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An Analysis of Transient Combustion States
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This paper presents a three zone combustion model and a method of analysis with the intention of developing a prediction procedure with short computing time which can be applied to the estimation of the dynamic characteristics of combustion chamber.

The calculated results with respect to the transient flame developing with the supply of propane gas fuel to the chamber with pulse- and step-wise changes of flow rate are compared with the measured results obtained through an on-line operation of a digital computer.

It was revealed that this method of analysis is useful for the estimation of the space- and time-varying changes of the states which are most required for understanding the phenomenon, such as heat release rate, temperature and velocity.

Key Words : Analysis, Three Zone Combustion Model, Transient Combustion State, Diffusion Flame

1. Introduction

In the transient combustion states from the beginning of fuel and air supply through ignition to continuous combustion, occurrence of troubles are frequently observed such as flash back, blow-up, unburnt fuel exhaust and oscillatory combustion. If dynamic combustion characteristics of combustors or furnaces can be quantitatively analysed, occurrence of those problems will be preventable and helpful data for flow rate control and improvement of furnace shape to secure smooth combustion may be obtained. However, at present, several analyses are available only about the non-steady phenomena, such as combustion oscillation, in a continuous combustion state, but there are few studies which deal with transient phenomenon of diffusion flame. As regards the principal methods for analysing non-steady combustion, there are ; (1) Lehmann's analysis^[1] calculating whether the pressure pulse shot in a solid propellant rocket motor is enlarged or not as the two dimensional changes with respect to the axial distance x and time t ; (2) the author's analysis^[2] simulating a self-excited oscillation phenomenon in a furnace also in the dimensions of x and t ; (3) Bucher's method^[3] analysing a high-frequency self-excited oscillation in a plate type rocket motor ;and (4) Rivard's one^[4]

computing non-steady state field in a gas laser. Among these methods, Bucher's and Rivard's ones also take account of the changes of states in the transverse direction y . However, from the condition imposed so that the computation might not be diverged the restriction such that the time division Δt for an iteration step should always be held lower than the lesser part of the quotients obtained by dividing the minimum divisions $\Delta x_{\min}$ and $\Delta y_{\min}$ by the sum of sound and flow velocity. Seeing that Bucher, by the use of one of the highest performance computers, could only simulate short-time non-steady phenomenon and that Rivard could deal only with the computation of the partial region of combustor, it appears that simulations of transient combustion state in a furnace are practically difficult to perform in the dimensions of x , y , and t .

In applying the method of calculation to the stage of planning or drawing combustion chambers, although estimation of the values simply averaged over a chamber cross-section is meaningless, there is no need of deriving local values of states.

At present, it is therefore desirable to develop a simple method of analysis with short computing time, and capable of estimating the principal changes of values in a section as well as their axial changes. In response to these requirements, we aim to predict in future the pulsed combustion state^[5] and the transient combustion states in boilers and furnaces and to estimate the oscillatory combustion state. Accordingly, this report proposes a three zone model of combustion and presents an analysing method capable of simulating the movement of regions developing with the progress of combustion as well as the instantaneous states of the transient flame such as temperature and velocity. Numerical analysis is performed about the flame established when the fuel

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of propane gas with an abrupt flow rate change penetrates into the flows of air and fuel for stabilizing the flame, both of which are supplied at constant pressure. The results are compared with the experimental ones obtained through an on-line operation of a digital computer.

2. Experimental methods

The burner used in the experiments is modelled upon an industrial gas burner and is composed of two coaxially arranged brass pipes, the large one of which has $\phi 15.9$ od. and $\phi 12.7$ id. and the small one has $\phi 11.7$ od. and $\phi 10$ id. Inner pipe is a main fuel (commercial propane gas) nozzle. The propane gas for stabilizing flame was always supplied through the annular part between both pipes and was fired throughout the experiments. With the intention of taking Schlieren photographs of flame a rectangular cross-sectional combustion chamber of 90 mm wide and 45 mm deep was used, and the abovementioned burner was set at this center. The transient flame was formed between the air flow supplied from the surrounding region of the burner and the main fuel flow step- or pulse-wise supplied by opening an electromagnetic valve. In either fuel supplying condition, the air and stabilizing fuel flow are maintained at the rates of $Q_a = 16$ l/s and $Q_f = 0.016$ l/s respectively.

Fig.1 shows the velocity variation with time of the main fuel flow discharged from the burner after opening the electromagnetic valve. In this figure, the waveform for the step type fuel supply is shown only up to 0.2 s, subsequently the velocity decreasing gradually to 0.33 m/s (fuel flow rate $Q_f = 0.026$ l/s) at 0.8 s.

In order to raise the measuring accuracy and efficiency of data processing, all the experimental operations such as opening and closing the electromagnetic valve, high-speed Schlieren photographing, data sampling and calculating, and statistical treatment were carried out in an on-line operation of the digital computer (FACOM U-200). For the measurements of dynamic-pressure and temperature a condenser microphone type velocimeter [6] was used. A pair of fine thermocouples (Pt-Pt/Rh 13%; 25.4 μ m) for measuring temperature was installed in the probe of this velocimeter, and the velocity and temperature were then simultaneously computed while compensating instantaneously [7] for both the response lag due to thermo-inertia and the non-linear relation between temperature and thermo-electromotive force. Namely, by the use of the dynamic-pressure and thermocouple signals, instant time-constant, temperature and velocity were determined by solving iteratively the system of equations which were composed of such relations as kinematic viscosity against temperature, Nusselt's equation by Ahmad, response time-constant of

thermocouple against heat transfer coefficient, Bernoulli's equation, calibration curve of thermo-electromotive force against temperature and first order time-lag equation of response.

3. Theory

3.1 Modelling of transient diffusion flame

Transient combustion field in the combustion chamber is modelled as the movements of three zones. Fig.2 shows a schematic view of this model. Those three zones are divided into; Zone 1 in which the fuel ejected from the nozzle is flowing in the unburnt state; Zone 3 in which the air supplied from the cross-section surrounding the nozzle is streaming and; Zone 2 in which the fuel and air are mixed and are reacting with each other and the products between them are flowing. As regards the quantities of state in a zone at a certain axial position, representative values are considered as uniformly distributed. It is then assumed that pressures over the three zones are equal but the other quantities take their own values in each zone. Using this model, fundamental equations for representative values are derived and development of equations are carried out, and the following assumptions are then employed:

(1) The combustion chamber is axisymmetric and has a constant cross-sectional area in its axial direction, and the fuel is supplied from the center at the inlet section of the chamber.

Fig.1 Variation of fuel supply velocity.

Fig.2 Combustion model.

(2) Ideal gas law applies.

(3) Decreasing rates of the cross-sectional areas of Zones 1 and 3, in other words, increasing rate of the area of Zone 2 is proportional to the summation of the mass flow rate of the fuel which moves from Zone 1 to Zone 2 and that of the air from Zone 3 to Zone 2.

(4) It is thought that momentum and mass exchanges proceed only at the interfaces between regions. Concerning the heat transfer, exchange between regions at their interface, axial exchange in Zone 2, and radiation transfer from Zone 2 to chamber wall are taken into consideration.

(5) The Reynolds analogy [8] is applicable to the momentum, mass and heat transfers at interfaces, and then, momentum transfer obeys the concept proposed by Prandtl and Reichardt-Bousinesq [9].

(6) The reaction rates of fuel and oxygen respectively denoted by w_f and w_{O_2} are equal to the respective stoichiometric parts of the flow rates of the fuel and oxygen which enter a control volume concerned from upstream of Zone 2 and through interfaces from Zone 1 and 3.

(7) The gas constant R , specific heat at constant pressure c_p , specific heat ratio κ , kinematic viscosity ν , Prandtl number Pr , turbulent Prandtl number Pr_t , Schmidt number Sc , turbulent Schmidt number Sc_t , emissivity of grey body ϵ , absorptivities ϵ_g , ϵ_w respectively for gas temperature T and wall temperature T_w and empirical constant which decides the turbulent exchange rate remain constant regardless of location and time.

3.2 Fundamental equations

Based on the modelling mentioned in Sec. 3.1, fundamental equations are expressed by the use of the following symbols ;

A	: cross sectional area of a region,
c_b	: radiation constant,
d	: outer diameter of a region,
$D^{(e)}$	: effective diffusivity,
g	: gravitational acceleration,
G	: friction force,
h	: heat flux,
H	: calory of heat transfer,
J	: Joule's equivalent of heat,
N	: quantity of mass transfer,
p	: pressure,
q_{hr}	: heat release rate,
q_{ra}	: radiation heat transfer rate,
t	: time,
T	: absolute temperature,
u	: flow velocity,
v	: mass flux between regions,
w_{O_2}, w_f	: reaction rates of oxygen and fuel, respectively,
x	: axial distance,
Y_{O_2}, Y_f	: mass fractions of oxygen and fuel, respectively,
$\lambda^{(e)}$	: effective heat conductivity,
ρ	: gas density,
τ	: friction stress,
ϕ	: mass ratio of fuel to oxygen,
ψ	: mass fraction of oxygen in the air,
{ suffixes }	
k	: region number (= 1, 2, 3),
w	: wall.

Ideal gas equation (1), equation representing change of region area (2) and continuity equation (3) are respectively expressed as;

$$p = \rho_k g R T_k \dots\dots\dots (1)$$

$$\rho_k \frac{DA_k}{Dt} = -N_k \dots\dots\dots (2)$$

$$\frac{\partial \rho_k A_k}{\partial t} + \frac{\partial}{\partial x} (\rho_k u_k A_k) = -N_k \dots\dots\dots (3)$$

Equations of species conservation (4), momentum (5) and energy balance (6) for Zone 2 are ;

$$\left. \begin{aligned} \frac{\partial \rho_2 A_2 Y_f}{\partial t} + \frac{\partial \rho_2 A_2 u_2 Y_f}{\partial x} &= \rho_2 A_2 D_2^{(e)} \frac{\partial^2 Y_f}{\partial x^2} + N_1 - w_f \\ \frac{\partial \rho_2 A_2 Y_{O_2}}{\partial t} + \frac{\partial \rho_2 A_2 u_2 Y_{O_2}}{\partial x} &= \rho_2 A_2 D_2^{(e)} \frac{\partial^2 Y_{O_2}}{\partial x^2} + \phi N_1 - w_{O_2}, \\ w_f &= \phi w_{O_2} \end{aligned} \right\} \dots\dots\dots (4)$$

$$\begin{aligned} \frac{\partial \rho_k u_k A_k}{\partial t} + \frac{\partial}{\partial x} (\rho_k u_k^2 A_k) \\ = -A_k \frac{\partial p}{\partial x} + \rho_k g A_k - G_k - u_k N_k \dots\dots\dots (5) \end{aligned}$$

$$\frac{Dp}{Dt} - \kappa \frac{p}{\rho_k} \frac{D\rho_k}{Dt} = - \frac{J(\kappa-1)}{A_k} H_k \dots\dots\dots (6)$$

The symbol D/Dt denotes substantial time derivative. Friction force G_k acting on Zone k , mass inflow N_k into Zone k and heat transfer calory H_k are estimated according to Tab. 1. On the basis of the Reynolds analogy, mass flux v , friction stress τ and heat flux h generated at the interface between Zone 1 and 2 are given in Tab.2, in which each term is affixed with the suffix 12.

Symbols d_1 , d_2 and d_3 respectively denote outer diameters of Zones 1, 2 and 3; r_j stands for the imaginary radius of Zone 2 [= $(d_1 + d_2)/4$], and ρ_{12} denotes average density of Zones 1 and 2 [= $(\rho_1 + \rho_2)/2$]. Among the principal terms in Tab. 1, $N_1 = \pi d_1 \rho_1 v_{12}$ indicates the quantity of fuel component transferred per time per axial distance and N_3 the air quantity engulfed from Zone 3 to 2. The symbol $G_1 = \pi d_1 \tau_{12}$ denotes fluid friction force exerting on Zone 1. The term $\pi d_1 h_{12}$ in H_1 refers to heat transfer calory from Zone 2 to 1. The second term is the formally added one for reflecting the assumption that one half of the heat quantity generated through the fluid friction between Zone 1 and 2 enters Zone 1, but in the combustion field this term is negligibly small compared with the others. The term $-\pi d_1 \rho_2 g v_{12} c_p (T_1 - T_2)$ in H_2 is the thermal value necessary to raise the temperature of that gas of Zone 1 which is engulfed into Zone 2, to the temperature of Zone 2. The final term expresses the turbulent heat value conducted in Zone 2 in the longitudinal direction.

Radiation heat transfer per time per length q_{ra} is expressed as ;

$$q_{ra} = \epsilon c_b \left[\epsilon_w \left(\frac{T_w}{100} \right)^4 - \epsilon_g \left(\frac{T_2}{100} \right)^4 \right] \frac{\pi}{2} (d_1 + d_2) \dots\dots\dots (7)$$

By use of lower calorific value H_f and reaction rate w_f , heat release rate q_{hr} is represented by

$$q_{hr} = w_f g H_f \dots\dots\dots (8)$$

Large numbers of the reaction rate expressions applicable to a steady turbulent jet flame are available, but the expression for a non-steady flame has not yet been proposed. Besides, if the reaction rate expression, for instance, such as EBU model [11] is used in this calculation, the numbers of variables and equations would increase, and at the same time, the meaning of the author for having established a simplified method of calculation would be diluted. Consequently, by the use of only the variables hitherto introduced, we have tried to express the instant reaction rate in the following forms.

Fuel and oxygen flowing into a certain control volume (width of Zone 2 $\times$ length division Δx) in Zone 2 within a time division Δt can react with each other maximally at the stoichiometric fraction. The maximum values are then symbolized by $w_{f,max}$ and $w_{O_2,max}$, and the instant reaction rates are considered, as was stated in the assumption (6), as being equilibrated to those values. Thus,

$$w_f = w_{f,max}, \quad w_{O_2} = w_{O_2,max} \quad (9)$$

In the vicinity of the fuel port, fuel flow rate transported to the interface is of the same order as air flow rate, the reaction is therefore limited by the oxygen diffusion. In this case, the expression alternative to Eq.(9) and easy to understand is,

$$w_f = \phi N_s, \quad w_{O_2} = \phi N_s \quad (10)$$

The fuel remaining unburnt in the upper stream flows toward downstream in Zone 2 and ceases to react at the location where the fuel flow from Zone 1 becomes little, and this process can be expressed by the equations (4) and (9). This treatment of reaction rate and the assumptions (5) and (6) in Sec. 3.1 are the traditional thinking which has been applied to steady jet flames. To clarify to what degree they are applicable to a non-steady flame is one of the problems to be examined.

Tab.2 Flux and stress between Zones 1 and 2.

v_{12}	$\left[\frac{\nu}{S_c} + \frac{\chi r_f u_2 - u_1 }{S_{ct}} \right] \frac{1}{r_f} = \frac{D_{12}^{(e)}}{r_f}$
τ_{12}	$\rho_{12}(\nu + \chi r_f u_2 - u_1) \frac{u_1 - u_2}{r_f}$
h_{12}	$\rho_{12} g c_p \left(\frac{\nu}{P_r} + \frac{\chi r_f u_2 - u_1 }{P_{rt}} \right) \frac{T_1 - T_2}{r_f} = \frac{\lambda_{12}^{(e)}}{r_f} (T_1 - T_2)$

Tab.1 Quantities exchanged between zones.

	$k=1$	$k=2$	$k=3$
N_k	$\pi d_1 \rho_1 v_{12}$	$-\pi d_1 \rho_1 v_{12} - \pi d_2 \rho_3 v_{32}$	$\pi d_2 \rho_3 v_{32}$
G_k	$\pi d_1 \tau_{12}$	$-\pi d_1 \tau_{12} - \pi d_2 \tau_{32}$	$\pi d_2 \tau_{32} + \pi d_3 \tau_{3w}$
H_k	$\pi d_1 h_{12}$ $\frac{\pi d_1 \tau_{12}(u_1 - u_2) }{2f}$	$-\pi d_1 h_{12} - \pi d_2 h_{32} - q_{hr} - q_{ra}$ $\frac{\pi d_1 \tau_{12}(u_1 - u_2) }{2f}$ $\frac{\pi d_2 \tau_{32}(u_3 - u_2) }{2f}$ $-\pi d_1 \rho_1 g v_{12} c_p (T_1 - T_2)$ $-\pi d_2 \rho_3 g v_{32} c_p (T_3 - T_2)$ $-\frac{\pi}{4} (d_2^2 - d_1^2) \lambda_2^{(e)} \frac{\partial^2 T_2}{\partial x^2}$	$\pi d_2 h_{32} + \pi d_3 h_{3w}$ $\frac{\pi d_2 \tau_{32}(u_3 - u_2) }{2f}$ $\frac{\pi d_3 \tau_{3w} u_3 }{f}$

4. Calculation method and numerical values used in the computation

By the use of the condition of expression (1), conservation equations from Eq.(2) to Eq.(6) are rearranged into the total derivative forms consisting of the following non-dimensional variables.

$$\left. \begin{aligned} P &= \left(\frac{P}{P_{**}} \right)^{(\kappa-1)/2\kappa}, \quad A_s = \frac{a_s}{a_{**}}, \quad U = \frac{u}{a_{**}} \\ X &= \frac{x}{l_{**}}, \quad Z = l \frac{a_{**}}{l_{**}}, \quad F_{\pm} = U \pm \frac{2PA_s}{\kappa-1} \end{aligned} \right\}$$

(where double signs indicate order type)(11)

Where, the symbol a denotes sound velocity, suffix o represents the ambient state and suffix os indicates the imaginary state derived on the assumption that the present state is changed adiabatically to the ambient state. This operation firstly gives the forms of total differential equations [2], [10] with respect to F_+ and F_- (pressure mode). The characteristic direction for F_+ is taken as the sum of the flow velocity and the sound velocity and that for F_- is taken as the difference between the flow velocity and the sound velocity. The operation secondly gives those with respect to the zone area A_k , sound velocity ratio A_s , concentrations Y_f , Y_{O_2} (entropy mode), for which the characteristic direction is taken as the flow velocity.

Numerical computation was carried out by transforming those total derivative forms respectively obtained for three regions into the difference forms calculable in the characteristic direction. The grid dividing method (2) was then applied in which entropy mode grids are densely divided compared with those for pressure mode. As the boundary condition for inlet section,

Tab.3 Principal numerical values used for calculation.

Terms	Symbols	Values	Units
Empirical const. for mixing	χ	0.003	
Laminar Schmidt number	S_c	0.72	
Turbulent Schmidt number	S_{ct}	0.7	
Laminar Prandtl number	P_r	0.72	
Turbulent Prandtl number	P_{rt}	0.7	
Emission const. of grey body	C_b	1.36×10^{-5}	kcal/(cm ² sk ⁴)
Emissivity of grey body	ϵ	1.0	
Absorptivity of gas for τ	ϵ_g	0.4	
Absorptivity of gas for τ_w	ϵ_w	0.4	
Calorific value of propane gas	H_f	12034	kcal/kg fuel
Length of combustion chamber	L	103	cm
Diameter of combustion chamber	D_1	7.18	cm
Nozzle diameter for fuel	D_2	1.0	cm
Nozzle diameter for pilot fuel	D_3	1.1	cm

non-dimensional supply pressure $[P + \{(\kappa - 1)/2\} (U/A_s)]$ was given. The fuel pressure for the stabilizing flame and the air pressure were maintained constant, but the supply pressure at the fuel nozzle section was changed with time, in the wave in the wave forms as was estimated from the experimental values shown in Fig.1. Length of the furnace ($L = 103$ cm) was divided into 8 sections for pressure modes and into 80 sections for entropy modes. Numerical computation was run by the use of the computer, ACOS 700, NEC CO., LTD., and 34 min cpu time was required for simulating the phenomenon for one second, real time.

Numerical values used for calculation are shown in Tab.3. Both the values of ϵ_g , ϵ_w for propane gas are estimated at 0.4 from Ref.(12). The value 0.003 was given for the empirical constant χ . This value was determined on the assumption that the stoichiometric parts of the air and fuel carried to the reaction region, that is, Zone 2, are releasing heat. It may then be possible to give $\chi = 0.0256$ [13] by regarding Zone 2 as the region where foreign gases are turbulently mixed or to yield a greater value by taking account of that increase of engulfed air [14] which is peculiar to non-steady jet. For the latter case, heat-release rate should be set lower than the value estimated from expressions (9) and (10) by assuming that a part of the mixed gas is then reacting.

5. Results and considerations

Figure 3 shows a photograph of pulse type combustion process taken by the use of high-speed Schlieren photography. Actual vertical dimension of a picture is 140 mm. For a while after the opening of electromagnetic valve (designated in the figure by E.M.V. Open), the image of the flame for stabilizing with dimmy shadow is seen, and after 20 ms elapse a black part developing toward vertical from nozzle is

Fig.3 High-speed Schlieren pictures of pulse type flame.

observed. This part is occupied almost with unburnt fuel and combustion reaction is going on between the shadow part with large scale disorder and the black part. Outside part without darkness and brightness change is the surrounding air flow region. Owing to turbulent mixing these regions have not smooth boundaries; however, from these sequential pictures the difference between three kinds of shadows and the change of shadow with time can be clearly distinguished. These observations offered one of the grounds to the present model which deals with the combustion field in three separated regions.

Figure 4 depicts the temperature and region diameters at the steady fuel-supplying state (step type, 1.2 s after E.M.V. opens) settled after the opening of the electromagnetic valve. Fig.5 shows a change of combustion efficiency in the same condition. The calculated values in Fig.4 are representative of Zones 1, 2 and 3. The measured values are the averaged ones of the mean-time temperature on the center axis and the mean-time temperature at a radial location at which temperature becomes maximum. The dot-dash line illustrated in the figure of region diameter is a measured curve which represents twice the radius, the temperature at which becomes equal to the average value of the maximum and the surrounding air temperatures. If we observe these figures by considering that the measured values are the ones at the specific location and that the calculated values are representative ones within a certain region, it can be deduced that this analysis is useful as a simple calculating method at least for obtaining such principal combustion states as flame length, temperature distribution, and heat-release distribution in steady combustion.

Fig.4 Steady combustion state.

In what follows, examples illustrating the transient combustion states will be described. Figs. 6 and 7 are the calculated results and are respectively related to the step type and the pulse type fuel-supplying processes. These figures schematically illustrate in what manner the temperature of Zone 2 T_2 , diameter d_1 , d_2 , heat release rate per time per length q_{hr} , convection heat transfer from Zone 2 to Zones 1 and 3 (this rate corresponds to $\pi d_1 h_{12} + \pi d_3 h_{32}$ in Tab.1) and radiation heat transfer q_{ra} change according as the time passes after the start of fuel supply and as the axial distance changes. Line number 0 in this figure is the predetermined initial distribution for calculation. Also shown are the curves from number 1 to 4 for every 0.1 s step and the distributions for step type when the fuel supply settles to steady state. The line number of the steady state is denoted by the symbol s. Aforementioned Figures 4 and 5 illustrate the parts of the calculation results for this steady fuel-supply condition.

From the step type results in Fig. 6, it can be recognized that after the time elapse of 0.2 s the diameter of Zone 2 becomes nearly constant and that the fuel zone diameter is still 0.3 cm at the furnace exit ($x = 1.03$ m), therefore, that large quantities of unburnt gases are being exhausted out of the furnace. Furthermore, they well express the characteristic features at non-steady fuel supply such that once the flame reaches the maximum length (subsequent lines of 3), d_1 disappears from the down-toward the upperstream, and hereupon, that the change of heat release appears to be propagating from downstream toward upstream. The different points of the pulse type in Fig. 7 from the step type are that maximum length of fuel region reaches at most the order of furnace length since supplied fuel amount is little and that from before 0.2 s a constriction of fuel region begins in the vicinity of the fuel entrance section and that, as are indicated by lines 3 and 4, fuel region seems to disappear simultaneously from up- and downstream. The states such as these correspond qualitatively with the change of Schlieren image in Fig. 3.

Fig.5 Combustion efficiency at steady combustion state.

It is found that the temperature and heat release rate for pulse type in the heat-releasing field are of the same order as those for step type and that about 40 percent of heat release is transferred to the wall by radiation heat transfer. About 30 percent or less of the remainder 60 % is transferred to Zones 1 and 3 and spent in heating the surrounding air and fuel gases. The remainder of them is thus spent in raising the temperature of Zone 2 itself and that of the air and fuel engulfed into Zone 2. Within the period when the flame is propagating toward downstream, heat release becomes a maximum in the vicinity of the fuel flow tip. The fact that a strong turbulence of the shadow existed at the tip of the fuel flow in the Schlieren pictures and the measured result of ion current confirmed this phenomenon.

Figures 8 and 9 show the temperature and velocity wave forms measured on the center axis, $x = 54$ cm. For step type, the calculated results for Zones 1, 2 and 3 are shown, and for pulse type, those for

Fig.6 Combustion process for step type fuel supply.

Fig.7 Combustion process for pulse type fuel supply.

Zones 1 and 2 are indicated. The fluctuations observed in the calculated wave forms depend on the propagation of those pressure and temperature waves which represent each region. The measured wave forms are random fluctuations generated by the propagation of the temperature, pressure and vortex waves. For the comparison between both results, random fluctuations in the measured results were removed by ensemble-averaging the data for 50 times measurements. The ensemble-averaged results are indicated with short dash lines. For reference, are shown the curves of the rms value of the variance fluctuations with respect to ensemble average in both cases of step type and pulse type. The rms data (symbol 0) of the variance fluctuations with respect to the mean-time values within every 120 ms are indicated in the case of step type.

Temperature wave forms of step type and the temperature and velocity wave forms of pulse type obtained by calculation represent well the characteristics of the phenomenon. But some differences are seen between the velocity wave forms of step type by calculation and those by experiment. This is because the instant values of non-steady phenomenon are strongly influenced by the phenomena at all of the times and locations in the past; hence, the differences between the calculated and the experimental results have been accumulated. However, considering that the steady state calculation results after

Fig.8 Temperature and velocity for step type fuel supply.

long time elapse could well express the actual state, it can be concluded that the estimation of the transient combustion states is possible. The rms values of fluctuations become high at the arrival of the flame, and this is because the time of flame arrival at the measuring point differs in each measurement. Excepting this part of difference, it is found that fluctuations for ensemble average and the ones for mean-time are almost of the same magnitude, and that the variation of the magnitude which occurs with time elapse is only a little.

Figure 10 indicates the results for pulse type and compares the measured wave forms with the calculated ones both for $x = 31$ cm. The former are the ensemble averaged ones of temperatures and velocities for fifty times measurements at four radial locations and the latter are the moving averaged ones obtained by averaging the calculated wave forms of temperatures and velocities within every 60 ms. These statistical treatment removes the randomness of the phenomenon and enables a comparison of low frequency variations. The results in the figure indicate that velocity wave forms as well as temperature wave forms are accompanied with considerable changes in the radial direction. Because, in this analysis, Zone

Fig.9 Temperature and velocity for pulse type fuel supply.

2 is defined as a reaction region, it is possible to make the wave forms of Zone 2 comparable with those of line number 2. Because Zone 1 is taken as the fuel region, the wave forms for Zone 1 are comparable with those obtained on the center axis. From the observation of the time-varying pattern of the waves it is concluded that the calculated wave forms correspond with those in the actual process although the flame arrival time in the calculation is delayed compared with the actual case.

From the hitherto mentioned results, it may be concluded that the thinking for turbulent mixing mentioned in Tab.1, which has been used for steady jets and steady jet flames, and the heat release expression (9) are available as the first approximation of the mixing and heat release in the transient combustion state.

6. Concluding remarks

As an attempt to predict the transient combustion state in heating and combustion furnaces and others, a three zones model of combustion was proposed and an analyzing method which uses the characteristic curve was presented. The calculated results were compared with the experimental ones, depending on the data such as the instant wave forms of temperature and velocity converted from the signals from the condenser microphone velocimeter in an on-line operation in a digital computer and the statistically treated results of those signals.

Results show that this method of analysis can give the information on the steady combustion state, and moreover, can express both of the features of the movement of regions which occur along with the progress of combustion and of the time-varying changes of the representative states in the regions.

That is to say, it has become possible to estimate the changes of temperature, velocity, heat release rate during transient combustion process and the transferred fractions in the heat release rate to the wall and to the surrounding air flow respectively. It is concluded that, even for transient combustion, usual empirical equations about turbulent exchange are applicable and that the expression (9) which has been used in estimating the non-steady heat release from the exchange rate is available.

Since the present method of analysis can calculate the principal change of state in a simple way, this method is advantageous in calculation time in comparison to the one hitherto used about non-steady and steady combustion.

As for the points to be improved, the present method should be expanded to a multizone model including regions such as

Fig.10 Radial distributions of temperatures and velocities at pulse type combustion state.

wall boundary layer, recirculating flow region and jet mixing layer. Thus improvements in prediction accuracy in non-steady heat release and turbulent heat exchange are expected, and at the same time, development of this method for various uses is anticipated. Because the computer time is proportional to the number of lateral divisions, it is likely that this expansion will never increase the calculation time by the present method over that by the method hitherto used.

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