established by Yu and his co-workers⁽¹⁰⁾. The principle of this method is that net heat-release due to chemical reactions might be obtained by comparing the pressure history at normal firing with the corresponding non-fired one. This approach is feasible since heat liberation can easily be related to the pressure change, and since the non-fired pressure curve may be obtained by using a nitrogen gas as charge in place of normal air. Let p and p_n denote pressures of the fired cycle and the non-fired cycle respectively, and the difference $(p-p_n)$ represents a net pressure rise. Figure 8 gives the $(p-p_n)$ versus time relations for various test conditions, in which t denotes the time elapsed from the start

of injection. The dotted curve in each group indicates the pressure difference between the non-injection and the nitrogen cycle (p_a-p_n), or in other words, a net pressure drop caused by a heat loss due to the spray development. According to the figure, the net pressure drop ($p-p_n$) remains zero until it begins to grow acceleratedly at a certain time (the first circular mark on each curve). This time will be taken as the beginning of net heat release, which we may designate as heat-release delay τ_0 . This delay may not be determined accurately partly because of the reproducibility of the experiment and partly because of the accuracy of the pressure measurement, nevertheless it is certain that such a time comes considerably earlier than the normally detectable ignition time as defined as an ignition delay like τ_1 . The second mark on each $(p-p_n)$ curve represents τ_1 which has already been defined as the time between the injection start and the minimum point of the pressure curve. In every case, there is an appreciable difference between τ_0 and τ_1 , which is largely dependent on the test condition. In other words, the so-called ignition delay is considerably longer than the true one owing to the very faint pressure rise. The delay period up to the time when two curves ($p-p_n$) and (p_a-p_n) become equal, or the time for pressure to recover from the drop due to heat absorption, may also be defined as another kind of ignition delay τ_2 . This is the same with that of Yu's definition,

Fig. 8 $p_a - p_n$ and $p - p_n$ versus time for different temperatures, pressures, and oxygen fractions

and of course it is much longer than τ_1 .

From (a) and (b) in Fig. 8, it is noticed that the lower the pressure and temperature, the slower the $(p-p_n)$ proceeds with time. In other words, a long ignition delay causes a gradual pressure rise at the very beginning of ignition, thereby bringing about a large $(\tau_1 - \tau_0)$. This is also the case when the oxygen fraction in the charge is changed, as has been indicated in (c). A low O_2 fraction leads to a long ignition delay which is followed by a slow pressure rise, while at a high O_2 fraction, the reverse is the case. As is shown in these figures, a faint pressure rise in the very early stage elongates the ignition delay substantially. Probably, this small pressure rise would be governed not only by chemical factors but also by physical ones such as spray development, evaporation, and mixing of fuel with air.

4. Main part of burning

Further experiments were performed on the rapid compression machine to study the progress of combustion following the ignition delay. In Fig. 9 pressure records are reproduced for various initial air temperatures at a fixed initial pressure of $p_i = 42 \text{ kgf/cm}^2$ and an overall fuel-air equivalence ratio of $\phi = 0.61$. The abscissa expresses the pressure measured from the initial pressure $\Delta p = p - p_i$. At first, Δp decreases gradually with time due to heat loss from the chamber walls.

Fig. 9 Pressure diagrams for different temperatures

Fig. 10 dp/dt diagrams for different temperatures

Even in the case when fuel is not introduced, this decrease is evident as shown on fine curves in the figure. In the fired case, the pressure behaves just as in an ordinary diesel combustion; a rapid pressure rise takes place which is followed by a gradual change until a peak pressure is reached. As the air temperature is lowered, both maximum rate of pressure rise and pressure jump during the initial stages increase, as would be conceived, owing to prolonged ignition delay.

In Fig. 10 the time-derivative of pressure, or the rate of pressure rise dp/dt is shown as a function of time for several cases each corresponding to Fig. 9. Clearly this form is convenient for consulting the details of the course of burning, since this value is proportional to the rate of heat release minus the rate of heat loss. In every diagram, there is a characteristic peak due to explosive burning, which is followed by a sudden depression. In the diffusion burning phase, or in the main part of burning, the rate of pressure rise again increases with time until it reaches a maximum, and then it decreases quickly. In the case of a long ignition delay, a lot of heat is liberated in explosive burning phase, thus bringing about a shorter and slower combustion in the diffusion burning phase. Owing to this, the time from injection to the end of combustion does not largely depend on τ_0 .

In Fig. 11 are given diagrams obtained at different equivalence ratios ϕ for fixed p_i and T_i . There is little change in the course of combustion at different ϕ until the end of explosive burning. This is clearly because of the same initial injection rate and the same ignition delay. In the subsequent course, however, a large difference is seen at different equivalence ratios ϕ ; the peak rate in the diffusive burning stages quickly increases as ϕ increases. It is rather surprising that the time from this peak to the end of burning appears to be little affected or rather shortened with an increase in ϕ . Thus the substantial burning period does not differ largely at different equivalence ratios.

Figure 12 presents dp/dt curves at several oxygen mole fractions in the gas charge under fixed initial pressure and

Fig. 11 dp/dt diagrams for different fuel-air equivalence ratios

temperature of $p_i = 42 \text{ kgf/cm}^2$ and $T_i = 680 \text{ K}$. With an increase in the oxygen fraction the ignition delay becomes shorter and explosive burning is slowed. This leads to an increase in the heat liberated during diffusive burning phase. Just as in the case when T_i is changed, the whole period of combustion from the injection start is not largely affected by the oxygen mole fraction.

Summarizing the above results obtained under various conditions, we may point out the following matters:

- (1) In the case of a short ignition delay, only a small heat may be liberated during explosive burning period, the remainder being allotted for the diffusion burning. During the diffusion burning, a high peak rate of heat release can be reached without causing a large prolongation of the heat release in final stages.
- (2) In the diffusion burning stages, the time to reach maximum rate of heat release appears to well correlate with the end of injection. However, this demands further clarification.
- (3) In the case when only a small heat is released in the diffusion burning period, such as in the case of low air temperature, low oxygen concentration, and small equivalence ratio, the rate of heat release during the diffusion burning decreases monotonically with time without showing a peak.
- (4) In the case of a high oxygen fraction in the charge, the diffusion burning does not proceed so quickly as expected from an abundance of oxygen.

In order to give rational explanations to the above-stated facts, a discussion is made of the processes involved in the spray combustion. During the diffusion controlled burning, the so-called jet diffusion flame would be established in which combustion takes place at the jet boundary consuming an air entrained from the surroundings (11). From this, picture, the accelerating rate of heat release observed early in the diffusive combustion may be attributed to the increase in the velocity of the air-entrainment with time, and the decreasing rate after the arrival at the peak rate may be to the decrease in the velocity of entrainment due to the decreasing jet momentum.

Fig. 12 dp/dt diagrams for different oxygen fractions

In the actual circumstances, however, a large flame zone develops around the spray after the initial rapid pressure rise. In this period, large and small-scale gas motions and turbulence would be generated both by the spray development and by gas expansion, and, in the subsequent stages, they would play a dominant role in mixing the fuel in the spray core with the surrounding fresh air. In other words, turbulent mixing might be essential to the diffusion controlled burning. One of the present authors has proposed a stochastic model to describe such a highly turbulent unsteady diffusion flame occurring in a confined space (12). This model predicts how the fuel-air heterogeneity progresses with time at random turbulent mixing. It has been suggested that if only a poor fuel-air mixture is given in the initial stage, the pressure in the combustion chamber rises only slowly at first but is strongly accelerated later from time to time. Essentially, this matter would provide an explanation to what has been described in the item (1) referring to the observed trend at a shorter ignition. In Fig. 13 are reproduced theoretical pressure-time relations for various initial degrees of fuel-air mixing g_F at an equivalence ratio of 0.60 under a fixed initial condition of 40 kgf/cm^2 and 750 K . In this figure, the ordinate expresses the non-dimensional time expressed in terms of the characteristic time of turbulent mixing. The parameter g_F means the fraction of such fluid particles that is shared with an equal amount of fuel at $\tau = 0$. Theoretically, once the fuel-air mixture ignites, all heat is instantly released, thus causing a pressure jump. This jump corresponds to the initial explosion in the actual progress and its magnitude can be determined by chemical equilibrium considerations. According to Fig. 13, the computed initial pressure jump is seen to increase with g_F ; in other words, a good mixture will give rise to a strong knock consistently our experience. It is further noted that at a high g_F the pressure makes a curve like a first-order delay starting from a high level, while at a low g_F the pressure change proceeds slowly at first and gradually becomes faster.

Fig. 13 Pressure histories predicted by the stochastic model

This trend for g_F is similar to that in the case when the air temperature is varied (see Fig. 9); the pressure history at a short ignition delay resembles a curve of a low g_F , and the one at a long ignition delay does a curve of a high g_F . In other words, a reduction in the ignition delay is equivalent to a decrease in g_F , and this is because the earlier the diffusive combustion starts, the less the spray develops and the less fuel-air mixture is available just at the end of explosive combustion. In summary, it may be stated that turbulent mixing would govern combustion during the so-called diffusion controlled phase. The characteristic heat-release rate curve at a short ignition delay is attributed to the fact that it takes a long time before the local fuel-air zone dilutes to form a mixture of adequate degree.

5. Conclusions

In this paper, a rapid compression machine has been developed to study basic aspects of the diesel combustion phenomenon. This machine features a free-moving piston which is pneumatically driven and is stopped by cramming into a tapered ring. A satisfactory performance has been obtained, permitting a study of ignition of a fuel spray and the subsequent combustion. From a series of tests carried out on this compression machine, the following conclusions are obtained:

- (1) The temperature dependence of the ignition delay at the high temperature range is lower than that at the lower temperature range. The pressure dependence of the delay is lost at pressures higher than a critical pressure.
- (2) Just after ignition, there is a faint heat release whose velocity increases acceleratedly with time. The evolution of heat at this stage is affected by the mixture formation. The ignition delay normally measurable would possibly be a resultant time including this effect.
- (3) In the case when only a small portion of heat is liberated in explosive combustion period, the succeeding diffusive combustion proceeds acceleratedly, giving a high rate of heat-release in the middle stages. On the contrary, in the case when a large portion of heat is liberated initially, the rate of heat-release during diffusive combustion decreases monotonically with time.

- (4) Turbulent mixing appears to govern combustion during the diffusion burning period. For an undeveloped spray, it takes a long time before a local fuel-air mixture dilutes to produce an adequate mixtures. This mechanism will give a proper explanation to the matter described in the item (3).

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