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Development of Small-scale Swiss-roll Combustor and its Scale Effect

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Abstract

This paper reports the development of small scale (64-45mm) Swiss-roll combustors. The scale effect is examined by using simple isothermal and hydrodynamic models for flame stabilization. Results showed the importance of temperature distribution along the combustor surface plate on the burning rate and flame stabilization. Based on the analysis, three kinds of coin-size (diameter 26 mm) combustors were newly fabricated and their performance of flame stabilization was confirmed. This study suggests the possibility of much smaller combustors thorough further optimization of design parameters.

Keywords: Microcombustion, Microcombustor, Swissroll

1 INTRODUCTION

The high energy densities of fossil fuels strongly motivate the development of small combustors which can be applied for portable energy supply system [1, 2]. Larger heat loss due to the large surface-to-volume ratio in small devices makes flame stabilization difficult. To overcome this difficulty, heat recirculation, high pressure, catalytic reaction, etc. can be applied.

Swiss-roll combustor was introduced as one of the promising configurations to maximize the effect of heat recirculation [3, 4]. Even though several studies have been conducted on Swiss-roll-type combustors, successful operations have been only reported when the scale of the combustors was larger than the critical size (quenching distance) or when a catalytic material was used for flame stabilization [3, 5]. Our previous research [6] showed successful flame stabilization in Swiss-roll combustors with the channel width less than classical quenching distance. It is suggested that such small Swiss-roll combustors could be applied as heaters. Although some studies [7, 8] were conducted for examining the characteristics of heat-recirculated combustors, the scale effect of Swiss-roll combustor requires further investigations.

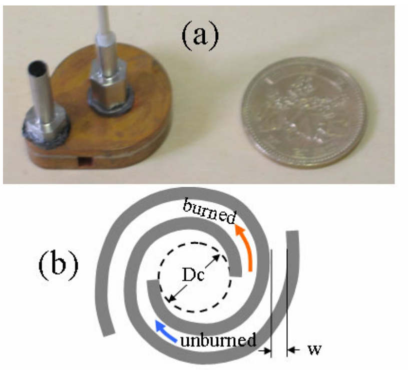

Fig. 1. (a) 500-Yen size combustor (CC-combustor in Table 1), (b) Configuration at the center.

In this study, subsequent to the developments of two kinds of Swiss-roll combustors (diameters 64 and 45mm), a simple one-dimensional analysis of Swiss-roll combustors with isothermal and hydrodynamic models of flame stabilization was introduced. Based on the analysis, three coin-size combustors were newly fabricated (one of them is shown in Fig. 1) and their characteristics were examined.

2 EXPERIMENTAL RESULTS

The specifications of eight different combustors used in this study are summarized in Table 1. W-, S-, and D-combustors (diameters 64mm) are reported in our previous study [7]. W-combustor has wider combustion space at the center. S- has a shallow channel and D- has a deep channel. Diameters of the three combustors were 64 mm, and the widths of the channels were 2 mm, which is slightly smaller than the quenching distance of propane/air mixture at a standard state (300 K, 1 atm).

Two kinds of smaller combustors (diameters 45mm) were fabricated to investigate the scale effect. One combustor was designed to have reduced diameter (noted as 'RD'), while the scale of channel was the same as that of the D-combustor. Another combustor was designed to have reduced scale (noted as 'RS'), 70 % of the D-combustor.

All the surfaces of combustors except upper surface were thermally insulated. Combustors were heated before ignition by supplying hot gas through the combustors. Then a flame was ignited by a spark-igniter installed at the center of the combustor. Pure propane and air mixture was used. Mean velocity was defined at the inlet of the combustor, in

Table 1 Detailed specifications of the combustors

Case	Do mm	H mm	Dc mm	Channel d	Channel w	Channel n	t_w mm	t_{cap} mm
W	64	16	12.7	6	2	3	1.5	5
S	64	16	3.5	6	2	4	1.5	5
D	64	27	3.5	15	2	4	1.5	6
RD	45	26	3.5	15	2	2	1.5	5
RS	46	20.5	2.5	10.5	1.4	4	1.05	5
CS1	26	6	3.3	3	1	2.5	0.75	1.5
CS2	26	9	3.3	4	1	2.5	0.75	4
CC	26	8	3.3	4	1	2.5	0.75	1(3)

W: wide combustion room, S:shallow combustor, D: deep combustor, RD: reduced diameter (70%), RS: reduced scale (70%), CS1 and CS2: coin-size combustor of stainless steel, CC: coin-size combustor of copper. Do: diameter of the combustor, H: height of the combustor, Dc: diameter of the combustion space, d: depth of channel, w: width of channel, n: turning number of inlet channel, t_w: wall thickness. All surfaces except upper side are insulated, Width of the inlet-part of the combustion room is half w, t_{cap}: thickness of the cap

which the cross sectional area was defined by depth and width (d and w in Table 1) of the channel.

Flammable conditions of the scale-down combustors (RD and RS) are shown in Fig. 2. The regime above each line corresponds to the flammable conditions in steady state. It is noted that the flammable regime of the RD-combustor having short channels was almost the same as that of the D-combustor (64mm). This fact implies that channel length of the RD-combustor is sufficient as a residence time for pre-heating. Another notable fact is that the flammable regime of the RS-combustor is wider than those of the S- or W-combustor in spite of smaller width of the channel. This implies that the cross-sectional area or the perimeter of the channel is important for flame stabilization. Large difference in flammable limits between the 'S' and 'W' combustors shown in Fig. 2 is also noted. This shows that the scale of the combustion space has significant effect on the flame stabilization.

Temperature distributions of the combustor surfaces in radial direction are compared in Fig. 3. Even though the mean velocity increases the mean temperature of the combustor, the temperature distributions show a similar trend if the center temperatures were similar. Thus the mean temperature of the combustor may have strong correlation with the heat loss and the chemical energy supplied to the combustors.

3 ANALYSIS

A simple one-dimensional model was made based on three assumptions. 1) Heat transfer from burned gas to the combustor surface plate occurs at the combustion space. 2) Heat loss from the combustor is governed by the radiant heat transfer from the combustor surface to the ambient at 300 K. 3) Temperature distribution of the combustor is not affected by that of the gas phase, since the total thermal

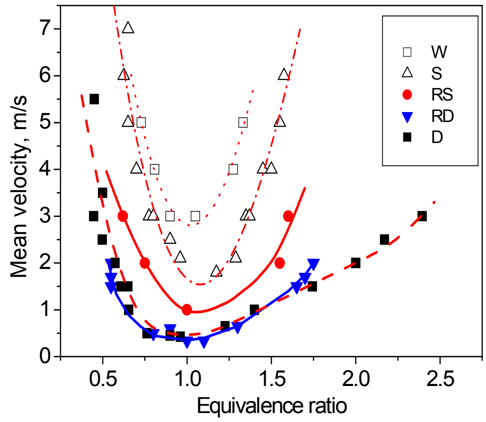

Fig.2. Flame stabilization conditions of various combustors

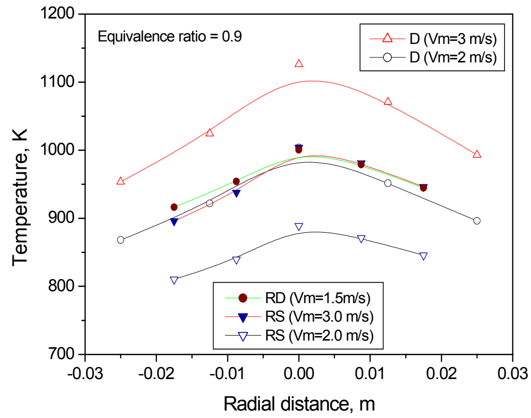

Fig. 3. Similarity in temperature distribution between different scale combustors

capacity of the combustor is much larger than that of the gas phase.

Two simple criteria of the flame stabilization are used. The first is an isothermal model in which temperature of the combustion space is higher than a critical temperature for stable reaction just like an ignition temperature of the mixture. The second is a hydrodynamic model in which burning rate of the mixture balances with the mass flow rate.

Isothermal model requires the value of the combustor temperature T_c, which could be solved independently. An energy balance in radial direction can be written as a summation of the heat flux through the surfaces of a finite control volume.

$$k_c\left(A_e \left.\frac{dT_c}{dr}\right|_e - A_w \left.\frac{dT_c}{dr}\right|_w\right) - A_n q''_{rad} + A_s q''_{comb} = 0\,, \qquad (1)$$

where k_c is thermal conductivity of the combustor, and A represents surface area of the control volume. Subscript e, w, n, and s depict four surfaces in radial or vertical direction, respectively. Radiant heat loss q''_{rad} and the heat flux from heat release by combustion q''_{comb} can be written as follows,

$$q''_{rad} = \varepsilon\sigma(T_c^4 - T_0^4), \qquad (2)$$

$$q''_{comb} = \dot{m}_F \Delta Q / (\pi D_c^2 / 4) \quad (\text{when } r < D_c/2), \qquad (3a)$$

$$q''_{comb} = 0 \quad (\text{when } r \geq D_c/2), \qquad (3b)$$

where $\dot{m}_F$ is propane mass flow rate and ΔQ is the low heating value of propane. Emissivity ε was assumed as unity [6]. Temperature distributions of the combustor in radial direction were shown in Fig. 4 for the representative cases. Where the effective thickness t in model was assumed as 11.5 mm (about two times of the t_{cap}). From the figure, we could find some facts as follows. 1) S (65*W*) and S (100*W*) cases show the agreement with the experimental results that higher flow rate increases the mean temperature. 2) S (65*W*) and W (65*W*) cases shows that the W-combustor requires higher flow rate for flame stabilization. If the ignition temperature is assumed as 900 K, W combustor cannot reach the ignition temperature even at the combustion space. 3) S (65*W*) and S (65*W*, t5mm) cases show the effect of conductive heat transfer in radial direction. Reduced conduction increases local temperature at the combustion space. Thus smaller flow rate is required for flame stabilization. 4) S (65*W*) and RS (65*W*) cases show that the RS-combustor has higher temperature for the same rate. Eventually, the isothermal model shows that higher input energy by large velocity and thinner surface plate are preferable for stable combustion.

Hydrodynamic model requires temperature distribution of the mixture along the stream. Due to the large thermal capacity of the combustor, the temperature of the mixture could be evaluated from the temperature of the combustor explicitly. Location of the channel could be written as a relation between radius and angle,

$$r(\theta) = a\theta + b \quad (0 \leq \theta \leq 2\pi N) \qquad (4)$$

where $a = (w + t_w)/\pi$ and $b = D_c/2$. The length of the channel was assumed as following,

$$dx = dr \quad (\text{when } r < a \text{ or } 2\pi aN + b < r), \qquad (5a)$$

$$dx = \sqrt{1 + (r/a)^2}\, dr \quad (\text{when } b \leq r \leq 2\pi aN + b). \qquad (5b)$$

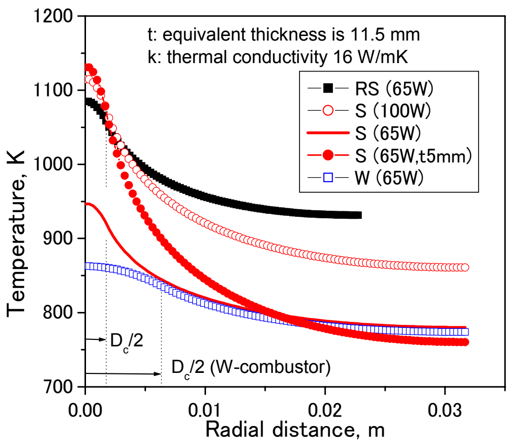

Fig. 4 Temperature distribution and flame stabilization estimated using a simple one-dimensional model

By integrating Eqs. (5a) and (5b), correlation between radius and the channel length was evaluated. Then the gas temperature could be evaluated from the following correlation concerned with the temperatures of the combustor.

$$\dot{m}c_p \nabla_x T - kd\nabla_x \cdot \left(w(x)\nabla_x T\right) - 2h(d + w(x))(T_c(x) - T) = 0 \qquad (6)$$

where $\dot{m}$ is mass flow rate through the channel, thermal conductivity k was assumed as 4.1×10^{-2} W/m/K at the 600 K, convective heat transfer coefficient was defined as $h = Nu_{D_h} k / D_h$, and the Nusselt number was assumed as 4. Hydraulic diameter was defined as $D_h = 2A_c/(w(x) + d)$. The width of stream tube $w(x)$ was constant in the channel while it varied at the combustion space. Maximum width of the stream tube was assumed as the diameter of the combustion space D_c or its half $D_c/2$.

If the total burning rate of the mixture is assumed, the location of the flame is estimated. There are many empirical models concerned with the burning velocity at high temperature. The burning rate in this study was evaluated using the burning velocity suggested by Metghalchi and Keck [9] and the ideal gas law as following,

$$\dot{m}_b = \rho V w(x) d = \rho_o S_L^o (T/298)^{1.13} w(x) d \qquad (7)$$

By matching Eq. (6) and Eq. (7), the flame position can be decided.

According to the hydrodynamic model, when the mass flow rate increases the flame moves downstream. However a flame could not be stabilized in downstream of the combustor since the temperature gradient becomes negative along the stream and blow-off occurs, finally. On the contrary, if the mass flow rate is small, flame propagates to upstream and quenched. Thus a flame can be stabilized between quenching limit at lower velocities and the blow off limit at higher velocities. Results will be shown in the next section.

4 DEVELOPMENT OF SMALLER COMBUSTORS

In this state, the diameter of a 500 JPY coin (26 mm) was chosen as a target-size. The width of the channel was fixed as 1.0 mm due to the limitation of the employed fabrication technique. Analytical model was applied to estimate the possibility of flame stabilization and to decide the specific parameters of the combustor. One example is the effect of the thickness of the cap (top and bottom plates) of the combustor. Fig. 5 shows the flammable limits against the thickness of the combustor cap in a given design of the combustor (corresponding to CS1, CS2, and CC combustor shown in Table. 1). Larger thickness of the cap makes blow-off limits at lower velocity. The effect of the hydrodynamic structure of the combustion space on the blow-off was also examined by choosing different values of D_c or $D_c/2$ as the maximum width of the stream tube.

Flame can be stabilized between blow-off and quenching limits. Large deviation depending on the maximum width of the stream tube implies sensitivity of the blow-off limits on the geometry of the combustion space. And slightly wider flammable region is expected when the cap thickness of the combustor is thinner.

Depending on the analytical model, three coin-size combustors (CS1, CS2, and CC) were made as shown in Table 1. The CS1- and CS2-combustors were made of stainless steel, while CC-combustor was made of copper.

The CS1-combustor has thinner cap. Thinner cap should be helpful to stabilize the flame according to the analysis. However, stabilizing a flame for CS1-combustor was failed here. When the cap of the combustor is thin, the temperature distribution in radial direction is strongly affected by the temperature of the gas phase in the channel. Thus thermal interaction may be more complicated depending on the configuration of the combustor.

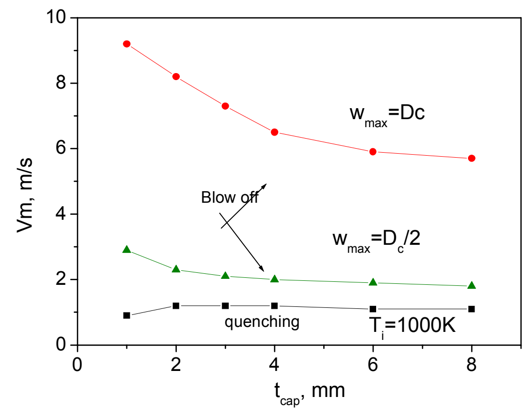

Fig. 5 The effect of the cap thickness of the combustor on flame stabilization for coin-size combustors

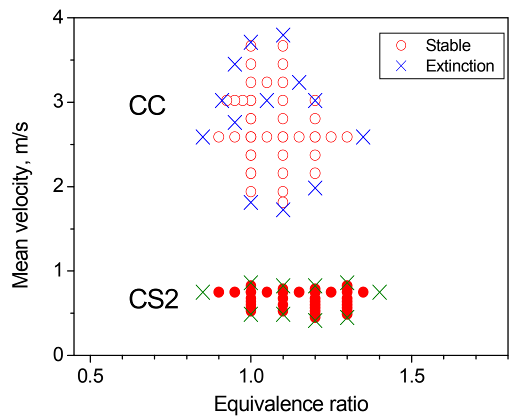

Fig. 6 Flame stabilization of coin-size combustors (CC and CS2)

Flame could be stabilized for two combustors having thicker caps (CS2 and CC), and their flammable limits were shown in Fig. 6. However significant difference with the larger combustors (D-, S- and W-combustors) is that there exists an upper limit above which flame is blown-off as expected from the hydrodynamic analytical model. The minimum velocity for flame stabilization in the CS2-combustor was about 0.5 m/s, which was smaller than that of the RS-combustor (in Fig. 2) in spite of its much smaller scale.

In the case of the CC-combustor, flame was stabilized at higher velocities. Since the thermal conductivity of copper (390 W/mK) is larger than that of stainless steel (16 W/mK), radial temperature distribution of the CC- combustor is flat compared with that of the CS2-combustor. From the viewpoint of the isothermal model, the CC-combustor requires higher input energy (or mixture flowrate) than that of the CS2-combustor.

5 CONCLUSIONS

Self-sustaining flames could be established in several small Swiss-roll combustors. Similarities in flammable limits and temperature distribution were observed. Based on the experimental results, a simple one-dimensional analytical model was suggested. The model explained experimental results clearly and predicted the characteristics of coin-size combustor. Based on the model results, three coin-size combustors were fabricated, and their flammable limits were examined. Flame could be successfully stabilized by two kinds of coin-size combustors and the blow-off limits at higher mixture velocities were observed.

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