Journal of Thermal Science and Technology

Vol. 6, No. 3, 2011

Development of Ultra-Micro Combustor Using Cylindrical Flames*

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Abstract
The purpose of this study is to develop an ultra-micro combustor that uses two types of coaxial cylindrical flames, rich premixed flame and diffusion flame. The combustor consists of inner and outer porous tubes, and rich propane-air mixture and air issued, respectively, through the inner tube outwardly and through the outer tube inwardly, forming a cylindrical stagnation plane sandwiched by the inner rich premixed flame and the outer diffusion flame. Petal type flame was also observed in the downstream of the cylindrical flames. Keeping the equivalence ratio φ_i and flow rate q_i of the rich mixture constant, air flow rate q_a was varied. The O_2 and CO concentrations and temperature of the burnt gas were measured, and heat loss rate η_{hl} and combustion intensity L were evaluated. The obtained results are described as follows. (1)The relation curve of η_{hl} with the overall equivalence ratio φ_{all}, which is evaluated from the total flow rate of the fuel and the air, has a minimum value. (2)The relation curve of the minimum value of η_{hl} with L has a minimum value. (3) The CO concentration of the burnt gas increases as q_a is increased because of local extinction of the petal type flame. (4)When q_a is increased further, petal type flame is also extinguished. After that, the O_2 concentration increases and the CO concentration decreases.

Key words: Combustion, Micro-Combustor, Heat Loss Rate, Combustion Intensity

1. Introduction

Recently, very small power-supply systems and very small propulsion systems called combustion type Power MEMS (Micro Electro Mechanical Systems) have attracted attention. Representative systems of this technology include the ultra micro gas turbine (UMGT) and the micro rotary engine [(1)]. Experimental studies related to UMGT combustors focusing on a premixed plane flame have been conducted domestically [(2)–(4)]. Downsizing of combustors has many problems such as the increase in heat loss rate. Downsizing of combustors and quenching distance are closely connected each other. In studies related to ultra-micro combustors, hydrogen has been mostly used as a fuel because it can shorten the extinction distance. In contrast, there are only a few studies that have used hydrocarbon fuels.

As a basic study of the local structure of a turbulent flame, we have made some studies of a counter flow type cylindrical flame that stretches in the axial direction [(5)–(8)]. In a study on cylindrical diffusion flames, we performed experiments using propane and methane as fuels, nitrogen as a diluent of fuel, and air as an oxidizer. Fuel flow was radially supplied from the central axis toward the surrounding area, and air flow was supplied from the surrounding area toward the central axis in order to form a cylindrical diffusion flame. This study clarifies that the flame can be strengthened with the decrease in flame diameter [(7),(8)].

*Received 29 July, 2011 (No. T2-10-0283)
Japanese Original : Trans. Jpn. Soc. Mech. Eng., Vol.76, No.772, B (2010), pp.2258-2263 (Received 9 Apr., 2010)
[DOI: 10.1299/jtst.6.406]

Application of this flame strengthening effect to the combustion in a very small domain is effective in downsizing the combustors. Therefore, we designed and developed a new-concept ultra-micro combustor which incorporated above-mentioned flow-field shape inside the combustor. The proposed combustor can form not only a single cylindrical diffusion flame but also a double flame consisting of a cylindrical premixed flame and a diffusion flame. We treated the latter case in this study. When the fuel-rich mixture is supplied radially along the central axis, a diffusion flame as well as a fuel-rich premixed flame is formed by the reaction between the extra fuel flowing out of the fuel-rich premixed flame and the air supplied from the surrounding area. Propane and air were used as a fuel and an oxidizer. Flame conditions were interpreted with photographs. In addition to the investigation of the flame stability, the temperature inside the combustion chamber and the oxygen and carbon monoxide concentrations of the burnt gas were measured. By calculating the heat loss rate and combustion intensity with the obtained data, we evaluated the combustion characteristics of the proposed combustor.

2. Experimental setup and procedure

2.1 Combustor

Figure 1 shows a schematic of the combustor. The combustor's outer tube is made of brass, and has outer diameter of 20 mm, length of 45 mm, inner diameter of 9 mm, and a depth of 39 mm. A stainless inner tube of 1.2 mm in diameter is installed along the same axis as the outer tube. On the inner side of the outer tube, twelve 0.6-mm openings were made circumferentially along one line; 11 such lines with openings were aligned at 2-mm intervals in the axial direction. On the inner tube side, eight 0.3-mm openings were made circumferentially on one line; 21 such lines with openings were aligned at 1-mm intervals in the axial direction. The combustion chamber capacity is 2.44 cm^3. By supplying air from the outer tube and a fuel-rich mixture from the inner tube, a double flame is formed: cylindrical fuel-rich premixed flame inside and cylindrical diffusion flame outside.

Cooling of a small combustor narrows the range of stable combustion conditions, resulting in the decrease in combustor outlet temperature. Nevertheless, a cooling system was installed to maintain the boundary conditions of the combustor's external side constant. In this experiment, the equivalence ratio of the mixture supplied from the inner tube φ_i and the mixture flow rate q_i were maintained constant (q_i=18.9 cm^3/s). Flame extinction limit was obtained by increasing the flow rate of air q_a from the outer tube. Here, the mixture flow velocity, which is defined as the supplied mixture flow rate q_i divided by the surface area of the 20-mm-long mixture tube including the mixture supply openings, is set to 25 cm/s. When the mixture flow velocity is low, the flame is easily extinguished because of the heat loss caused by the inner tube. According to our previous research on the extinction of a cylindrical diffusion flame [(8)], this flow velocity of 25 cm/s is the minimum value above which the effect of inner tube heat loss is negligible.

2.2 Measurements of temperature and combustion gas constituents

Thermocouple of Pt-Rh40%/Pt-Rh20%, 100 μm in wire diameter was used for the temperature measurement. Since the inside of the combustion chamber was narrow, the measurement space was limited, that is, when the center of the combustor outlet was set to the origin in the cylindrical coordinate system (r;z), the temperature was measured at the radius r = 2.5 mm, which is almost exactly the middle of the internal wall of the outer tube and the external wall of the inner tube, and in the axial distance range from z = 0 to z = −25 mm, downward from the combustor outlet surface. Measurements were made for the mixture equivalence ratio φ_i ranging from 1.5 to 5.5, by varying the air flow rate q_a.

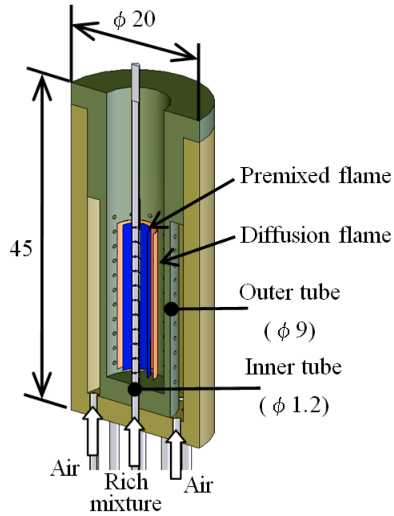


Fig. 1 Schematic of the micro-combustor

The burnt gas was analyzed with the exhaust combustion gas analyzer (HODAKA's HT-1300N) to measure the concentrations of oxygen and carbon monoxide. The tip of gas sucking probe was a stainless tube with an outer diameter of 1.2 mm and an inner diameter of 0.8 mm. The tube was cooled with water at the position 47.5 mm away from the tip. The gas suction was made at z = 0 mm and r = 2.5 mm. Analysis of burnt gas was conducted for four different equivalence ratios, φ_i=1.5, 2.5, 3.5, and 4.5, with various values of q_a.

2.3 Heat loss rate and combustion intensity

The heat loss rate η_{hl} is defined by the following formula (1) [4].

$$\eta_{hl} = \frac{H_i - H_o}{Q} \approx 1 - \frac{(m_f + m_a)c_p(T_o - T_i)}{Q} \tag{1}$$

Here, H_i and H_o are the enthalpy flow rates at the combustor inlet and outlet, respectively, Q is the heat generation rate, m_f and m_a are the mass flow rates of fuel and air, respectively, c_p is the average specific heat at constant pressure, and T_i and T_o are the temperatures of the combustor inlet and outlet, respectively. T_i was set to 293 K. Above formula can be applied when the distributions of flow velocity and temperature are uniform. In this study, however, to study the change in heat loss rate qualitatively, it was assumed that flow velocities and temperatures varied uniformly, and T_o measured at r = 2.5 mm was set as the representative outlet temperature.

The combustion intensity L is defined by the following formula (2).

$$L = \frac{m_f H_f}{V} \tag{2}$$

Here, H_f is the calorific value of the fuel, and V is the combustion chamber capacity.

3. Experimental results and discussion

3.1 Air flow rate at extinction and flame conditions

In each case of this experiment, the extinction experiment was conducted after the formation of a cylindrical flame was confirmed. The reasons are as follows: In many cases, a petal-type flame is formed in the downstream side of the cylindrical flame, and this cylindrical flame serves as a flame holder for the petal-type flame. Although the petal-type flame is formed even when there is no cylindrical flame, the stable combustion region narrows considerably compared with the case when a cylindrical flame is formed. Therefore, the cylindrical flame is required to expand the stable combustion region. Figure 2 shows the variation of $q_{a,ext}$ with the equivalence ratio of mixture φ_i, where $q_{a,ext}$ denotes

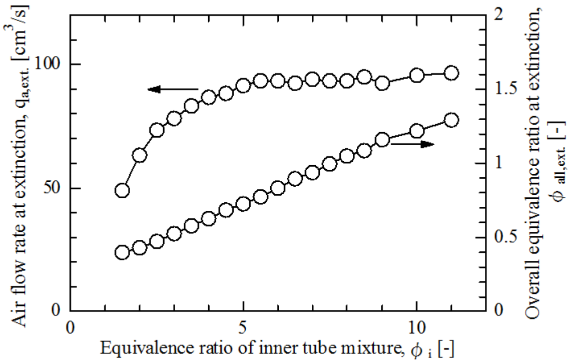
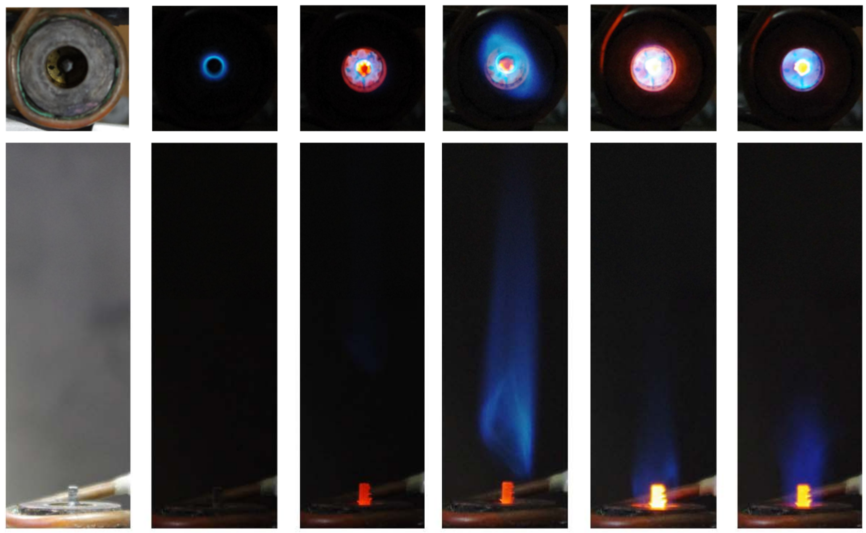


Fig. 2 Air flow rate and overall equivalence ratio at extinction

Top view

Side view



No flame

(a) ϕ_i=2.5, q_a=13.3cm^3/s

(b) ϕ_i=2.5, q_a=16.7cm^3/s

(c) ϕ_i=4.5, q_a=30.0cm^3/s

(d) ϕ_i=4.5, q_a=50.0cm^3/s

(e) ϕ_i=4.5, q_a=76.7cm^3/s

Fig. 3 Flame images

the air flow rate q_a at the complete extinction. The mixture supply rate is kept constant at q_i = 18.9 cm^3/s. This figure also shows the dependence of the equivalence ratio $\varphi_{all,ext.}$ on φ_i, where φ_{all} is the overall equivalence ratio calculated from the total flow rates of fuel and air supplied to the combustor and $\varphi_{all,ext}$ is the value of φ_{all} at the extinction. According to this figure, $q_{a,ext.}$ and $\varphi_{all,ext.}$ increase as φ_i is increased. The equivalence ratio at the lean flammability limit of propane-air mixture is 0.51; however, in this combustion method, $\varphi_{all,ext.}$ becomes less than the lean flammability limit in the range of φ_i = 1.5–2.5. Regarding the extinction pattern observed in this experiment, it is noted for φ_i less than 6 that the flame existing in the combustion chamber is finally extinguished. In contrast, for φ_i more than 6, a flame formed above the combustion chamber outlet is blown out, resulting in the complete extinction. Since the purpose of this study is to develop a combustor that can form a stable flame within a very small area, the results for the cases of φ_i less than and equal to 5.5 are described hereafter.

The flame condition varies depending on φ_i and the air flow rate q_a provided from the outer tube. Figure 4 shows the flame condition map where the combination of φ_i and φ_{all} is changed. Here, since the experiment is conducted with a constant value of φ_i, the decrease in φ_{all} implies the increase in q_a. As shown in Figure 3, the flame combustion states are classified into five states, (a) to (e). Each state corresponds to the following symbol in Figure 4, ○, ●, ▲, ■, and ▽, respectively. Each image in Figure 3 is explained as follows. Image (a): Only the cylindrical flame is formed in the combustion chamber. Image

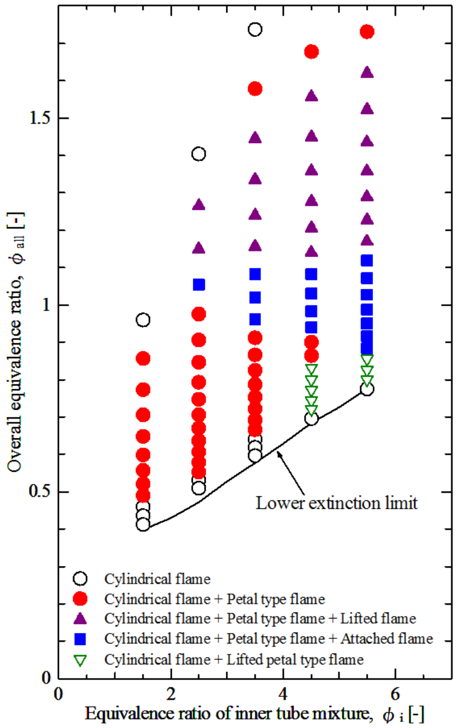


Fig. 4 Combustion pattern

(b): Petal-type flame is formed in the downstream side of the cylindrical flame, but no flame is formed outside the combustion chamber. Image (c): In addition to the cylindrical flame and the petal-type flame formed within the combustor, a lifted flame can be observed above the combustor outlet. This flame can be observed mainly when φ_{all} is rich in fuel. Image (d): Lifted flame observed in image (c) adheres to the combustor outlet. In image (e): Petal flame formed in the combustor detaches from the cylindrical flame due to the increase in q_a, and lifts above the combustor. The combustion state patterns change from (a) through (e) as the air flow rate q_a increases. When q_a is further increased, the lifted petal-type flame is extinguished first and then the cylindrical flame is also extinguished, meaning the occurrence of complete extinction. Figure 4 indicates that as φ_i decreases, the number of experimental points of the flame being observed outside the combustor decreases, and therefore it can be expected that combustion is almost completed within the combustion chamber. In addition, if φ_i is less than 4.5, there exists the area in the lean region of φ_{all} where flames can be formed only within the combustion chamber.

3.2 Temperature distribution in the combustion chamber

The temperature was measured at the lower positions than the combustion chamber outlet ranging from z = 0 to z = −25 mm at r = 2.5 mm. Under the experimental conditions, the radius of the cylindrical flame was smaller than 2.5 mm, and therefore the temperature measurement positions were outside the cylindrical flame. Figures 5 (a) and (b) show the axial distribution of temperature for the equivalence ratios of the mixture φ_i being 2.5 and 4.5, respectively. It is shown in this figure that the temperature T increases as the thermocouple moves from z=0 towards the flame, and it can exceed the adiabatic flame temperature T_{ad} of the mixture with the overall equivalence ratio φ_{all}. In this experiment, the temperature T inside the combustion chamber became higher than T_{ad} for φ_i = 2.5–4.5 in the φ_{all} range of 0.6–0.7. As φ_i is significantly rich in fuel, a premixed flame is not formed usually, and even if the flame is formed, it would be extremely weak. For this reason, the existence of an outside diffusion flame contributes significantly to the formation of the stable premixed flame observed in this experiment. No coating to prevent catalytic reaction was applied onto the thermocouple used for temperature measurement; therefore the effect

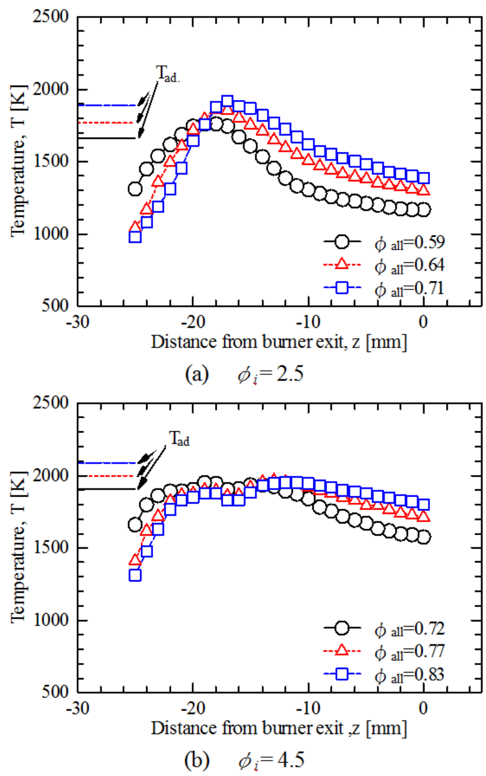


Fig. 5 Temperature distributions in the micro-combustor

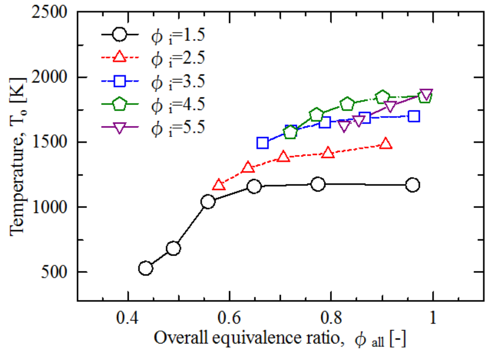


Fig. 6 Temperature at the burner exit

of catalytic reaction could also be included in the measurement results. However, when it is considered that a diffusion flame is stabilized under the conditions of nearly stoichiometric ratio and that a petal flame is formed in the downstream side of the cylindrical flame, there might exist a region where T exceeds T_{ad} even if the effect of catalytic reaction is excluded. In the temperature distribution of Figure 5 (b), a minimum value seems to exist around $z = -16$ mm. According to Figure 4, the combustion condition when the minimum appears corresponds to the combustion state when the petal-type flame formed in the combustion chamber detaches from the cylindrical flame and lifts above it. Therefore, the location of the minimum temperature can be considered to correspond to the location where the cylindrical flame and the petal-type flame are separated.

Figure 6 shows the relationship between the combustor outlet temperature T_o and φ_{all} with φ_i as a parameter. This figure indicates that the temperature increases as φ_{all} is increased, but its gradient becomes moderate. For φ_i larger than 3.5, T_o exceeds 1500 K. As shown in Figure 5 (a), T_o for φ_i = 2.5 exceeds 1500 K in the region fairly away from the combustion outlet. Therefore, when φ_i is small, the combustor can also be downsized in the axial direction.

3.3 Heat loss rate and combustion intensity

Figure 7 shows the variation of the heat loss rate η_{hl} with the overall equivalence ratio φ_{all}. The temperature measured at the combustor outlet (r = 2.5 mm, z = 0 mm) was used as T_o in formula (1) to calculate η_{hl}. According to this figure, there exists a minimum value on the η_{hl} curve when φ_{all} is varied. The combustion state is likely to vary depending on whether φ_{all} is larger or not than φ_{all}^*, the value of φ_{all} at which η_{hl} takes the minimum. First, we explain why η_{hl} decreases as φ_{all} is made leaner from stoichiometric ratio. In this study, the equivalence ratio φ_i and the mixture flow rate q_i were maintained constant, and φ_{all} was decreased by increasing the air flow rate q_a. Hence, the decrease in φ_{all} increases the second term on the right-hand side of formula (1), leading to the decrease in η_{hl}. Next we describe why η_{hl} increases as φ_{all} is made so leaner as to exceed φ_{all}^*. Regarding the flame observed in this situation, local extinction occurs in the petal-type flame or the lifted petal-type flame formed in the combustion chamber due to a further increase in q_a. Therefore, when φ_{all} is made leaner than φ_{all}^*, the region of incomplete combustion caused by local extinction expands and increases η_{hl}. It is noted that η_{hl} for φ_i = 1.5 is higher than that for other values of φ_i in the entire area of φ_{all}. It is also clearly shown by formula (1) that when the value of φ_{all} is fixed, q_a decreases with the decrease in φ_i, resulting in the increase in η_{hl}.

Figure 8 indicates the variation of $\eta_{hl,min}$ with the combustion intensity L, where $\eta_{hl,min}$ is the minimum value of η_{hl} at the designated value of φ_i shown in the figure. It is found that as L increases, $\eta_{hl,min}$ decreases first and takes the minimum at L corresponding to φ_i around 4.0, and then increases. The increase of $\eta_{hl,min}$ at the maximum combustion intensity of φ_i= 5.5 might be explained by the combustion state shown in Figure 4 and the combustor outlet temperature T_o shown in Figure 6, which implies that the combustion reaction does not terminate in the combustion chamber and a flame is formed outside the combustor.

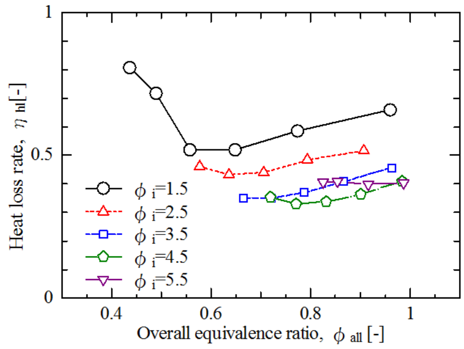


Fig. 7 Relationship between heat loss rate and overall equivalence ratio in each equivalence ratio of inner tube mixture

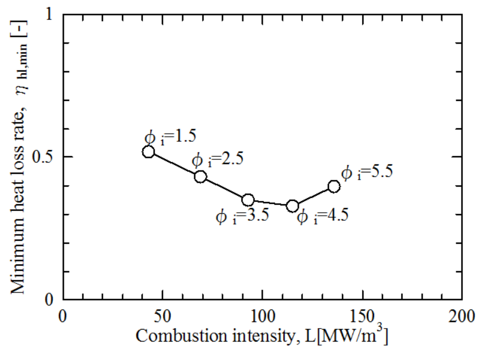
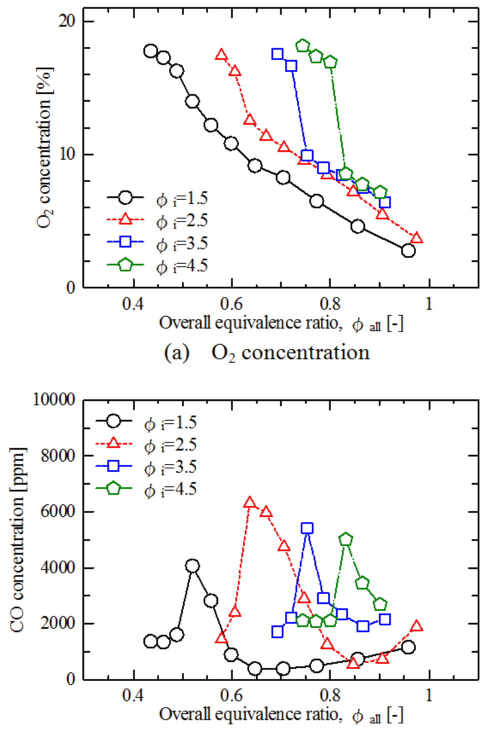


Fig. 8 Relationship between minimum heat loss rate and combustion intensity in each equivalence ratio of inner tube mixture



(a) O_2 concentration

(b) CO concentration

Fig. 9 O_2 and CO concentrations at the burner exit

3.4 Burnt gas composition

Figures 9 (a) and (b) show the concentration variations of O_2 and CO, respectively, with the overall equivalence ratio $\varphi_{all.}$

According to Figure 9 (a), when the curves are compared at the same value of φ_{all}, the O_2 concentration increases as φ_i increases, indicating that the consumption rate of oxygen contained in the air supplied from the outer tube decreases as φ_i increases. Noticing the trend of variation of O_2 concentration, there exists an inflection point on each curve. The inflection points are situated at φ_{all} around 0.52, 0.63, 0.73, 0.82 for φ_i of 1.5, 2.5, 3.5, 4.5, respectively. These values of φ_{all} are in accordance with those at which the heat loss rate η_{hl} takes the minimum value, as is shown in Figure 7.

Variation of CO concentration with φ_{all} shown in Figure 9(b) is examined next. Concerning a single premixed flame, it is generally known that the CO concentration reduces monotonously as the mixture equivalence ratio decreases. However, the results obtained in this study are not so simple, i.e., for all values of φ_i, the CO concentration reduces first as φ_{all} is decreased from the stoichiometric ratio, and then it reaches the minimum value at a certain value of φ_{all}. When φ_{all} is further decreased, the CO concentration suddenly increases once and after taking the maximum value, it reduces again. The fact that the CO concentration reduces as φ_{all} is decreased from the stoichiometric ratio corresponds qualitatively with the characteristics of an ordinary premixed flame, i.e., the CO concentration reduces as the equivalence ratio is decreased. Concerning the phenomenon of sudden increase of CO concentration, incomplete combustion might be related to this. As mentioned earlier, however, the decrease in φ_{all} means the increase in the air flow rate q_a. Therefore, there is a possibility also that the reactant residence time reduced due to the increase in q_a and CO was not completely oxidized. The value of φ_{all} at which the CO concentration reaches the maximum value and begins to decrease again is nearly accordance with that at which the inflection point appears on the CO concentration curve. Hence, it is likely that in this situation the unburned fuel was discharged, resulting in a reduction in CO concentration. Finally, the following two points are considered: the minimum value of CO concentration increases as φ_i is increased, and the value of φ_{all} at which the CO concentration becomes minimum increases as φ_i is increased. As mentioned earlier, if the value of φ_{all} is fixed, q_a is larger when φ_i takes a larger value and the reactant residence time in the combustion chamber reduces. Therefore, CO is not completely oxidized before it reaches the combustor outlet. As a result, the minimum value of CO concentration increases as φ_i is increased and the value of φ_{all} at which the CO concentration takes the minimum value also increases.

In this study, experiments were performed using the combustor with a cooling system to keep the same boundary conditions of the external side of the combustor. We consider that even better combustion characteristics will be realized by removing the cooling system to improve the heat loss rate and conducting combustion under the condition in which the CO concentration reaches the minimum value as indicated in Figure 9.

4. Conclusions

We designed and created an ultra-micro combustor using the stagnation-flow field of the cylindrical coordinate system. We performed experiments by supplying fuel-rich mixture from the inner tube and air from the outer tube in order to form a cylindrical fuel-rich premixed flame and diffusion flame on the outer side of the premixed flame. Propane was used as the fuel and air as the oxidizer. The findings obtained are summarized as follows:

(1) Depending on the supply flow rates of fuel and air, combustion can be performed even when the equivalence ratio φ_{all}, which is calculated from the total flow rates of fuel and air supplied to the combustor, is below the lean flammability limit.

(2) Depending on the equivalence ratio φ_i of the mixture supplied from the inner tube and the air flow rate q_a supplied from the outer tube, the combustion chamber temperature T could exceed the adiabatic flame temperature of the mixture that corresponds to φ_{all}.

(3) The heat loss rate η_{hl} takes a minimum value for φ_{all}. In addition, a minimum value exists on the curve plotting the minimum value of η_{hl} at each value of φ_i against the combustion intensity L.

(4) The concentration of O_2 increases as φ_{all} decreases. However, when the local extinction of the petal-type flame formed in the downstream side of the cylindrical flame becomes remarkable, the O_2 concentration significantly increases.

(5) The CO concentration decreases as φ_{all} is reduced. However, when φ_{all} becomes leaner beyond a certain point, the CO concentration begins to increase and reaches a maximum value, and then decreases again.

(6) The minimum value of CO concentration at each value of φ_i increases as φ_i is increased, and φ_{all} at which the CO concentration takes a minimum value also increases.

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