Lean-Premixed Combustion of Kerosene Using Porous-Media Vaporizer for 100 W-class Miniature Gas Turbine Combustor

Takashi Sakurai¹, Naoyuki Takayama¹, Yusuke Owada¹, and Ryo Harada¹

¹Department of Aeronautics and Astronautics
Tokyo Metropolitan University
6-6 Asahigaoka, Hino, Tokyo 191-0065, JAPAN

ABSTRACT

A miniature combustor using kerosene fuel for a 100 W-class micro gas turbine was developed. The unique feature of this combustor is the use of porous media as a liquid-fuel vaporizer. The porous media was heated by combustion, causing the vaporization of the liquid fuel in the miniature combustor and the mixing of the vaporized fuel with air, which resulted in lean-premixed combustion. Experimental results indicated that the flame was stable including the design operation condition and that the combustion efficiency was >99.5%. The NOx emission was low as <15 ppm. The orifice located in the combustion chamber had significant effects on the flame stability and combustion efficiency. The effects of this orifice on the flame stability and the emissions were examined. The proposed miniature combustor using the porous-media vaporizer is suitable for practical use in the 100 W-class micro gas turbine.

INTRODUCTION

As a power source for robots and UAV drones, the 100 W-class micro gas turbine is more attractive than Li-ion batteries and fuel cells owing to its higher energy density (U/kg) and power density (W/kg). In 2012, IHI Corp. developed a several hundred watt-class micro gas turbine generator [1, 2]. For this micro gas turbine, we developed a miniature combustor, which uses propane as a fuel because of its ease of gasification and mixing with air [3]. However, the use of kerosene or diesel fuel is preferable from the viewpoint of fuel storability.

For liquid-fuel combustion in gas turbines, pressurized or air-assisted atomization is generally applied. However, these methods are not suitable for our miniature combustor, owing to its small size. Because the typical dimensions, i.e., diameter and length, of our miniature combustor are ≤50 mm, the atomized fuel would impinge on the combustor wall before vaporization. This was confirmed by Amagai et al. [4], whose combustor had almost the same dimensions as our combustor. Therefore, it is difficult to apply conventional atomization methods to the present miniature combustor.

We proposed applying porous media for fuel vaporization [3]. Figure 1 shows the principle of the fuel vaporization method using a porous media. The liquid fuel and a small amount of air are supplied to the porous media. Once the flame forms on the porous media, heat is transferred from the flame to the porous media, and the temperature of the media increases, resulting in the vaporization of liquid fuel. Simultaneously, the vaporized fuel mixes with air in the porous media and forms a premixed flame.

We consider that this porous-media vaporizer is suitable for the miniature combustor owing to its scale effect. As the combustor size decreases, the surface area-to-volume ratio (S/V) increases. Because the heat convection between the porous media and the fuel is the dominant process for the fuel vaporization, the amount of vaporization depends on the surface area of the porous media. Therefore, the porous-media vaporizer is an effective vaporizing and mixing device for liquid fuel for the miniature combustor.

Studies on combustion using porous media were reviewed by Abdul Mujeebu et al. [5, 6]. To our knowledge, there have been few studies involving the application of porous media to miniature combustors for the vaporization of liquid fuels. Agrawal et al. applied porous media as a flame holder for the combustion of kerosene in a miniature combustor [7]. In their study, the kerosene was atomized and mixed with air by using a flow-blurring injector. Stable and efficient combustion was obtained; however, the NOx emission was high (200 ppm) because of the high burned gas temperature. Dunn-Rankin et al. proposed liquid-fuel film combustion using porous media for a miniature combustor [8, 9]. Although several porous-media shapes and combustion characteristics were studied, the combustion efficiency and emissions were not evaluated. There is a lack of knowledge regarding the applicability of porous media in miniature combustors for liquid-fuel vaporizers. Additionally, their combustion characteristics have not been clarified.

In this study, the applicability of the porous-media vaporizer for the lean-premixed combustion of kerosene for a 100 W-class micro gas turbine combustor was investigated. An annular-type miniature combustor having a volume of 139 cm³ and a size of 65.45 × 70 mm² was designed. The occurrence of kerosene vaporization using the porous media was experimentally verified. The flame behavior, flame stability, and combustion efficiency and emissions were evaluated.

Fig. 1 Schematic of the proposed fuel vaporization method using a porous media
COMBUSTOR AND EXPERIMENTAL CONDITIONS

Miniature micro gas turbine

The miniature micro gas turbine, which is the research target of this study, is palmtop size and has a generator output power of several hundreds of watts, as shown in Fig. 2. The design operating conditions of this micro gas turbine are a pressure ratio of 3, an air mass flow rate of 12 g/s, and a rated shaft speed of 580,000 rpm. This miniature micro gas turbine was developed by the joint research group of IHI, Tohoku University; other universities, including ours; and companies, and its self-sustained operation and generation of electrical power have been demonstrated [2].

The aim of the present study was to apply the kerosene-fueled miniature combustor to the aforementioned miniature micro gas turbine. The miniature combustor was designed to be reverse flow-type and annular-shaped. The design operating conditions were as follows: a pressure of 304 kPa, an air mass flow rate of 12 g/s, and an overall equivalence ratio ($\phi$) of 0.26, which corresponds to a turbine inlet temperature of 1,123 K. The research targets were as follows: 1) the volume of the combustion chamber was ≤100 cm$^3$; 2) under the design operating condition, the CO and NOx emissions were <100 ppm and <30 ppm, respectively, with a stable flame and high combustion efficiency.

Miniature combustor

Figures 3 and 4 show schematics and a photograph, respectively, of the miniature combustor. The combustor is composed of a combustion chamber and a premixing chamber. An orifice plate is located between these chambers. The combustion chamber has a diameter of 54.5 mm and a height of 50 mm. In the center, there is a shaft with a diameter of 20 mm, which models the exhaust tube, as shown in Fig. 2. The volume of the combustion chamber is 101 cm$^3$. To prevent external air from entering the combustion chamber, the chamber outlet is an orifice with a diameter of 30 mm. The premixing chamber has a diameter of 54.5 mm, a height of 19 mm, and a volume of 38 cm$^3$. This chamber is arranged to promote the mixing of the vaporized kerosene with combustor inlet air. The total volume of the combustor is 139 cm$^3$. The combustor is designed to be compact for satisfying the target requirements.

The combustor inlet air is tangentially supplied to the premixing chamber through four slits, as shown in section B-B' of Fig. 3. The air slits are rectangular and have a width of 3 mm and a height of 5 mm. This air forms the swirl flow both in the premixing chamber and in the combustion chamber and improves the flame stability. The combustor inlet air velocity at the slit exit ($U_a$) is called the reference air velocity in the flame stability limit.

Kerosene is supplied to the combustion chamber after passing through the porous-media vaporizer, as shown in section A-A' of Fig. 3. To form the premixture of vaporized kerosene and air in the porous-media vaporizer, a small amount of air (with a flow rate of 10 L/min at the normal temperature and pressure) is supplied with kerosene, and this air is called air for premixing. It is noted that the flow rate ratio of fuel to air for premixing was fuel-rich condition. Thus, further mixing of this rich premixture with combustor inlet air is necessary to form the lean-premixed flame. The porous-media vaporizer is cylindrical, with an inner radius of 54.5 mm and a thickness of 3.75 mm, as shown in Fig. 5. For the porous media, the
sintered porous ceramic FA100 (Fuji Chemical Industries, Ltd.) is used, which is mainly composed of Al2O3 and has an average pore diameter of 230 µm and a porosity ranging from 45% to 50%.

The orifice plate and most of the combustor are made of brass. In the outer wall of the combustion chamber, a quartz tube is used to observe the flame. In a previous study, the orifice plate was effective for fuel/air mixing and flame holding [3]. The orifice contraction ratio (CR) is defined as

\[
CR \equiv 1 - \frac{A_{\text{orifice}}}{A_{\text{chamber}}}
\]

where \(A_{\text{orifice}}\) represents the cross-sectional area of the orifice, and \(A_{\text{chamber}}\) represents the cross-sectional area of the premixing chamber. In this study, four orifice plates with CRs of 0.5, 0.6, 0.7, and 0.8 were used as the design parameters. Two K-type sheathed thermocouples, with outer diameters of 0.5 mm, were located in the combustor, as shown in Fig. 3, to measure the temperatures of the porous-media vaporizer and the chamber wall.

We intended to form the flame in the combustor, as shown in Fig. 6. The lower part of the flame is located near the porous-media vaporizer and transfers its heat to the vaporizer. The upper part of the flame forms along the recirculation zone at the backward-facing step (back-step) of the orifice plate. We designed the combustor to have a primary combustion region and a dilution region for obtaining a stable flame under an overall equivalence ratio of 0.26. It was assumed that 32.5% of air mass flow discharged from the compressor was divided to the primary combustion region and the other 67.5% of air to the dilution and the liner cooling of the combustor. Based on this assumption, the design operating conditions of the primary combustion region at atmospheric pressure and room temperature were an equivalence ratio of 0.8 and \(U_r=33.9\) m/s. Since the primary objective of this study was to demonstrate the lean-premixed combustion of kerosene using the porous-media vaporizer, the combustion characteristics in the primary combustion region were investigated, and dilution air was not supplied.

**Experimental conditions and method**

A schematic of the experimental setup is shown in Fig. 7. Air was supplied from the compressor, and the mass flow rates of the combustor inlet air and air for premixing were measured by mass flow meters. The uncertainties of the mass flow rates for the combustor inlet air and air for premixing were ±3% of the readout values. Kerosene was supplied to the combustor by using a fuel pump, and its mass flow rate was measured by a mass flow meter with uncertainty of ±4% of the readout value.

The experiment was conducted under the atmospheric pressure. Both air and kerosene were supplied to the combustor at room temperature. For ignition, a hydrogen pilot flame was inserted into the combustion chamber from the combustor outlet. For flame-stability measurement, the porous-media vaporizer was heated above 443 K, which is the initial boiling point of kerosene, using the pilot flame. For emission measurements, the burned gas was collected at the combustor outlet using a quartz probe with an inlet diameter of 1 mm. The burned gas was then cooled and dried, and the CO, total hydrocarbons (THC), and NOx concentrations were measured using a nondispersive infrared detector gas analyzer (Shimadzu TCD-7000), a flame-ionization THC detector (Shimadzu HCM-1B), and a chemiluminescence NOx detector (Yanco ECL-77A), respectively. It is noted that the NOx detector measured the sum of NO and NO2 concentrations. The measured ranges and uncertainties are presented in Table 1.

### Table 1 Measured ranges and uncertainties of the detectors for the emission measurements

| Detector | Range | Uncertainty |
|----------|-------|-------------|
| CO      | 0–1,000 ppm | ±2.0% |
| THC     | 0–100 ppm   | ±5 ppm    |
| NOx     | 0–50 ppm    | ±2.5%     |

**CALCULATION OF HEAT REQUIRED FOR FUEL VAPORIZATION WITH RESPECT TO HEAT RELEASED BY COMBUSTION**

In the proposed porous-media vaporizer, the heat required for fuel vaporization must be lower than heat released by combustion. Hence, these two heat quantities were estimated. Figure 8(a) shows the heat released by combustion for different air flow velocities and equivalence ratios, and Fig. 8(b) shows the heat required for fuel vaporization, which was calculated using the following equation:

\[
Q_{\text{evap}} = m_{\text{air}}c_{\text{air}}(T_{443} - T_{298}) + m_f c_f(T_{443} - T_{298}) + m_f L_f
\]

where \(m_{\text{air}}\) represents the mass of air, \(c_{\text{air}}\) represent the specific heat of air, \(m_f\) and \(c_f\) represent the mass flow rates of fuel, \(L_f = 250\) kJ/kg represents the latent heat of kerosene [10].

Comparison indicates that the heat required for vaporization is ≤10% of the heat released by combustion for different burning conditions. Hence, we considered that the vaporization of kerosene by using the porous-media vaporizer was calorically possible. Additionally, the transferred heat for vaporization is not wasted. The increased enthalpy of kerosene due to heat addition turns into the increase of the heat released by combustion. Then, the released heat is transferred to the porous-media vaporizer again. Thus, once the combustor attains the heat-equilibrium condition, recirculation of heat is established in the combustor. For effectively transferring the heat between the burned gas and the vaporizer, brass, which has a high thermal conductivity, was used as the combustor material, and the shape of the porous-media vaporizer was designed to be cylindrical.

The aforementioned heat recirculation occurs in the case of steady-state, and the situation differs at the start or in the transient phase of the gas turbine operation. In these cases, the porous-media vaporizer is not heated. When we performed the burning
Flame stability limit

The flame stability limit for different orifice CRs is shown in Fig. 11. The vertical axis indicates the total equivalence ratio, which was calculated using the fuel mass flow rate and the sum of the combustor-inlet-air and air-for-premixing mass flow rates. The plots show the flame blow-off point; above this point, the stable flame formed. The design operating point of the primary combustion zone is indicated by solid triangle. The dashed line shows the propagation limit of the kerosene–air premixed flame at the atmospheric pressure and 443 K, which was evaluated by calculating the burning velocity, as described later. This propagation limit is a rough criterion for judging whether the observed flame was a premixed flame.

EXPERIMENTAL RESULTS AND DISCUSSION

Transient burning behavior from ignition

Figure 9 shows the time histories of the temperatures for the porous-media vaporizer and the combustor wall after ignition. As mentioned in the experimental conditions and method section, a hydrogen pilot flame was used for ignition. Once the flame was sustained in the combustor, the pilot flame was removed, and the temperature of the porous-media vaporizer gradually increased owing to the heat released by combustion. The air mass flow rate and equivalence ratio were kept constant. The air flow velocity increased slightly because of the increase in temperature due to the heat transferred from the combustor wall. The temperatures of the vaporizer and the combustor wall were approximately 343 K at the start, because the flame stability measurement was performed before this experiment.

Figure 10 shows the characteristic flame appearances observed during the temperature increase of the vaporizer after ignition as shown in Fig. 10(a), the flame was blue. At times, a few luminous lines were observed, indicating suspended kerosene droplets. Because the vaporizer temperature was far below the boiling point of kerosene under this condition, most of the liquid fuel soaked out the vaporizer, and some of it was flown off by the swirling combustor inlet air. A little of the evaporated fuel burned at the back-step of the orifice plate as a blue flame. When the vaporizer temperature increased to 358 K, as shown in Fig. 10(b), the suspended fuel droplets were not apparent, and the flame appeared both in the premixing chamber and in the combustion chamber. Most of the flame color became yellow and luminous. It is considered that although the fuel temperature increased, the evaporation rate in the vaporizer was low and a large amount of fuel evaporated in the vicinity of the flame region in the premixing chamber. Consequently, mixing of fuel and air was poor and a luminous diffusion flame was formed. When the vaporizer temperature exceeded 380 K, the luminous flame was not observed, and a blue flame formed. Although the vaporizer temperature did not reach the initial boiling point of 443 K, the vaporizer temperature on the inner side was expected to increase more significantly than the measured value, because the thermocouple was located on the vaporizer outer side. Therefore, it is considered that most of the fuel was vaporized in the vaporizer and mixed with the not only air for premixing but also combustor inlet air in the premixing chamber, resulting in the blue flame.

In this experiment, the preheating time of the porous-media vaporizer was approximately 500 s. Because this combustor is for experiments and has a high heat capacity, the modified design for the fast preheating of the vaporizer is an issue for practical use.
Stable combustion, including the design operating point, was achieved for all the orifice CRs. The equivalence ratios of the flame blow-off points increased with the air flow velocity. A larger orifice CR yielded better flame stability for \( U_a \geq 20 \text{ m/s} \). For \( U_a < 20 \text{ m/s} \), the effect of the orifice CR on the flame stability was not apparent. The blow-off points, except for CR = 0.8, existed at an equivalence ratio lower than the propagation limit. This implies that the flames formed under these conditions were diffusion flames. On the other hand, for \( U_a \geq 20 \text{ m/s} \), it is considered that a premixed flame was formed.

The typical flames observed in the combustor are shown in Fig. 12 and Fig. 13 shows the schematics. In the case of CR = 0.8, a bright yellow and blue mixed flame, indicating the main reaction zone, occurred in the premixing chamber at \( U_a = 5.7 \text{ m/s} \). This flame is referred to as the orifice upstream flame. With increasing the combustor inlet air velocity, the flame occurred the transition of its position and shape between \( 5.7 \text{ m/s} < U_a < 8.5 \text{ m/s} \). For \( U_a \geq 8.5 \text{ m/s} \), an annular pale blue flame was located at the back-step of the orifice plate. This flame is referred to as the orifice downstream flame. Same kind of flame transition were observed for other CRs under identical combustor inlet air velocity.

The flame transition is associated with the quality of mixedness of kerosene-air mixture in the premixing chamber. For the orifice upstream flame, the non-uniform flame color along the circumferential direction in the premixing chamber was seen, which implied the low quality of mixedness. It is considered that this flame was formed by the yellow luminous diffusion flame in the premixing chamber due to fuel-rich condition of mixture. As the air velocity increased, the diffusion flame was blown off and the equivalence ratio of the premixture in the premixing chamber became fuel-lean condition. Then, the orifice downstream flame was anchored at the back-step of the orifice plate. Because this flame was a uniform blue flame and had the burning velocity, as shown in Fig. 11, it is considered that the orifice downstream flame had the same feature as the premixed flame.

It is considered that the stability of the orifice downstream flame was dominated by the recirculation zone of the burned gas formed at the back-step of the orifice plate, as schematically shown in Fig. 13. We evaluated the flame stability according to the Damköhler number [11–13]:

\[
Da = \frac{t_w}{\tau_c}, \tag{3}
\]

where \( t_w \) is the flow contact time of the premixture with burned gas in the recirculation zone at the orifice back-step, and \( \tau_c \) is the characteristic time for the chemical reaction. \( Da \leq 1 \) indicates that the flow contact time is shorter than the reaction time; thus, the ignition of the premixture cannot occur, and the flame is not sustained. Therefore, the flame stability criterion is expressed as \( Da > 1 \).

The air flow velocity for the flame blow-off limits was evaluated according to the blow-off limit for the flame holder in the premixed turbulent flow [14]. In this case, the Damköhler number is expressed as follows:

\[
Da = \frac{d}{U_b} \sqrt{\frac{\nu}{S_L}}, \tag{4}
\]

where \( t_w = d/U_b \) and \( \tau_c = \nu/S_L^2 \). Because the reference diameter \( d \) is related to the recirculation zone, the orifice CR, which is proportional to the area of the orifice back-step, is used as \( d \). Assuming that the blow-off occurs at \( Da = 1 \), the blow-off velocity is related to the parameters in the following equation:

\[
U_b \propto (CR \cdot S_L^2)/\nu, \tag{5}
\]

where \( U_b \) is the blow-off velocity, i.e., the axial air flow velocity at the orifice. This air flow velocity was calculated as the sum of the combustor-inlet-air and air-for-premixing mass flow rates. In Eq. (5), CR represents the orifice CR, \( \nu \) represents the kinematic viscosity of the premixture, and \( S_L \) represents the burning velocity for the one-dimensional laminar premixed flame computed by Cantera [15]. The Luche mechanism was employed for the chemical reaction of the kerosene–air mixture [16]. Figure 14 shows the computed burning velocity for different equivalence ratios and premixture temperatures. In Eq. (5), \( S_L \) was computed with a premixture temperature of 443.15 K. The propagation limit in Fig. 11 is based on this computed result.

The flame stability condition is shown in Fig. 15. The plots indicate the flame blow-off points shown in Fig. 11, and the lines represent the empirical correlation calculated using Eq. (5). The blow-off velocities estimated using the empirical correlation do not have meaning, because Eq. (5) was multiplied by an arbitrary coefficient for consistency with the experimental results; however, they show the same tendency for different parameters. The right
side for each plot and line corresponds to the stable flame, and the left side corresponds to the flame blow-off. As the air velocity at the orifice increased, a higher equivalence ratio (corresponding to a shorter reaction time) was needed to maintain the stable flame. The trends of the equivalence ratio and blow-off velocity in the plots exhibit good agreement with the empirical correlations. This implies that the stability of the orifice downstream flame was dominated by the recirculation zone at the orifice back-step. A larger orifice CR yielded better flame stability. It is considered that increasing the orifice CR increases the area of the orifice back-step, enlarging the recirculation zone and thus increasing the characteristic mixing time. This result verifies the empirical correlation (Eq. (5)).

**Combustion efficiency and emissions**

Figure 16 shows the emissions and combustion efficiency measured at the combustor outlet. The air flow velocity at the orifice \( U_{\text{orifice}} \) was the experimental parameter, and the equivalence ratio was 0.8. Under these conditions, a stable flame was formed for all the orifice CRs, and the design operating point was also included. As described in the previous section, the air flow velocity at the orifice corresponds to the characteristic time for the mixing of the premixture with the burned gas; thus, it is proportional to the residence time of the premixture in the reaction zone. An increase in the air flow velocity yields a decrease in the residence time.

The CO concentrations monotonicly increased with the increase of the air flow velocity for all the orifice CRs, indicating that the residence time of the premixture significantly influenced the completeness of the chemical reaction. A larger orifice CR yielded a lower CO concentration at the same air flow velocity. This is related to the size of the recirculation zone at the orifice back-step. An orifice with a larger CR forms a larger recirculation zone, resulting in a longer mixing time of the premixture for the reaction.

The measured THC data exhibit scattering, and the consistency of the experimental parameters is poor. As indicated by the data for CR = 0.8, the THC concentration showed the same relationship with the air flow velocity as the CO concentration.

The combustion efficiency was evaluated according to the CO and THC concentrations at the combustor outlet by using the following equation:

\[
\eta_c = 1 - \left( \frac{m_b + m_a}{m_f} \right) \left( \frac{X_{\text{CO}} \Delta H_{\text{CO}} + X_{\text{CH4}} \Delta H_{\text{CH4}} + X_{\text{H2}} \Delta H_{\text{H2}}}{\Delta H_{\text{B}} M_b} \right),
\]

where \( X_i \) represents the mole fraction, \( \Delta H_i \) is the heating value, and \( M_i \) represents the molecular weight. The subscripts \( f \), \( a \), and \( b \) indicate fuel, air, and burned gas, respectively. The heating value of the kerosene–air premixture was used for \( \Delta H_{\text{B}} \); thus, \( \Delta H_{\text{B}} = \Delta H_f \). Combustion efficiencies of >99.8% were attained for the different orifice CRs, except for CR = 0.5, under an air flow velocity of 4 m/s. For these burning conditions, the NO\textsubscript{x} concentrations were approximately 10 ppm, satisfying the target value of the combustor. As the air flow velocity increased above 4 m/s, the efficiencies gradually decreased owing to the shorter residence time for the reaction. Because the decrease of the combustion efficiency leads to the decrease of the flame and burned-gas temperatures, the NO\textsubscript{x} concentrations also decreased with the increase of the air flow velocity. For CR = 0.5, the diffusion flame formed, and high NO\textsubscript{x}...
Conclusions

A kerosene-fueled miniature combustor for a 100 W-class micro gas turbine was developed. The combustor had an annular shape with dimensions of $64.5 \times 70 \text{ mm}^2$ and a volume of 139 cm$^3$. To vaporize the liquid fuel and form the premixture, a cylindrical porous media was used. Experiments were performed to evaluate the applicability of the porous-media vaporizer and validate the concept of the combustor. According to the results, the following conclusions are drawn.

1) In the proposed combustor, the heat required for the vaporizer is supplied by the combustion via heat recirculation. Calculation results for the heat balance indicated that the heat required for vaporization was ≤10% of the heat released by combustion for different burning conditions. The vaporization of kerosene using the porous-media vaporizer was feasible from an energy viewpoint.

2) The vaporization of kerosene and the lean-premixed combustion using the porous-media vaporizer were experimentally demonstrated at a vaporizer temperature of $>380 \text{ K}$. The preheating time of the porous-media vaporizer was approximately 500 s.

3) The flame was stable for a wide range of combustor inlet air flow velocities and equivalence ratios, including the design operating point. Most of the observed orifice downstream flame was considered to be the premixed flame.

4) The flame stability of the orifice downstream flame was dominated by the recirculation zone formed at the orifice back-step. The stability criterion for the blow-off air velocity at the orifice was derived according to the Damköhler number. A larger orifice CR yielded better flame stability owing to the larger recirculation zone.

5) In the case where a stable orifice downstream flame was formed, low NOx emissions of approximately 10 ppm were attained, with satisfactory combustion efficiencies. The completeness of the combustion and the NOx emissions strongly depended on the residence time in the recirculation zone at the orifice back-step.

Acknowledgments

This research was supported by JSPS KAKENHI Grant Number JP26820373.

References

[1] https://www.ihi.co.jp/hi/all_news/2011/aeroengine_space_defense2012-2-16/index.html (in Japanese)
[2] Tanaka, S., Hikichi, K., Togo, S., Murayama, M., Hirose, Y., Sakurai, T., Yuaota, S., Teramoto, S., Niino, T., Morii, T., Esashi, M., Isomura, K., 2007, “World’s smallest gas turbine establishing Brayton cycle,” Proc. 7th International Workshop on Micro and Nanotechnology for Power Generation and Energy Conversion Applications, PowerMEMS 2007, Vol. 11, pp.359–362.
[3] Harada, R., Kobori, Y., Sakurai, T., 2015, “Development of a prevaporized combustor using a porous media for a kerosene-fueled micro gas turbine,” Proc. IGTC 2015, MoAMA 1.
[4] Amagai, K., Fukai, M., Kurogi, T., Murayama, M., 2011, “Fundamental characteristics of ultra-micro gas turbine combustor using liquid fuel,” Trans. Jpn. Soc. Mech. Eng. B., Vol. 77, No. 776, pp.911–915.
[5] Abdul Mujeebu, M., Abdullah, M.Z., Abu Bakar, M.Z., Mohamad, A.A., Abdullah, M.K., 2009, “A review of investigations on liquid fuel combustion in porous inert media,” Prog. Energy Combust. Sci., Vol. 35, pp.216–230.
[6] Abdul Mujeebu, M., Abdullah, M.Z., Abu Bakar, M.Z., Mohamad, A.A. Abdullah, M.K., 2009, “Applications of porous media combustion technology – A review,” Appl. Energ., Vol. 86, pp.1365–1375.
[7] Sadassivuni, V., Agrawal, A.K., 2009, “A novel meso-scale combustion system for operation with liquid fuels,” Proc. Combust. Inst., Vol. 32, No. 2, pp.3155–3162, doi:10.1016/j.proci.2008.06.039.
[8] Sirignano, W.A., Pham, T.K., Dunn-Rankin, D., 2002, “Miniature scale liquid-fuel film combustor,” Proc. Combust. Inst., Vol. 32, pp.925–931.
[9] Li, Y-H., Chao, Y-C., Amade, N.S., Dunn-Rankin, D., 2008, “Progress in miniature liquid film combustors: Double chamber and central porous fuel inlet designs,” Exp. Therm. Fluid Sci., Vol. 32, pp.1118–1131.
[10] CAMEO Chemicals. Database of Hazardous Materials. Available at: https://cameochemicals.noaa.gov/chemical/960. [Accessed 28 Dec. 2018].
[11] Glassman, I., Yetter, R.A., 2008, “Combustion,” 4th Ed., Academic Press, pp.240–247.
[12] Zukoski, E.E., Marble, F.E., 1955, “Combustion Research and Reviews,” Butterworth, London, p.167.
[13] Zukoski, E.E., Marble, F.E., 1956, “Experiments Concerning the Mechanism of Flame Blowoff from Bluff Bodies,” Proceedings of the Gas Dynamics Symposium Aerothermochemistry, Evanston, IL, pp.205–210.
[14] Dawson, J.R., Gordon, R.L., Kariuki, J., Mastrakos, E., Masri, A.R., Juddo, M., 2011, “Visualization of blow-off events in bluff-body stabilized turbulent premixed flames,” Proc. Combust. Inst., Vol. 33, pp.1559–1566.
[15] Goodwin, D.G., Moffat, H.K., Speth, R.L., 2017, “Cantera: An object-oriented software toolkit for chemical kinetics, thermodynamics, and transport processes.” http://www.cantera.org. Version 2.3.0. doi:10.5281/zenodo.170284
[16] Luche, J., Reuillon, M., Boettner, J.-C., Cathonnet, M., 2004, “Reduction of large detailed kinetic mechanisms: application to kerosene/air combustion,” Combust. Sci. Technol., Vol. 176, No.11, pp.1935–1963.