Smoke Layer Formation by Fires in Forced-Ventilation Enclosure

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ABSTRACT

Gas analysis and temperature measurement using a room with controllable heat sources reveal the condition of smoke layer formation in fires in forced-ventilation and highly airtight enclosures. Three combinations were used. 1) An inlet opening located in a lower part and an outlet at an upper level, 2) both an inlet and an outlet located on the ceiling, 3) An inlet at the upper level and an outlet at the lower. Methane combustion lasted for longer than 30 minutes in the first case with an air supply rate of 50m^3/h and a heat release rate of 100kW. In the other configurations, combustion lasted for much less than 30 minutes, and there was hardly any smoke layer formation. The stability of the smoke layer was also affected by the inlet air velocity. The mechanism of the smoke layer destruction due to increased inlet velocity was also investigated using CFD results.

KEYWORDS: Enclosure Fires, Forced-Ventilation, Smoke Layer, Computational Fluid Dynamics

NOTATION

A_r area of floor (m^2)
A_{op} area of opening (m^2)
C_μ 0.09
G air supply rate (m^3/s)
H_{op} opening height (m)

H_r height of room (m)
k turbulent energy (m^2/s^2)
L mixing length scale around the border (m)
t time (s)
Y_o oxygen concentration of the outside fresh air (Vol m^3/m^3)
Y_l oxygen concentration of the lower air layer (Vol m^3/m^3)
Y_u oxygen concentration of the upper smoke layer (Vol m^3/m^3)
z thickness of the smoke layer (m)
X direction along the inlet velocity
Y vertical direction
Z lateral direction
ε dissipation of turbulent energy (m^2/m^3)
ν_t kinetic eddy viscosity (m^2/s), $\nu_t = C_\mu k^2/\varepsilon$
ϕ diameter
Γ exchange coefficient indicating the volume transfer between the smoke layer and air layer, $\Gamma = A_r(H_r-z)(1/L)\nu_t(1/L)$ (m^3/s)
Ψ parameter for evaluating the stability of smoke layer (-)

INTRODUCTION

Most studies on enclosure fires have been carried out assuming natural ventilation. The characteristics of these fires are predicted using an opening factor of $A_{op}\sqrt{H_{op}}$ [1, 2, 3]. Recently, buildings have become more airtight. Unless the breakage of glass and siding occurs, or unless open windows or doors exist, rooms are kept in a highly airtight condition. In such situations, the opening factor theory cannot be applied to predicting the fire characteristics.

Besides, pressured smoke exhaust is becoming common, so ventilation would continue even after a fire breaks out. Experimental studies focused on the temperature profiles of fires in forced-ventilation enclosures were reported by Alvares and Backovsky [4, 5]. Zukoski et al. [6] described the combustion processes in conditions with a smoke layer and an air layer. Bayler [7] analyzed the temperature distributions and proposed extinction mechanisms observed in fires in enclosures with and without overhead forced-ventilation.

The border between the two layers may become unstable when the temperature of the smoke layer is not higher than that of the lower air layer, or when a significant convective flow enters the enclosure from the outside and disturbs the flowfield. An increase in the amount of smoke and a decrease in the oxygen gas concentration around the flame could produce carbon monoxide. The influence of forced-ventilation conditions on smoke layer formation by enclosure fires has to be studied for efficient exhaust and the prevention of the spread of toxic gases. Experiments with small-scale rooms were conducted [8]; here, full-scale enclosures are used for the above purpose.

EXPERIMENTAL DESCRIPTION

Figure 1 illustrates the facilities used for the experiments. They consist of a blower, a full-scale room with inlet and outlet openings, and a diffusion-type gas burner.
The experimental conditions were varied as follows.
1. Fuel: methane or propane.
2. Heat release rate: 50kW, 100kW, 150kW, 200kW (assuming perfect combustion).
3. Air supply rate through the inlet: from $50m^3/h$ to $700m^3/h$.

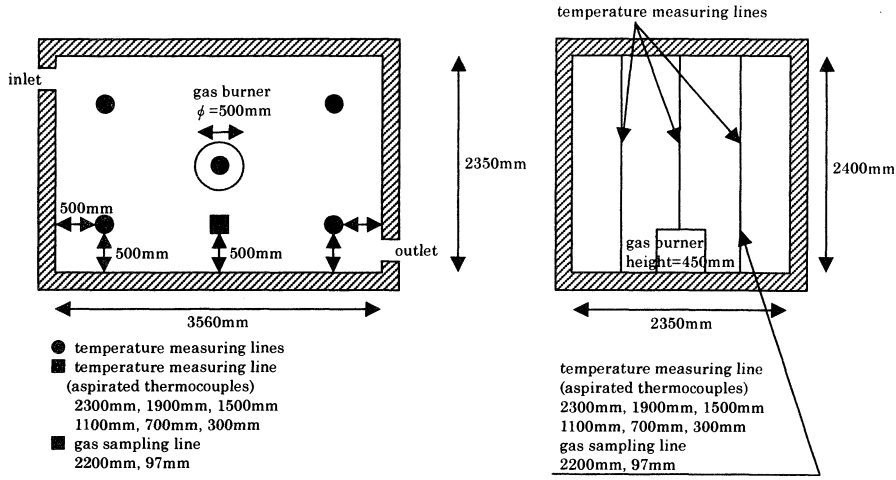


FIGURE 1 Model room (left side: plan, right side: cross section)

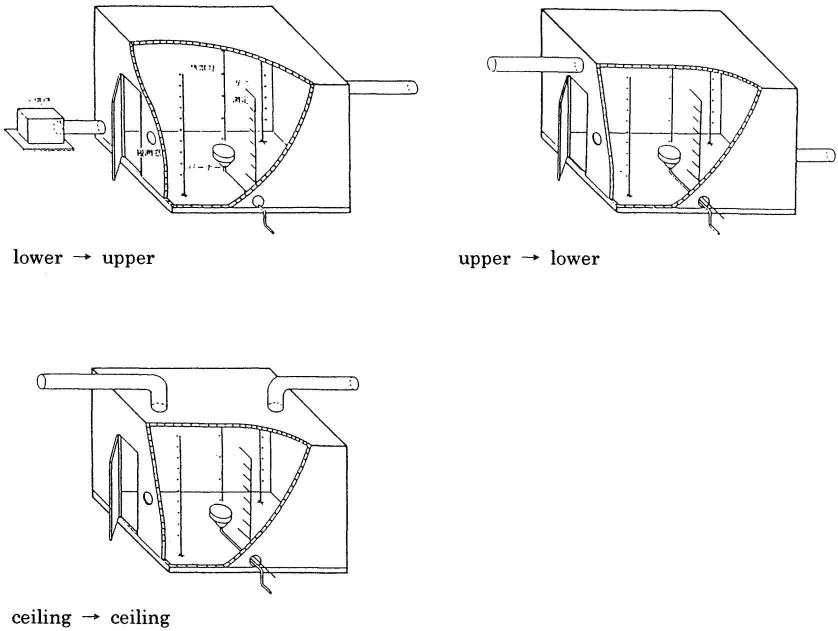


FIGURE 2 Combinations of inlet-outlet locations

4. Diameter of the inlet opening: 5cm, 10cm, 14cm, 20cm.
5. Velocity at the inlet: from 1m/s to 34m/s (determined by the air supply rate and the diameter of the inlet opening).
6. Combinations of inlet-outlet locations (Figure 2): lower inlet → upper outlet, ceiling inlet → ceiling outlet, and upper inlet → lower outlet.

EXPERIMENTAL RESULTS

Figures 3(a)-(d) through 5(a)-(d) show gas histories obtained from the experiments at a fire strength of 100kW by methane combustion and an air supply rate of $50m^3/h$. Figure 3: lower inlet → upper outlet, Figure 4: ceiling inlet → ceiling outlet, and Figure 5: upper inlet → lower outlet. Figures 3(a), 4(a) and 5(a) indicate the history of the gas temperature measured at the six points along the vertical line located in the middle between the gas burner and a wall (■ in the plan of Figure 1, each point's height above the floor level: 230cm, 190cm, 150cm, 110cm, 70cm, 30cm). Figures 3(b)-(d), 4(b)-(d) and 5(b)-(d) show consumption and production by combustion measured at the two points along the same line (2200mm and 97mm), and also those measured in the exhaust duct. Combustion lasted much longer in the case of the lower inlet → upper outlet (Figures 3(a)-(d)) in comparison with the ceiling inlet → ceiling outlet, and the upper inlet → lower outlet (Figures 4(a)-(d), and Figures 5(a)-(d), respectively). In those two configurations, meaningful combustion lasted less than 10 minutes. Thus, for fixed air and fuel supply rates (here, $50m^3/h$ and 100kW), the duration of combustion was dependent on the location of the inlet and outlet openings.

Figure 6 shows vertical profiles of temperature at 30 minutes after ignition. In the case of the lower inlet → upper outlet, the typical two-layer formation can be observed (●: $150m^3/h$ and 50kW, ○: $280m^3/h$ and 200kW). But with the ceiling inlet → ceiling outlet (▲: $150m^3/h$ and 50kW, △: $280m^3/h$ and 150kW, △: $280m^3/h$ and 200kW) and the upper inlet → lower outlet (■: $150m^3/h$ and 50kW), the difference in temperature between the upper and lower parts in the room is much smaller.

SMOKE LAYER FORMATION IN FORCED-VENTILATION ENCLOSURE FIRES

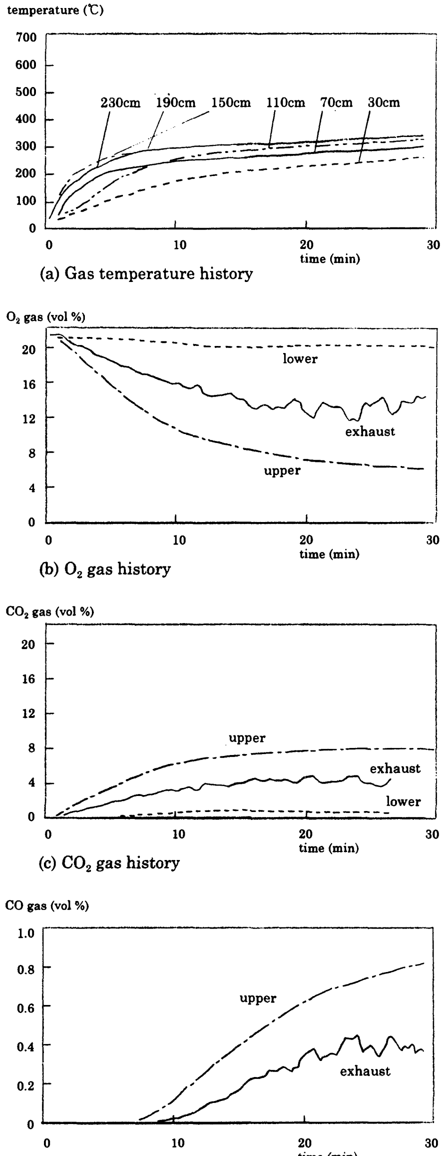
Prior to this discussion, the parameter for judging smoke layer formation should be defined. Figure 7 shows the general idea of the two-layer zone model. A difference in oxygen consumption between the upper and lower parts of the model is referred to for introducing the above parameter for smoke layer formation in forced-ventilation enclosure fires. The oxygen balance in the lower air layer can be described as follows.

$$A_r (H_r-z)(dY_l/dt)=G(Y_o-Y_l)+\Gamma (Y_u-Y_l) \qquad (1)$$

The left side of the equation means lower layer's oxygen volume transition per unit time. The 1st term on the right side of the equation is oxygen volume transition by net convection from outside fresh air and into upper layer per unit time, and the 2nd term on the right side of the equation is oxygen volume transition by net turbulent diffusion between the two layers per unit time.

Assuming a steady state, namely, that the left side of the equation (1) is equal to 0, the equation can be transformed into the following equation.

$$\Psi \equiv (Y_o-Y_l)/(Y_o-Y_u)=\Gamma/(G+\Gamma) \qquad (2)$$



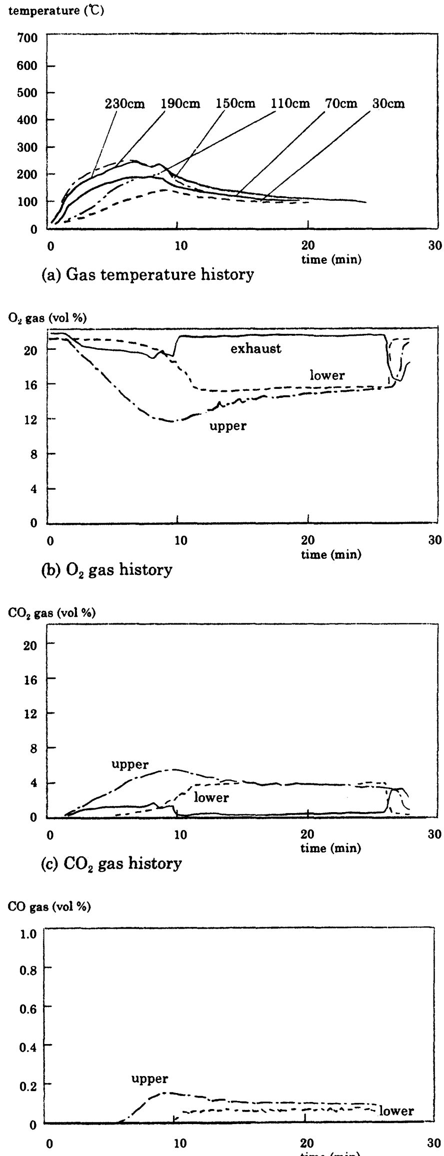
(a) Gas temperature history

(b) O_2 gas history

(c) CO_2 gas history

(d) CO gas history

FIGURE 3 Gas history (lower → upper)



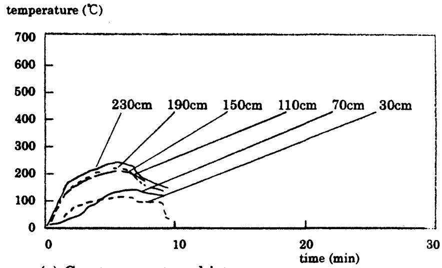
(a) Gas temperature history

(b) O_2 gas history

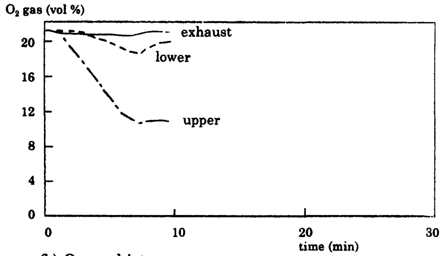
(c) CO_2 gas history

(d) CO gas history

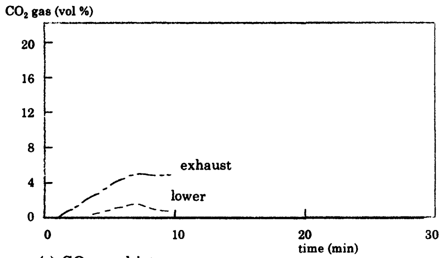
FIGURE 4 Gas history (ceiling → ceiling)



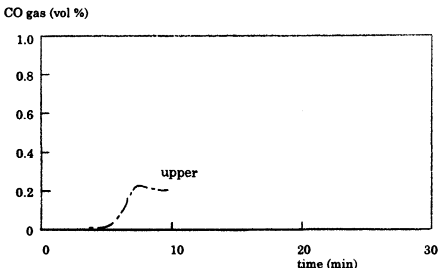
(a) Gas temperature history



(b) O_2 gas history



(c) CO_2 gas history



(d) CO gas history

FIGURE 5 Gas history (upper → lower)

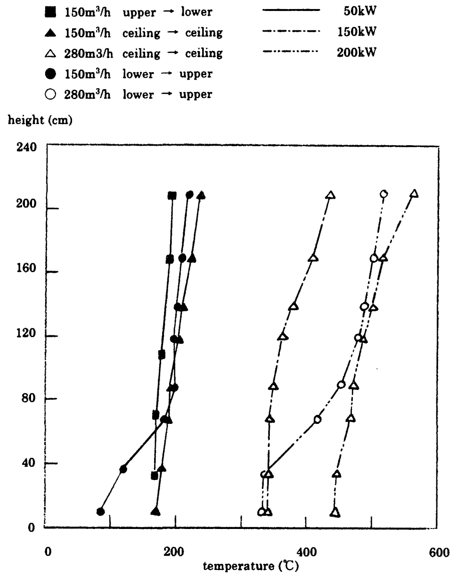


FIGURE 6 Temperature profiles

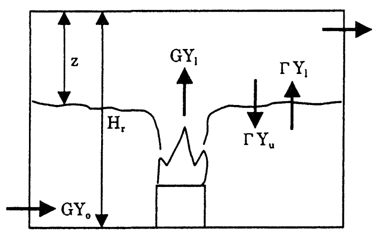


FIGURE 7 A general idea of two layer zone model

When a stable smoke layer is formed, substantial transfer by turbulent diffusion through the border between the two layers can be disregarded. Applying $G>\Gamma$ to (2) leads to $\Psi<0.5$. When the smoke layer becomes unstable, namely that the effect of the turbulent diffusion is dominant, the parameter Ψ must be close to 0.5 or even greater than 0.5 because of the larger value of Γ in comparison with the stable situation. The parameter Ψ can be used as a criterion for evaluating the formation of the smoke layer.

Figures 8(a)-(c) show the relation between the parameter Ψ and the air supply rate for various fire strengths for each combination of inlet-outlet locations. The turbulence within an enclosed space, but not in a fire room, is usually affected by the inlet air velocity [9]. In the case of a fire room, instability is also influenced by gas temperature distributions. Applying the knowledge to the forced-ventilation enclosure fire, the following must be said.
① Small inlet air velocity makes turbulence within the enclosure small.
② The large difference in temperature between the smoke and air layers results in the stable smoke layer formation.
When the heat release rate (fuel supply rate) and the diameter of the inlet opening are fixed, both phenomena described above are satisfied at a small air supply rate. The latter phenomenon can also be achieved by increasing the heat release rate.

Figure 8(a) shows that Ψ has a tendency to become smaller at smaller air supply rates. For a given air supply rate, Ψ did not change drastically with the fuel supply rate in case of the lower inlet → upper outlet (see, for example, ○: 100kW, △: 150kW and □: 200kW at an air supply rate of $400m^3/h$). Figure 9 shows the relation between the Ψ and the inlet air velocity in the same case. When the inlet air velocity is less than 6m/s, Ψ is very small. Ψ increases dramatically with inlet air velocities, when they are greater than 6m/s. At velocities around 15m/s, Ψ is nearly 0.4. This means that there was hardly any smoke layer formation. Under the current experimental conditions, it is considered that stable smoke layer is mainly formed by small inlet velocity.

Meanwhile, in the case of the ceiling inlet → ceiling outlet, Ψ fell from 0.8 to 0.6 when the air supply rate rose from $130m^3/h$ to 610 m^3/h as shown in Figure 8(b). With the upper inlet → lower outlet, Ψ remained around 0.9-1.0 (Figure 8(c)). There was hardly any formation of a smoke layer in both cases. Airflow within enclosure fires is easyily disturbed when air whose temperature is much lower than the gas temperature in the enclosure is supplied from its upper part.

MECHANISM OF DESTRUCTION OF SMOKE LAYER USING CFD ANALYSIS

Computational Fluid Dynamics (CFD) was used in order to investigate the destruction mechanism of the smoke layer due to the larger value of the inlet velocity. The main purpose of the present CFD analysis is not comparison with the experiments but investigating the above subject. The simulation of the enclosure fire has been performed using the code SOFIE developed by Cranfield University and other organizations in Europe. This code employs a general non-orthogonal coordinate system, with finite volume discretisation, and here upwind interpolation was used for all variables. The SIMPLEC pressure correction algorithm [10] is applied to the co-located velocities and pressure field. Turbulent closure is effected through a two-equation k- ε model incorporating buoyancy modifications. Eddy break-up [11] has been implemented as the combustion model with radiation exchange by the discrete transfer method [12].

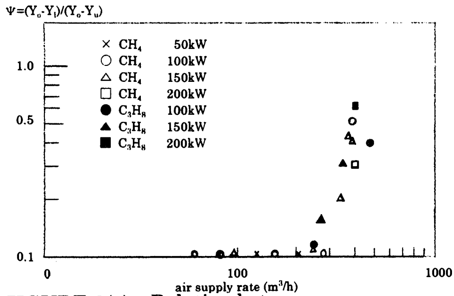


FIGURE 8(a) Relation between Ψ and air supply rate (lower → upper, ϕ =10cm)

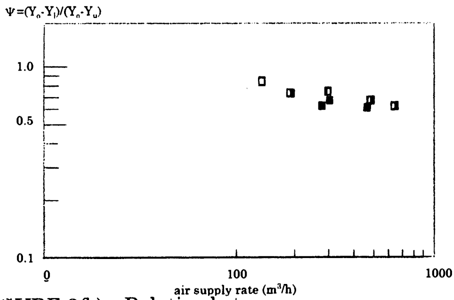


FIGURE 8(b) Relation between Ψ and air supply rate (ceiling → ceiling, CH_4, ϕ =20cm)

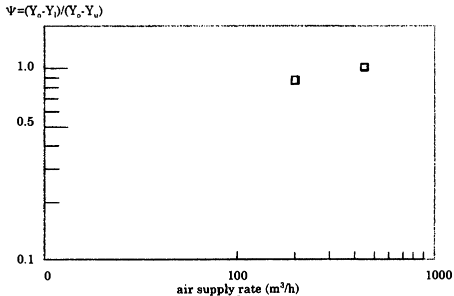


FIGURE 8(c) Relation between Ψ and air supply rate (upper → lower, CH_4, ϕ =20cm)

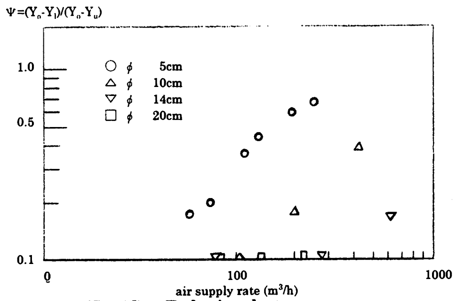


FIGURE 8(d) Relation between Ψ and air supply rate (lower → upper, CH_4, 50kW)

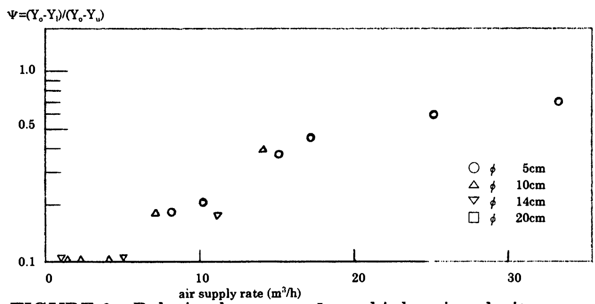


FIGURE 9 Relation between Ψ and inlet air velocity (lower → upper, CH_4, 50kW)

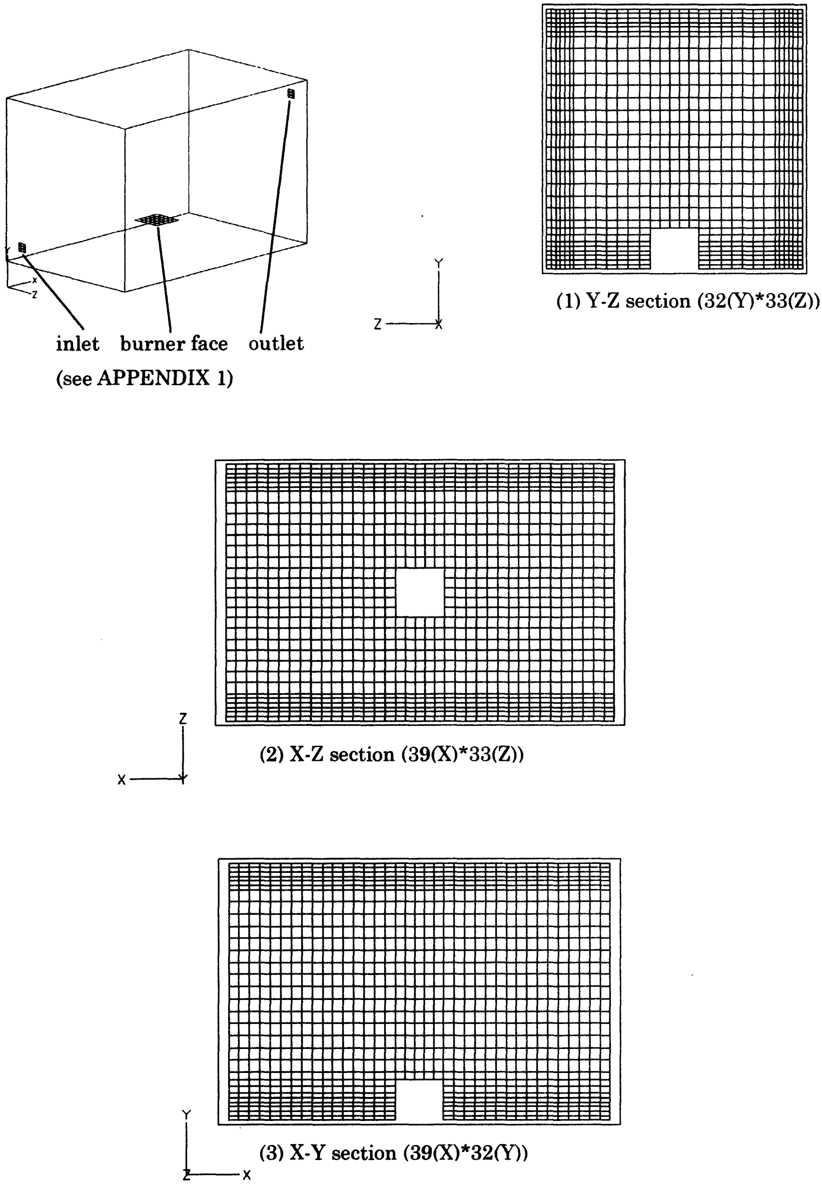


FIGURE 11 Computational grid used for CFD calculation

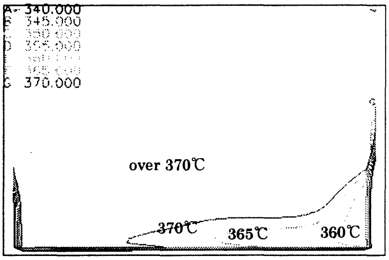


FIGURE 12 Temperature distribution (Z=1.8m, case 1)

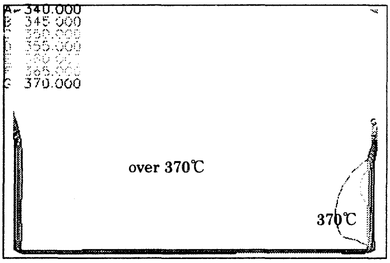


FIGURE 13 Temperature distribution (Z=1.8m, case 2)

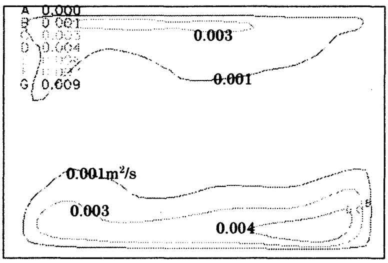


FIGURE 14 Distribution of kinetic eddy diffusivity (Z=1.8m, case 1)

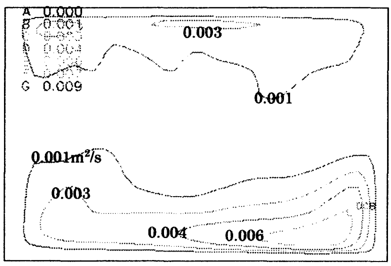


FIGURE 15 Distribution of kinetic eddy diffusivity (Z=1.8m, case 2)

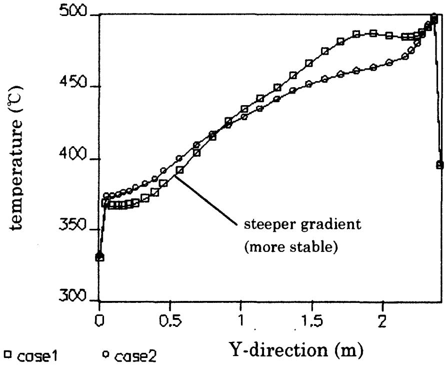


FIGURE 16 Temperature distribution (X=1.8m, Z=1.8m)

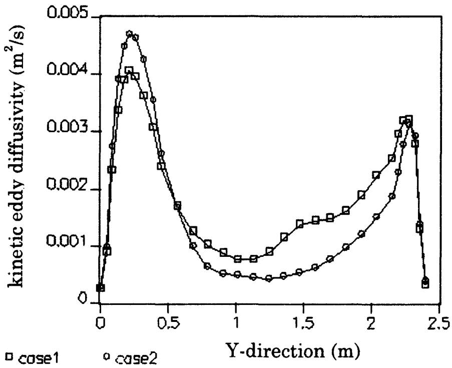


FIGURE 17 Distribution of kinetic eddy diffusivity (X=1.8m, Z=1.8m)

The computational domain covers 3.56m (X-direction), 2.4m (Y-direction) and 2.35m(Z-direction). This domain was discretized into 39(X)*32(Y)*33(Z)=41,184 meshes (Figure 11). For the inlet velocity, fixed values of 5m/s and 5.5m/s were set (respectively, case 1 and case 2). For the burner boundary condition, a fixed value of 1.3227m/s corresponding to 50kW combustion by CH_4 was used. For the outlet, all variables are set to zero-gradient. For the boundary conditions at the solid walls, the generalized logarithmic law [13] was employed in order to estimate the time-averaged wall shear stress.

Relaxation parameters for all variables are from 0.4 to 0.01, but 1.0 for enthalpy and 0.8 for pressure. A mass error residual level was 0.001 in all cases (APPENDIX 2). To achieve this degree of convergence, 10,000 iterations were run in the both cases. Using a PC with a Pentium 400MHz CPU, it took about 49,000 seconds.

Figures 13 shows the temperature distribution in the case 2. The area whose temperature is over 370℃ expands around the floor in comparison with case 1 (Figure 12). This increase around the floor can be also confirmed in Figure 16. The distribution of kinetic eddy viscosity in the case 2 is indicated in Figure 15. The value around the floor is larger than that in the case 1 (Figure 14). This can be also seen in Figure 17. This increase in kinetic eddy viscosity in case 2 is possibly caused by the increased inlet velocity [9]. The kinetic eddy viscosity around the floor in case 2 is 1.1-1.15 times larger than that in case 1. This value corresponds to the increase rate of the inlet velocity, namely 5.5/5.0 (5m/s in case1 and 5.5m/s in case 2).

CONCLUSIONS

From the experimental and computational results, the following conclusions can be drawn.
1. For given air and fuel supply rates, the duration of combustion is dependent on the inlet-outlet locations. Combustion lasts longer in the case of lower inlet → upper outlet than in the case of ceiling inlet → ceiling outlet and upper inlet → lower outlet.
2. In the case of lower inlet → upper outlet, the turbulence within the enclosure decreases with a small inlet air velocity. The smoke layer is formed under these conditions.
3. The smoke layer is destroyed by an inlet velocity larger than 6m/s.
4. The formation and destruction of the smoke layer were investigated using CFD results. Kinetic eddy viscosity becomes large in proportion to large values of inlet velocity. As a result, the smoke layer is destroyed by turbulent diffusion, and temperature distributions are more uniform in the whole fire room.

ACKNOWLEDGEMENTS

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APPENDIXES

1. Each shape of the inlet and outlet openings and the burner face here is not a circle but a square. But, each area corresponds to the area of the inlet opening (a circle, ϕ =14cm) and that of the burner face (a circle, ϕ =50cm) in the experiment.

2. The location of the inlet opening and its size affect the convergence level of the calculation. If there is a wide opening in the center line, the convergence level is e-04. Moving the inlet location by degrees from the center makes the convergence level worse.