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Numerical Analysis of Non-Combustion and Combustion Fields Accompanied with Swirl Flows*

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Numerical and experimental analyses were carried out with respect to the relationships among the change of diffusion and combustion processes, the location of fuel injection port and the distribution profiles of surrounding air flows.

With the object of simulating the complex flow field, a calculation was made under the conditions of three inlets using velocity profiles measured at respective inlets. Effective viscosity expressed in a similar formula to Prandtl one and numerical analyzing procedures of effective difference, finite reaction rate and of combustion efficiency were introduced to improve the applicability and the convergence characteristics of the calculation.

The calculation results agreed qualitatively with the experimental ones, and it was concluded that the diffusion rate of fuel is significantly influenced by the fuel supplying position and the surrounding air flows.

1. Introduction

Gaseous fuel supplied from a burner into a combustion chamber diffuses and burns in various manners according to the location of a burner port and to conditions of surrounding air flow rate and velocity distributions. This may be evident in view of the fact that air swirls are used in reducing NO_x formation [1],[2] and in controlling flame length. However, there are few reports which deal with both numerical calculation and experiment with the object of making clear the relationships between fuel supply position, velocity distribution profile, presence of swirling air, and diffusion- and combustion-processes.

Numerical simulation procedures are classified into two principal methods: one of which is the method proposed by Patanker [3], Spalding [4] and Mason [5] which solves the systems of parabolic partial-differential equations simplified under the assumption that there is no region of circulating flow, and the other is the method proposed by Gosman [6], which makes it possible with the use of "the upwind difference method" presented by Courant [7] to solve the systems of elliptic partial-differential equations applied even to the flow field accompanied with a circulating flow. Because the circulating flow is generated in that flow field accompanied with swirl, with which is concerned the present paper, the first calculation method is hard to apply [8]. Nevertheless, because in the second method inlet velocity distributions were assumed uniform and simplified formula of viscosity was used, an exact diffusion

process is difficult to simulate; in addition, because only the calculation method of the adiabatic-combustion field which correlates combustion temperature to fuel concentration in a fixed relation is expressed, the combustion process is difficult to discuss.

In the present study, numerical analysis and experiments are carried out on the relationships between fuel supply position, strength and distribution of surrounding air-flows with swirl, and initial diffusion- and combustion-processes. The numerical calculated results are compared and examined with the experimental ones. Numerical analyzing method is expressed in a form more generalized than the second method. Namely, an effective viscosity formula for a complex state of flow, a mixing length formula and an effective difference method of a differential coefficient necessary for estimating the viscosity are expressed on the basis of Prandtl's mixing length hypothesis [9] and Nikuradse's formula for the length [10]. The applicability and convergence characteristics of these expressions are compared with those in the use of Gosman's formula. Calculations of combustion efficiency and of finite reaction rate on the over-all reaction assumption are also carried out. The use of the velocity profiles measured respectively at three flow inlets permits detail examination of diffusion- and combustion-processes in the region near the inlets.

2. Main notations

c_p	: specific heat at constant pressure
D	: diffusion coefficient
g	: gravitational acceleration
h	: calorific value of fuel
p	: pressure
R	: maximum radius
T	: absolute temperature
v	: flow velocity

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w : reaction rate
 Y : weight fraction
 λ : heat conductivity
 μ : viscosity
 ρ : density
 x, r, θ : longitudinal distance measured from inlet, radial distance from center axis and angle in swirl direction respectively
 [Subscripts]
 (e) : for effective value
 (1) : for laminar value
 (0) : for reference value
 f, O_2, N_2, H_2O and CO_2 : respectively represents Fuel (= $C_nH_mO_o$), oxygen, nitrogen, aqueous vapor and carbon dioxide, and these notations are generically termed k .

3. Method of analysis

3.1 Arrangement of fundamental equations

The flow inlet section of the combustion chamber is composed of three concentric pipes. The combustion pipe is a straight one of maximum radius among three pipes. Gaseous fuel is supplied from the central pipe or from the ringed inlet pipe, and air flow with swirl is fed through the inlets coaxially situated outside of it. Flow field is steady and axi-symmetric. Velocities in three co-ordinate directions (v_x, v_r, v_θ), absolute temperature T , pressure p , concentration Y , density ρ and other variables vary according to both longitudinal distance x from entrance section and distance from pipe axis r . With the object of simulating this combustion field, perfect gas law and conservation laws of mass, chemical species, momentum and energy in a cylindrical co-ordinate system are summarized by the use of vorticity $\omega(\Omega)$, stream function ψ and mass flow rate G . In order to reduce the computing time, empirical equations of mixing length and algebraic turbulence viscosity are used, then, fluxes of stress, mass and heat transfer are expressed in terms of effective viscosity $\mu^{(e)}$, Prandtl number Pr and Schmidt number Sc on the basis of the similarity assumption among fluxes. The following system of partial-differential equations of general elliptic type is then obtained :

$$a \left\{ \frac{\partial}{\partial x} \left(\psi \frac{\partial \psi}{\partial r} \right) - \frac{\partial}{\partial r} \left(\psi \frac{\partial \psi}{\partial x} \right) \right\} - \frac{\partial}{\partial x} \left\{ br \frac{\partial (c\psi)}{\partial x} \right\} - \frac{\partial}{\partial r} \left\{ br \frac{\partial (c\psi)}{\partial r} \right\} + rd = 0 \quad (1)$$

Table 1 indicates the relationships among unknown variables and unknown coefficients a, b and c in the above equation. Wherein, the notations ω, ψ, Pr and Sc are expressed respectively by the following equations:

$$\Omega \equiv \omega/r = \frac{1}{r} \left(\frac{\partial v_r}{\partial x} - \frac{\partial v_x}{\partial r} \right) \quad (2)$$

$$G_r \equiv \rho v_r = \frac{1}{r} \frac{\partial \psi}{\partial r} \quad (3)$$

$$G_r \equiv \rho v_r = \frac{-1}{r} \frac{\partial \psi}{\partial x} \quad (4)$$

$$Pr = c_p \mu^{(e)} / \lambda^{(e)} \quad (5)$$

$$Sc = \mu^{(e)} / (\rho D^{(e)}) \quad (6)$$

The numerical calculation of the finite difference equations obtained from the integration of differential equations can be performed in a successive-substitution procedure by the use of conventional method, that is, Gauss-Seidel and S.O.R. methods, but an ingenious calculation technique is required to derive the difference formulae for w_k, T and $\mu^{(e)}$, and to obtain rapidly and securely the convergence. In what follows, the calculation procedures shall be described with stress on this technique.

3.2 Auxiliary equations

3.2.1 Boundary conditions

The boundaries of combustion chamber are inlets, outlets, pipe walls, and a central axis. The boundary values are assigned as constant values or boundary equations according to the physical conditions of respective boundaries as represented in Tab.2. The present calculation uses three inlets through which combustible gases are supplied in an arbitrary distribution, as against Gosman's calculation which deals with the flow field in cases of two inlets and uniform inflow at respective inlets.

3.2.2 Adiabatic combustion temperature T_b , reaction rate w_k and combustion efficiency ξ

Gosman disregarding the creation and consumption of chemical species computed the combustion temperature T_b by use only of the fuel- and air-concentrations. Namely, T_b is

Tab.1 Relationships among unknown variables ϕ and unknown coefficients a, b, c and d

ϕ	a	b	c	d
ψ	0	$1/(\rho r^2)$	1	$-\Omega$
Y_k	1	$\mu^{(e)}/Sc$	1	$-w_k$
T	1	$\mu^{(e)}/Pr$	1	$(\frac{1}{k} h_k^{(0)}) w_k + \tau G_x \frac{\partial c}{\partial x} + \tau G_r \frac{\partial c}{\partial r} / c_p$
$r v_\theta$	1	$\mu^{(e)}/r^2$	$1/r^2$	0
Ω	r^2	r^2	$\mu^{(e)}$	$-\frac{\partial}{\partial x} (\rho v_\theta^2) - r (g_x + \frac{\partial}{\partial x} (\frac{v_x^2 + v_r^2}{2})) \frac{\partial p}{\partial r} + r (g_r + \frac{\partial}{\partial r} (\frac{v_x^2 + v_r^2}{2})) \frac{\partial p}{\partial x}$

Tab.2 Relationships between boundary conditions and variables

Boundary Variable	Inlet	Outlet	Wall	Center axis
Velocity v_x	given	P.U.V.	zero	P.U.V.
Velocity v_r	zero	P.U.V.	zero	zero
Velocity v_θ	given	P.U.V.	zero	zero
Weight fraction Y_k	given	P.U.V.	P.U.V.	P.U.V.
Stream function ψ	integration of Eq.(3)	P.U.V.	const.	const.
Vorticity ω	Eq.(2)	P.U.V.	B.L.A.	estimated by use of values of two adjacent points
Temperature T	given	P.U.V.	P.U.V.	P.U.V.
Mass flow rate G	calculated by use of other values	calculated by use of other value	zero	calculated by use of other value
Pressure p	estimated by solving momentum eq.	estimated by solving momentum eq.	estimated by solving momentum eq.	estimated by solving momentum eq.
Density ρ	calculated from perfect gas law	calculated from perfect gas law	calculated from perfect gas law	calculated from perfect gas law

P.U.V. : estimated by using the values of two adjacent grid points in such a way that the parabola which passes through these points takes an ultimate value at the boundary grid point
 B.L.A. : obtained by calculating Eq.(1) under boundary layer approximation

derived from the following equation as a function Y_f , thereby a notation $Y_{f.st}$ is taken as a stoichiometric mixing ratio for the mixture composed of fuel and air : when Y_f is greater than or equal to $Y_{f.st}$

$$T_b = \frac{h_f^{(0)}}{c_p} \frac{Y_{f,sl}}{1-Y_{f,sl}} (1-Y_f) + T_{mix} \quad \left\{ \dots\dots\dots (7) \right.$$

when Y_f is less than that

$$T_b = \frac{h_f^{(0)}}{c_p} Y_f + T_{mix} \quad \left. \right\}$$

wherein, T_{mix} represents the mixing temperature expressed by the equation of

$$T_{mix} = \sum_k (c_p)_k Y_k T_k^{(0)} / c_p \quad \dots\dots\dots (8)$$

The present one uses T_b estimated by Eq.(7) as a starting value of temperature calculation in a reaction field. In the estimation of reaction rate, five chemical species are taken into consideration. They are fuel ($C_nH_mO_o = f$), O_2 , N_2 , H_2O and CO_2 , and are expressed hereinafter in a generic term k . The reaction rate w_k is expressed in a quadratic form against concentration, in such a way that it takes the maximum value in about stoichiometric composition. Thereby, the consumption of fuel and O_2 and the creation of H_2O and CO_2 are calculated. For instance, the consumption of fuel is calculated from:

$$w_f = k_0 T^{v_f} W_f \rho^2 \exp(-E/RT) \prod_k (Y_k/W_k)^{v_k} \quad \dots\dots\dots (9)$$

where, k_0 , v , W , R and E respectively represent the frequency factor, stoichiometric coefficient, molecular weight, universal gas constant and activation energy. The concentration term in the above equation is expressed as:

under a fuel rich condition

$$\prod_k \left(\frac{Y_k}{W_k} \right)^{v_k} = Y_f \cdot Y_{O_2} (Y_f + Y_{O_2}) / (W_f W_{O_2}) \quad \left\{ \right.$$

under an oxygen rich condition

$$\prod_k \left(\frac{Y_k}{W_k} \right)^{v_k} = Y_{O_2} \cdot Y_f (Y_f + Y_{O_2}) / (W_f W_{O_2}) \quad \left. \right\} \dots\dots\dots (10)$$

where, $Y_{f,s}$ and $Y_{O,s}$ respectively represent the stoichiometric concentrations assuming that there is no nitrogen component in the air.

Differential equations for Y_k in Tab.1 are solved by using the reaction rate expressed in Eqs.(9) and (10). For the combustion between single fuel species and air, once Y_k is decided, the combustion efficiency ξ and the oxygen excess ratio λ_{O_2} are derived from:

$$\left. \begin{aligned} Y_f &= \frac{(1-\xi)W_f}{B} \\ Y_{O_2} &= \frac{(\lambda_{O_2}-\xi)(n+m/4-o/2)W_{O_2}}{B} \end{aligned} \right\} \dots\dots\dots (11)$$

wherein, B is expressed as

$$\begin{aligned} B &= (1-\xi)W_f + (\lambda_{O_2}-\xi)\left(n+\frac{m}{4}-\frac{o}{2}\right)W_{O_2} \\ &+ \lambda_{O_2}\alpha_{N_2}\left(n+\frac{m}{4}-\frac{o}{2}\right)W_{N_2} \\ &+ \left[\lambda_{O_2}\alpha_{H_2O}\left(n+\frac{m}{4}-\frac{o}{2}\right) + \frac{m}{2}\xi\right]W_{H_2O} \\ &+ \left[\lambda_{O_2}\alpha_{CO_2}\left(n+\frac{m}{4}-\frac{o}{2}\right) + n\xi\right]W_{CO_2} \dots\dots\dots (12) \end{aligned}$$

and α_k represents the mole fraction of k component in the supplied air. Combustion temperature is determined as a numerical solution of the differential equation for T

in Tab.1 by the use of the obtained reaction rate.

3.2.3 Effective viscosity $\mu^{(e)}$

In the first place we outline the method and equation of effective viscosity comparing the present one with Gosman's one. The viscosity obtained empirically by Gosman is expressed as [12]

$$\mu^{(e)} = 0.012 d^{2/3} L^{-1/3} \rho^{2/3} \times \left[\sum_j \rho_j \nu_{sj} A_j (\bar{v}_{sj}^2 + \bar{v}_{sj}^2) \right]^{1/3} \dots\dots\dots (13)$$

where, d , L and A denote the diameter of combustion chamber, its length and cross-sectional area of inlet respectively; the subscript j indicates the numbers of inlets, the superscript $-$ means the cross-sectional mean value averaged over each inlet, and $\bar{}$ represents summing over all inlets. Using this equation the local change of $\mu^{(e)}$ is brought about only by the local change of the density ρ .

The present calculation uses the effective viscosity formula applying Prandtl's mixing length hypothesis ;

$$\begin{aligned} \mu^{(e)} &= \mu^{(1)} + \rho l^2 \left\{ \left(\frac{\partial v_x}{\partial r} \right)^2 + \left(\frac{\partial v_r}{\partial x} \right)^2 \right. \\ &\quad \left. + \left[r \frac{\partial}{\partial r} \left(\frac{v_\theta}{r} \right) \right]^2 \right\}^{1/2} \dots\dots\dots (14) \end{aligned}$$

Mixing length l is estimated by the combustion chamber and the minimum distance from the grid point to the wall surface are designated respectively by R and z , the mixing length is specified by

$$l/R = 0.14 - 0.08(z/R)^2 - 0.06(z/R)^4 \dots\dots\dots (15)$$

It has been confirmed that this equation is approximately applicable for the flow in the complicated shape flow field [13]. However, in using mixing length hypothesis arises a cumbersome problem that diffusion flux apparently stops. Pai recommends the "bridge" method as a counterplan against this problem [14], but his method is inconvenient since numerical values must be inspected in the course of computation. The present calculation proposes as an alternative counterplan the effective difference method for calculating the velocity gradient in the viscosity equation (14). For instance, with respect to the successive three-grid points (r', v_x') , (r'', v_x'') and (r''', v_x''') , the gradient is computed from the following equation

$$\left[\left(\frac{\partial v_x}{\partial r} \right)_{r=r''} \right]^2 = \frac{1}{2} \left[\left(\frac{v_x' - v_x''}{r' - r''} \right)^2 + \left(\frac{v_x'' - v_x'''}{r'' - r'''} \right)^2 \right] \dots\dots\dots (16)$$

3.3 Calculation procedures

Figure 1 represents the flow chart of the computer program and Tab.3 exhibits the principal values. After creating constant values and starting values, successive calculation is proceeded in the following three steps.

In the first step, non-combustion field is simulated, that is, the difference formulas respectively of the variables ψ , Y_k , rv_θ and Ω of the system of partial differential equations indicated in Eq.(1) and Tab.1 are solved on the assumption that reaction doesn't take place.

In the following step, the combustion field is computed on the assumption of adiabatic combustion. Using the results of the first step as the starting values of

this step, the other variables except temperature T are successively solved disregarding the creation and consumption of chemical species. The temperature is computed by Eq.(7) with the use of the concentration Y_k .

In the final step, the simulation of combustion states is completed under the assumption of over-all reaction. The calculation is then proceeded. Namely with respect to the distributions of T_b and Y_k solved at the previous step the temperature at the grid point where T_b is greater than ignition temperature T_{ign} is replaced by T_{ign} and the concentration at this grid is replaced by the value newly calculated considering that the temperature rise $T_{ign} - T_{mix}$ is brought about by fuel consumption and products creation. The temperature at the other grid points is replaced by T_{mix} . Taking these values as starting values of this step, all the variables shown in Tab.1 are solved by iterative calculation.

In each step the fractional change of variable at every iteration is calculated. It is then supposed that the computation converges when the summation of the absolute values of this fractional change over all the grid points decreases monotonically and becomes less than 0.002.

4. Results and considerations

The combustion chamber is composed of a straight pipe (radius $R = 0.04$ m, length $L = 0.21$ m). Air with swirl and fuel (commercial propane gas) are introduced from circular and ringed inlets partitioned by concentric pipes of 1 mm thick. Figure 2 shows the measured inlet-velocity profiles. In this figure, axial- and swirl-velocities within the radius from center line $r = 0$ to the wall $r = 40$ mm are respectively drawn by a solid and a dashed line. The types of velocity profiles are classified into 9

kinds from A to I according to the swirl strength and to the radial position of fuel inlet. Hereinafter, the experimental results are interpreted by use of the respective symbols also designated in the same figure. As input data of calculation these axial- and swirl-air flow velocities were used, but it was then assumed that the radial velocity is zero owing to its smallness.

Type A has only one inlet and is employed to decide the necessary number of grid divisions. It is previously ascertained that the combustion chamber of 0.04 m radius and 0.21 m length may be divided by the grid lines of 41×21 in order that the calculation

Fig.2 Distributions of inlet flow velocities and symbols of flow type

Tab.3 Principal values used in simulation

Density	$\rho(0)$	0.112	$\text{kg s}^2/\text{m}^4$
Pressure	$p(0)$	10332	kg/m^2
Laminar viscosity	$\mu(1)$	$1.86 \cdot 10^{-6}$	$\text{kg s}/\text{m}^2$
Temperature	$T(0)$	300	$^{\circ}\text{K}$
Specific heat	$c_p(0)$	0.24	$\text{kcal}/\text{kg}^{\circ}\text{C}$
Frequency factor	k_0	10^{12}	$\text{m}^4/\text{kcal s}^3 \text{ } ^{\circ}\text{K}^{1/2}$
Activation energy	E/R	24000	$^{\circ}\text{K}$
Prandtl number	Pr	0.7	
Schmidt number	Sc	0.9	
Fuel used	$\text{C}_3\text{H}_8\text{O}$, $n = 3$, $m = 8$, $o = 0$		

Tab.4 Convergence quality and applicability

(1) $|\delta\phi_{ij}/\phi_{ij}|_{\max}$ at the iteration number 60

	ψ	Ω	Y_f	$r v_{\theta}$
the present $\mu^{(e)}$	3×10^{-3}	6×10^{-2}	2×10^{-3}	4×10^{-4}
Gosman's $\mu^{(e)}$	4×10^{-3}	2×10^{-2}	6×10^{-3}	5×10^{-4}

(2) Measured and calculated values at the exit cross-section ($L = 0.21$ m) in non-combustion case

	$(v_x)_{\max}$ m/s	$(r) v_x = \max$ mm	$(v_{\theta})_{\max}$ m/s	$(r) v_{\theta} = \max$ mm
measured values	3.8 ± 0.5	36 ± 1.0	7.0 ± 1.0	31.5 ± 1.0
the present $\mu^{(e)}$	3.90	36.9	7.29	32.1
Gosman's $\mu^{(e)}$	4.47	32.0	5.72	28.0

Fig.1 Flow chart of computer program

results of the momentum at every cross-section may agree within 5 % error with the measured ones. Grid spacing is a non-uniform one and is decided in such a way that the mesh size nearest to the wall is 0.5 mm, but increases to about 1.5 times both with the radial distance from the pipe wall or with the longitudinal distance from the inlet.

Firstly the convergence quality of calculation is examined, since we use a more complex formula of effective viscosity than that of Gosman's. Fractional change $\delta\phi_{ij} /$

ϕ_{ij} of the variable ϕ_{ij} is calculated at every grid point. The maximum one $|\delta\phi_{ij} / \phi_{ij}|_{\max}$ among them is used as the index of convergence characteristics. The upper part of Tab.4 indicates a comparison of the index of the present equation with that of Gosman's both for calculations of non-combustion field. The lower part of the table shows a comparison of the calculated values obtained by both equations with the measured ones at maximum axial- and swirl-velocities and each location where these values become maximum. As to the convergence there seems no difference between the present equation and Gosman's one; however, as regards the coincidence with the measured values the present one is superior to Gosman's one. Hereinafter, we describe mainly the results of calculation by our equations and procedures.

Figures (3), (4) and (5) indicate the diffusion processes of fuel in the non-combustion state. The fuel gas diffuses rapidly within the longitudinal distance of several times the radius of fuel pipe, but diffuses gradually beyond this distance. The calculated results in the whole view (Fig.3) and the axial change of a half value width of fuel concentration (Fig.4) coincide well with the measured phenomena. In these figures, half value width is taken for the fuel supplying case from center axis as twice the distance from the center axis to the radial location where fuel concentration falls to the half value of the center concentration and for the case of supplying fuel from the ringed inlet as the radial distance between two points where the concentration on both sides of the maximum

Fig.3 Distributions of Y_f in non-combustion field (Type B)

concentration trace falls to the half value of the maximum concentration.

It can be seen from Fig.5 that the trace of maximum concentration for the fuel supply from center axis (type D) is shifted to the center axis, and contrarily, to the wall surface for type E.

These displacements may be respectively explained as a result of the phenomena that in the latter case a radial velocity toward the chamber wall is induced in the vicinity region of the fuel supplying port, whereas in the former case a radial velocity toward the center axis is induced and a recirculation flow region is created.

It can be seen from Fig.4 that in the case without swirl flow (types H and I) the diffusion rate of fuel jet, in other words, the spreading rate of fuel jet is small compared with the case with swirl flow (types B to G). In addition, the maximum diffusion is observed in the case of type D such that the fuel is supplied at the middle position between the center axis and the wall surface. Shifting the fuel supplying position toward either side of the center axis or the wall surface causes a decrease in fuel diffusion rate. Then, the rate becomes minimum in case of type I in which the fuel is supplied near the wall. With the object of examining this cause, in Fig.6 are illustrated the distributions of axial flow velocities v_x and effective viscosities for type D.

Fig.4 Change of half value width of fuel concentration

Fig.5 Trace of maximum concentration

From the examinations of Fig.6 together with Fig.4, the following conclusions can be drawn. The first cause that the diffusion rate in the type D becomes maximum compared with the other cases lies in that the viscosity, which is varied both by the existence of swirl flow and by distributing states of axial- and radial-flow velocities, becomes maximum between the center axis and the wall surface. The second cause is that a recirculating flow region is created in the vicinity of the center axis near the fuel supplying port as illustrated in Fig.6.

Figure 7 shows the swirl flow velocity in non-combustion state. The calculated profiles distinctly indicate the measured behavior that the swirl flow velocity close to wall surface decreases rapidly in downstream direction. Figure 8 compares the calculated combustion temperatures with the measured ones. It can be seen from the measured ones that the equi-temperature curves in the vicinity region of fuel port run in downstream direction and that a region of steep radial temperature gradient arises near the radial position $r = 0.02$ m. In the calculation results, these features become obscure and the width of equi-temperature

Fig.6 Distributions of axial flow velocities v_x and effective viscosities $\mu^{(e)}$ (Type D, calculated)

Fig.7 Distributions of swirl flow velocities (Non-combustion state, Type B)

curve becomes narrow over the chamber length, but in the other respects the calculated results coincide well with the measured ones. High equi-heat release lines are also drawn by two dashed lines. It is indicated that the reaction increases rapidly in the region downstream about the radius of fuel supply pipe, moreover that the high reaction lines extend as far as the outlet section in such a manner that those lines contain the maximum temperature curves. Figure 9 represents the results on the adiabatic temperature assumption. A region of steep temperature gradient can be clearly seen from this figure, but the region is shifted more to inner side than that drawn in Fig.8, and the temperature in the vicinity of inlet section becomes quite high. As a result, the calculation gives a quite different feature from the measurement. The reason why such a difference appears between the calculated and measured results lies in the assumption of the infinite reaction rate at which the combustion temperature was correlated in a fixed relation with the fuel concentration as was expressed in Fig.7. In an actual diffusion flame the fuel supplied doesn't burn out promptly when it encounters the air, but it must remain unburned up to a certain downstream distance after it is supplied. The characteristics of the high-reaction curves drawn in the above figure obey this statement, and, indeed, this statement may become apparent from the comparison between the experimental results of the combustion efficiency which will be mentioned later and the calculated results under the over-all reaction assumption.

Fig.8 Distributions of combustion temperatures (Type B, dashed lines: high reaction rate)

Fig.9 Combustion temperature under the assumption of adiabatic combustion

Figure 10 shows the distribution of combustion efficiencies of types B and E computed on the over-all reaction assumption, and Fig.11 compares this result with the measured one. As against the fact that under the assumption of adiabatic combustion the efficiency in the fuel poor region becomes unity, Fig.11 shows that the efficiency becomes less than unity both in the poor and rich regions of fuel irrespective of the fuel supply types. Hence, it can be said that the consideration of reaction rate is indispensable to the simulation of turbulent diffusion flame.

In case of type D, a fairly high efficiency is observed in the vicinity of center axis owing to the creation of a recirculating flow. As compared with Fig.9, in case of type B a high-efficiency broad region is seen near the maximum temperature region. In cases of type D and F, the progress of reaction is delayed in the region near the wall surface owing to high fuel concentration and a cooling action of the wall and as the result, high efficiency is seen only in the region inside of the fuel supply position. In these respects the present calculation coincides qualitatively with the measurements.

5. Conclusions

In this report the initial diffusion- and combustion-processes are examined through the comparisons between the experimental and numerically calculated results obtained by changing such parameters as the fuel supply position, the magnitude of axial- and swirl-velocities and their distributions. With the object of simulating complex flow fields, expressions for effective viscosity, the equation (14), the effective difference method, the equation (15), and the calculation procedures of finite reaction rate and of combustion efficiency are proposed. The flow fields with and without combustion calculated by the present method are compared and examin-

Fig.10 Distributions of combustion efficiencies (Calculated)

ed both with that by Gosman's empirical formula and with the measured results.

Summarizing the results obtained the following conclusions are drawn :

The convergence characteristics in the use of equations (14) and (15) are in the same degree as those in the use of the empirical formula, the equation (13), but the former gives a more adequate approximation of the phenomena. It is also revealed that the assumption of adiabatic combustion temperature is hard to apply to the combustion fields as used in the present study, and that the calculation has to be carried out by taking into account the finite reaction rate and the inlet velocity distributions in order to understand more closely the phenomena.

Through the calculation and measurement the following conclusion which seems to be useful for the improvement of combustion method is drawn: In addition to the fact that fuel diffusion rate can be increased by supplying a swirling air flow in such a way that a recirculation region is created, the rate can be still more promoted by supplying the fuel into the region of high effective viscosity.

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Fig.11 Distributions of combustion efficiencies ($x = 0.15 \text{ m}$)

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