Available online at www.sciencedirect.com

ScienceDirect

Experimental Thermal and Fluid Science 32 (2008) 1118–1131

Experimental Thermal and Fluid Science

www.elsevier.com/locate/etfs

Progress in miniature liquid film combustors: Double chamber and central porous fuel inlet designs

Yueh-Heng Li [a], Yei-Chin Chao [a], Nicola Sarzi Amadé [b], Derek Dunn-Rankin [b,*]

[a] Institute of Aeronautics and Astronautics, National Cheng Kung University, Tainan, Taiwan, ROC

[b] Department of Mechanical and Aerospace Engineering, University of California at Irvine, Irvine, CA 92697, USA

Received 15 August 2007; received in revised form 21 December 2007; accepted 18 January 2008

Abstract

One of the key limits to miniaturizing the size of liquid fueled combustors is the atomization process applied in meso-scale systems. A single-wall fuel-film combustor was introduced recently as one of the successful liquid fuel combustors at the meso-scale. Instead of atomizing the fuel, film combustors spread out a liquid film along the wall and absorb the heat transferred from the flame for vaporization. With a single-wall film design, however, there are some unexpected and disadvantageous combustion phenomena. This paper attempts to improve the single-wall film combustor by exploring separately a double chamber concept and a central porous fuel delivery concept. These two configurations help describe the limits and the potential of liquid fuel-film miniature combustors. The double chamber design demonstrates how heat transfer issues can be overcome by injecting the fuel-film on the outside of the primary combustor wall rather than on the inside, and the second design demonstrates a flame-holding mechanism using a porous material set on the bottom of the chamber. The combustion behavior in these two configurations is compared with that in the original single-wall miniature fuel-film combustor, revealing new aspects that are relevant to portable power generation with high specific energy and power.

Keywords: Miniature combustor; Meso-scale combustor; Porous injector; Liquid film fuel

1. Introduction

The opportunities and demand for devices that enhance autonomous human mobility, surveillance, and communication are growing rapidly, and new power sources to energize these mobile systems are becoming critical. Many of these lightweight devices demand tens of watts of power for durations on the order of tens of hours, thereby driving the power-source considerations toward those with highest energy density [1]. Portable electronics and other devices that require orders of 1 W power for reliable operation can be sufficiently powered by batteries, but devices that require between 10 W and 1 kW of power for long durations need alternative energy sources. For rapid energy release, combustion is a logical approach since it employs volumetric rather than surface reactions, as are utilized by batteries and fuel cells. Since internal combustion has the potential to provide simultaneously high power density and high energy density, it is natural to explore the feasibility of internal combustion for high density power generation. The micro-gas-turbine [2] the mini- and micro-rotary engine, the micro-rocket [3], hybrid micro-combustor [4], and the micro-Swiss-roll burner [5,6] are early examples of this strategy. These devices, though not yet optimal, have demonstrated the plausibility of internal combustion as a personal power-source. In addition, substantial recent efforts in microscale combustors [7,8] have demonstrated the feasibility of combustion at sub-millimeter scales. A major challenge for all miniature combustion concepts is the increasing surface-to-volume ratio (S/V) with decreasing

* Corresponding author. Tel.: +1 949 824 8745; fax: +1 949 824 8585.
E-mail address: ddunnran@uci.edu (D. Dunn-Rankin).

0894-1777/$ - see front matter © 2008 Elsevier Inc. All rights reserved.
doi:10.1016/j.expthermflusci.2008.01.005

Nomenclature

A_{surf}	chamber's surface area (averaged on the wall thickness) (m^2)
$c_{p,fuel}$	fuel specific heat (J/kg K)
h	convection coefficient (J/s K)
k_{wall}	wall thermal conductivity (W/m K)
L	fuel latent heat of vaporization at boiling point (J/kg)
$\dot{m}_{fuel}$	fuel flow rate (kg/s)
q_{cond}	conductive heat transfer rate through the wall (J/s)
q_{conv}	convective heat transfer rate (J/s)
q_{fuel}	heat transfer rate that the fuel can absorb (J/s)
q_{rad}	part of the conductive heat transfer rate that is lost for radiation (J/s)
R_{av}	averaged distance between chamber axis and wall (m)
R_c	chamber radius (m)
t	wall thickness (m)
T_{flame}	flame temperature (K)
T_{outer}	outer wall temperature (K)
T_{W1}, T_{W2}	wall temperature on the inside and outside of the inner wall (K)
ΔT	temperature difference between fuel boiling point and fuel injection (K)
σ	Stefan–Boltzman constant = 5.67×10^{-8} W/m^2 K^4
ε	wall emissivity

size (since this ratio scales as the inverse of the combustor length scale). Because wall temperatures are generally kept fairly low due to material considerations, a high S/V usually increases the relative heat losses and increases wall effects that can quench the combustion process or reduce ignition reliability. This led researchers to concentrate on quench-resistant fuels such as hydrogen [9], liquid fuel-film combustors [10,11], high-preheat concepts (as with the Swiss-roll burner), or catalytic surfaces [12,13]. In addition, small length scales mean short residence times, often making it difficult to fully complete combustion reactions in the time available. Swirl is also utilized in the meso-scale combustor to enhance fuel–air mixing, prolong residence time and generate the recirculation zone for flame stabilization. [14]. This paper studies further options involving fuel-film combustion at the small scales.

Although liquid hydrocarbon fuels inherently contain high specific energy and power, it is difficult to directly operate on liquid fuels without any heat source for evaporating the liquid fuel in meso-scale systems. If the fuel can be injected in a manner that provides large fuel surface areas, however, then the necessary vaporization rates can be maintained. Traditionally, sprays have been employed to achieve this goal, but the new concept introduced by Sirignano and co-workers involves injecting the fuel as a thin film on the inner surface of the combustion chamber [15–18]. The fuel-film serves as an intermediate layer through which heat that would normally be conducted to the chamber walls instead vaporizes the liquid film, thus preventing quenching, while still keeping the wall temperatures below the boiling point of the fuel. Moreover, the flame structure and combustion stabilization of fuel-film combustor was further discussed [18]. Based on the limitations observed in the original concept of the liquid film combustor, two new ideas emerged. One is a liquid film combustor with a double chamber; the other is a liquid film combustor with a central porous fuel inlet. This paper details the concepts and laboratory performance of these two combustors.

2. Miniature liquid film combustors with double chamber

2.1. Challenge I: non-uniform fuel-film distribution in single-wall combustor

Previous experiments and numerical simulations [16] on a single-wall miniature fuel-film combustor have shown that more time is needed for liquid evaporation and mixing inside the chamber in order to achieve complete combustion. Another challenge of all previous single-wall devices is the non-uniform distribution of the fuel-film on the surface of the combustor for some ranges of airflow, due to the fact that spreading of fuel was accomplished only by the swirling inlet air pushing liquid along the walls. Also, incomplete cooling of the walls near the exit of the combustor was an additional problem, caused by premature complete evaporation of the liquid before it covered all the accessible surface. These problems suggested the idea of creating the double chamber combustor.

2.2. Concept

Fig. 1, shows a schematic of the double chamber combustor. It has one cylindrical wall (main chamber or combustion chamber), inside which the combustion occurs, and a second cylindrical wall around it, which creates an annulus of few millimeters between the two walls, inside which the fuel is injected. The fuel-film cools the inner chamber from the outside. The two cylindrical pieces are welded together near the exit of the chamber to create a single piece. The fuel enters from the extremity of the chamber near the flame exit, and then travels along the wall until it reaches the other extremity from which it enters the main chamber and participates in combustion. Air (which is the oxidizer) is also injected in the annulus, to premix with the fuel that is continuously vaporized by the heat coming from the flame and passing through the wall. The presumed mechanisms of evaporation and combustion are illustrated

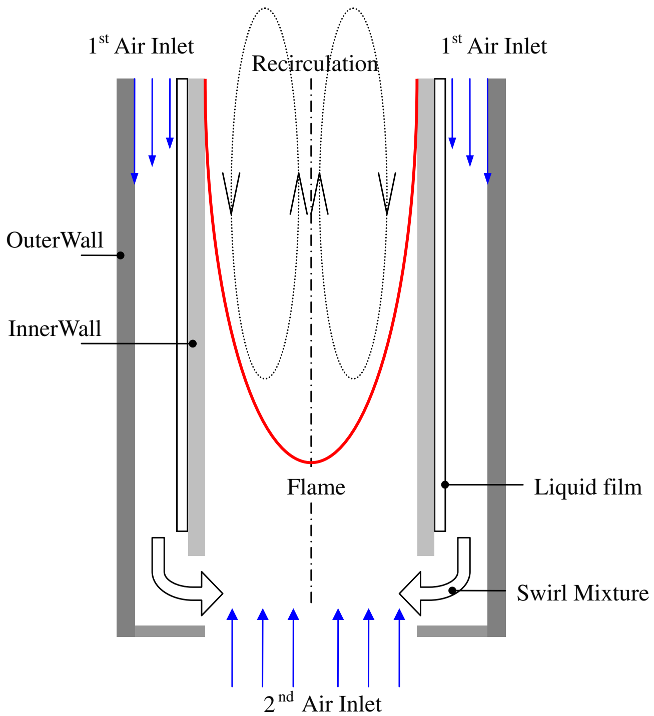

Fig. 1. Sketch of the configuration of the double chamber miniature combustor.

in Fig. 2. The swirling air has two functions: one is helping to keep the liquid fuel as a thin film on the surface of the inner wall, cooling down the outer wall at the same time (besides reducing heat loss, this internal evaporative cooling allows the combustor to be handled easily even when operating). The other is to premix the gaseous fuel and the air in an annular area and to increase the residence time available for premixed combustion in the main chamber. As for the vaporization mechanism, the heat passes through the walls from one side to the other due to conduction, until it finally reaches the fuel-film laying on the outer surface of the main chamber where it is absorbed. If the liquid flow rate is sufficiently high, the liquid can store all the heat in its liquid-to-gas phase change. New fuel is constantly injected on the wall for storing the heat coming from the flame and keeping active the film protection on the surface. When the fuel is vaporized, it mixes with the air in the annulus and enters the chamber where it burns. Residence times inside the chamber nearly double over the single-wall design because the distance covered by the fuel/air mixture doubles while velocity does not increase. In this design we expect more complete mixing and combustion inside the chamber.

2.3. Analysis

The purpose of the double-wall combustor is to improve the hot exit wall temperature of the single-wall miniature fuel-film combustor due to the non-uniformity of the liquid fuel-film. Consequently, estimating the level of fuel-film coverage on the wall is important when the double-wall combustor's dimensions, the type of fuel, the fuel flow rate, and the equivalence ratio are decided. Whenever a complete fuel-film coverage on the walls is accomplished, heat losses through the wall covered by the fuel are eliminated. In order to perform the theoretical analysis, simplified heat transfer equations for the system were used.

The simplifying hypotheses on which the heat transfer model is based are: (1) uniform flame temperature inside the chamber, (2) uniform velocity field whose magnitude represents the average axial velocity, (3) the thickness of fuel-film is fairly thin, and (4) the surface tension and viscosity of fuel-film are negligible. The fuel-film departs from the beginning of the inside wall, where the injection point is located, and then it moves towards the other extremity continuously and uniformly. The model of the vaporization

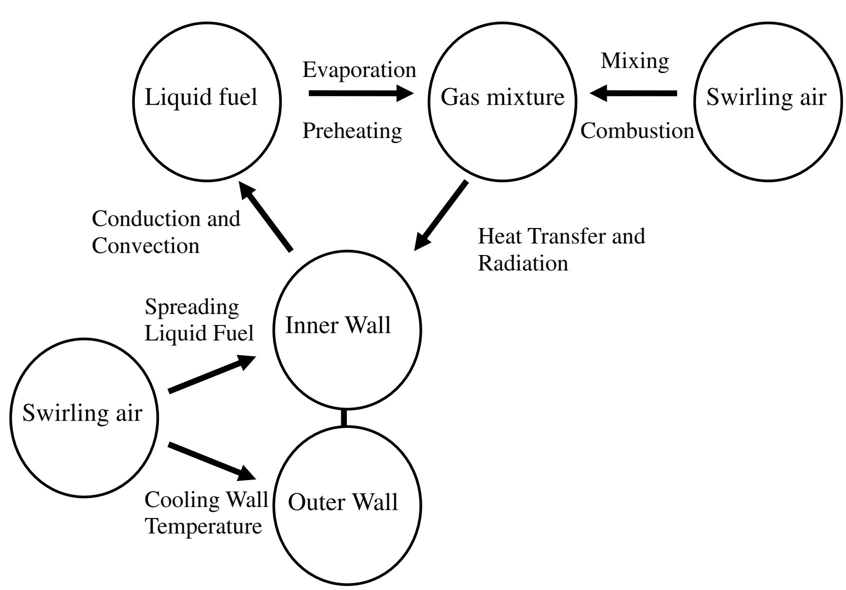

Fig. 2. Combustion and vaporization mechanism of the double chamber miniature combustor.

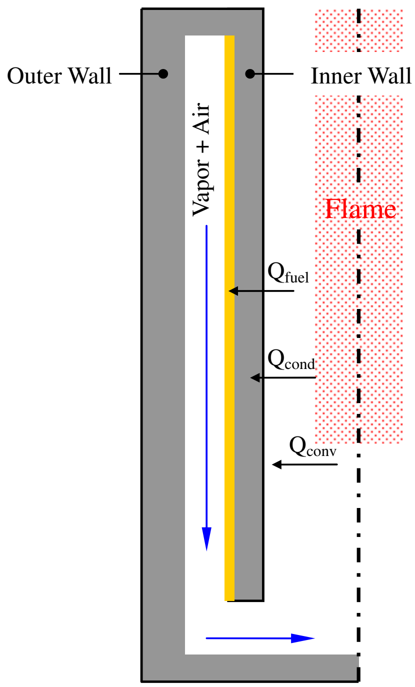

Fig. 3. Heat transfer model of the double-wall miniature combustor.

mechanism is shown in Fig 3. A flame is present inside the chamber which delivers heat to the adjacent walls. Once the heat conducts through the walls it heats the fuel-film on the surface. The liquid fuel can absorb all the heat to meet the requirements of liquid heat-up and latent heat of evaporation. Gaseous fuel then premixes with fresh swirling air in the annulus, and the mixture is delivered to the combustion chamber.

The equations for this model are:

Equation of convective heat transfer (Newton's law of cooling)

$$q_{\mathrm{conv}} = A_{\mathrm{surf}} h (T_{\mathrm{flame}} - T_{\mathrm{W1}})$$

Equation of conductive heat transfer (Fourier's law for cylindrical surfaces)

$$q_{\mathrm{cond}} = \frac{A_{\mathrm{surf}} k_{\mathrm{wall}}}{R_{\mathrm{av}} \ln\left(\frac{R_{\mathrm{c}}+t}{R_{\mathrm{c}}}\right)} (T_{\mathrm{W1}} - T_{\mathrm{W2}})$$

Equation for heat transfer in the fuel-film

$$q_{\mathrm{fuel}} = \dot{m}_{\mathrm{fuel}} (L + c_{p,\mathrm{fuel}} \Delta T)$$

The equation of radiative heat transfer from the chamber's inner wall to the outer wall was also included to provide a more realistic thermal interaction:

$$q_{\mathrm{rad}} = A_{\mathrm{surf}} \sigma \varepsilon \left(T_{\mathrm{W2}}^4 - T_{\mathrm{outer}}^4\right)$$

Radiation represents a loss of heat for the combustor, which exists also when the surface is entirely covered by the film (in this case though, the loss is very small). According to what was affirmed earlier, one more important relation to include was:

$$q_{\mathrm{cond}} = q_{\mathrm{conv}}$$

In order to calculate the percentage of surface covered by the fuel-film, the fuel flow rate $\dot{m}_{\mathrm{fuel}}$, the equivalence ratio ϕ and the length and diameter of the chamber (respectively, L_{c} and D_{c}, on which the surface area A_{surf} depends) is considered. According to this consideration, a new equation in the model is added:

$$q_{\mathrm{exceeding}} = q_{\mathrm{cond}} - q_{\mathrm{fuel}} - q_{\mathrm{rad}} \frac{q_{\mathrm{fuel}}}{q_{\mathrm{cond}} - q_{\mathrm{rad}}}$$

The quantity $q_{\mathrm{exceeding}}$ represents the amount of heat transfer rate that is in excess with respect to the amount that the fuel can store without vaporizing completely. The greater this quantity, the smaller the surface of the wall covered by the film. The final model is characterized by six equations and six unknowns, which are T_{W1}, q_{conv}, q_{rad}, q_{fuel}, q_{cond}, and $q_{\mathrm{exceeding}}$. All other quantities are either specified or calculated. From the solution of this system, $q_{\mathrm{exceeding}}$ can be found. From $q_{\mathrm{exceeding}}$, it is possible to calculate the percentage of wall's surface covered by the fuel. The percentage A_{cov} could be calculated by substituting q_{cond} with $q_{\mathrm{cond}} - q_{\mathrm{exceeding}}$ and by finding the new value of A_{surf}, which represents the surface that, at the same diameter, the chamber should have for complete coverage on the wall. A_{cov} is the ratio between the new A_{surf} and the old one.

Numerical simulations, using Matlab©, solved the system of equations and provided the response when the input parameters varied. The simulations have been run both for heptane and methanol fuel:

Heptane

$$L_{\mathrm{BP}} = 317,500\ \mathrm{J/kg}$$
$$c_{p,\mathrm{liquid,BP}} = 2560\ \mathrm{J/kg\ K}$$
$$\rho_{\mathrm{liquid,BP}} = 615\ \mathrm{kg/m^3}$$
$$T_{\mathrm{boil}} = 371.59\ \mathrm{K}$$

Methanol

$$L_{\mathrm{BP}} = 1,100,000\ \mathrm{J/kg}$$
$$c_{p,\mathrm{liquid,BP}} = 2813\ \mathrm{J/kg\ K}$$
$$\rho_{\mathrm{liquid,BP}} = 780\ \mathrm{kg/m^3}$$
$$T_{\mathrm{boil}} = 337.8\ \mathrm{K}$$

In these expressions, L is the latent heat of vaporization and BP indicates the boiling point of the fuel.

The input parameters of the simulation were the diameter of the chamber (D_{c}), the length of the chamber (L_{c})), the fuel flow rate ($\dot{m}_{\mathrm{fuel}}$) and the equivalence ratio (ER). The output included the percentage of chamber's surface covered by the fuel (A_{cov}), the flame's average temperature, the Reynolds number, the Nusselt number, the Prandtl number, the convective coefficient (h), and the flame's axial velocity.

According to the heat transfer model, Fig. 4 shows different diameters of the combustor corresponding to its film coverage at the same equivalence ratio (ER = 1). It appears that the film coverage of methanol approaches 100% when

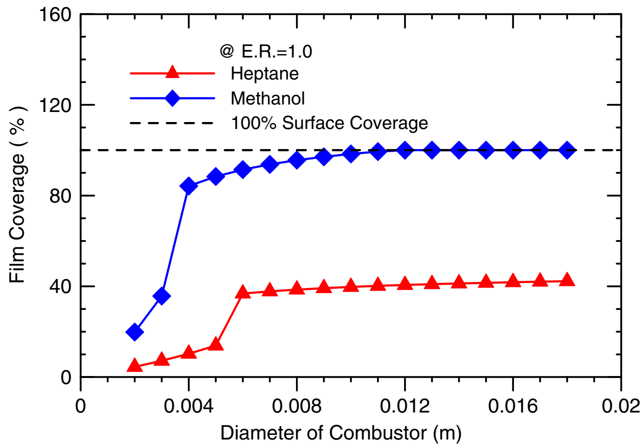

Fig. 4. Film coverage at different combustor's diameter for *n*-heptane and methanol.

the combustor's diameter is over 8 mm. But the film coverage of *n*-heptane does not exceed 40% even when the combustor's diameter grows larger. Fig. 5 shows the relationship between film coverage and equivalence ratio (ER) for a fixed inner diameter of the chamber (0.8 cm). While the equivalence ratio increases, both liquid fuels have higher film coverage, especially methanol. Methanol fuel can cover 100% of the surface for an equivalence ratio close to 2, while *n*-heptane does not go beyond 40% at equivalence ratio 2. As shown in Fig. 4, both methanol and *n*-heptane show an increasing trend of coverage with increasing combustor diameter and equivalence ratio. These results indicate that the practical combustor must run rich to allow proper cooling fuel coverage of the surface. The most important aspects of the simplified thermal model are presented above but complete documentation is also available to those interested in further details [19].

2.4. Experimental performance

Although methanol fuel coincidentally provides complete coverage of the surface at 8 mm diameter (or higher), the actual reason for setting the diameter of the practical combustor to 8 mm is to retain similarity to the previous single-walled version of the combustor. As noted earlier, with this diameter, the predicted film coverage of *n*-heptane is much lower than that of methanol.

In the experimental combustor, the surface of the outside of the inner wall is knurled to increase the capillarity effect and keep the fuel-film in contact with the surface. The overall length of the double chamber combustor is 7.5 cm, and the overall diameter is 2 cm, while the distance between the two combustor's walls is 3 mm. Two air inlets enter the chamber right near the exit; the fuel is injected through two inlets placed at 0.5 cm distance from the air inlet, towards the bottom. Two other air inlets are placed at the bottom of the chamber. All the inlet tubes are arranged to create swirling air and to spread fuel on the wall, as shown in Fig. 6. In particular, the fuel inlets are inclined 15° to inject more directly the fuel on the knurled surface. The thickness of the walls, both the inner and the outer, is 1.65 mm.

Air, metered by a flowmeter, is injected tangentially below the fuel ports at flow rates between 3.62 and 30 l/min, resulting in mean inlet flow velocities ranging from about 1.2 to 10 m/s with corresponding non-reacting Reynolds numbers between 750 and 7000, respectively. The overall equivalence ratio varies between 1 and 2. Fuels are *n*-heptane and methanol.

2.4.1. Methanol liquid fuel

The theoretical analysis predicts that the methanol can have a good film coverage compared to *n*-heptane. However, experiments on methanol/air combustion in the double-wall device reveal ignition problems due to methanol's high latent heat of vaporization. Similar problems were encountered with the single-wall combustor. For successful ignition, preheating the double-wall combustor with a torch is necessary in some conditions. Besides the ignition problem, the problem of quenching was predominant in

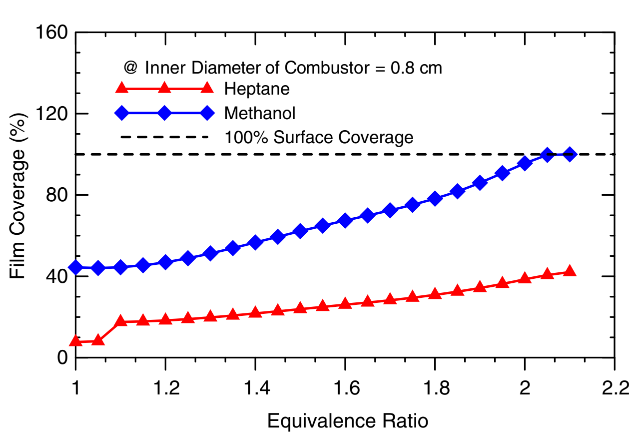

Fig. 5. Film coverage at different equivalence ratios for *n*-heptane and methanol.

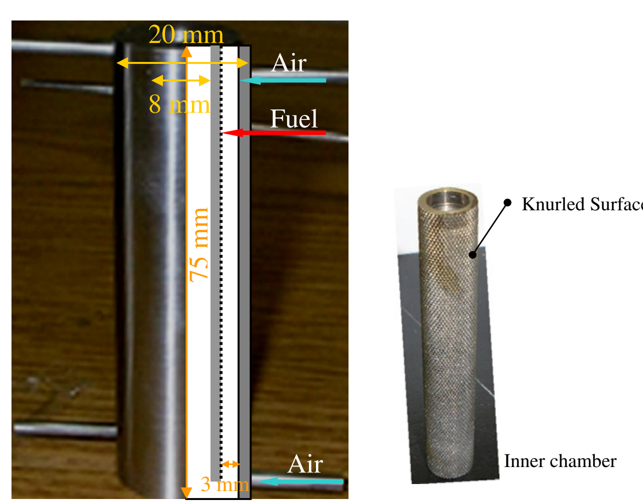

Fig. 6. Configuration of the double-wall miniature combustor.

the tests. The tests were successful using mixtures of equivalence ratios 1, 1.5 and 2, respectively, and thermocouples were used to measure the temperature outside the inner wall in different locations of the combustor, as shown in Fig. 7A. When the equivalence ratio is 1.0, the flame is both outside and inside the chamber, but it is unstable and it extinguishes within 30 s. When the equivalence ratio increases to 2, combustion occurs outside only (Fig. 7B). The flame is diffusive and is anchored to the exit. There is never extinction in this case regardless of the air flow rate. Temperatures were taken on the outside of the internal wall, as shown in Fig 7C. The results show that the film is present. Temperatures in measurement location #1 (close to the exit) are higher than in the other locations because some heat is transferred by conduction from the flame to the walls near the exit.

The methanol experiments showed that burning this fuel is very difficult. Although methanol was better at creating the fuel-film on the wall because it has a higher heat of vaporization, it had a worse behavior during combustion because it burned in a narrower range of conditions, and it was difficult to ignite. Furthermore, whenever the flame did not extinguish, it was usually burned outside the chamber. When using this fuel, a flame was generated at equivalence ratios greater than 1, but stability was not accomplished at any flow rate.

2.4.2. *n*-Heptane liquid fuel

n-Heptane liquid has a much lower heat of vaporization compared to methanol (*n*-heptane = 318 kJ/kg, methanol = 1100 kJ/kg), making it a more attractive fuel for increasing the mixing rate of gaseous fuel/air in the double-wall combustor. A stable, self-sustained and confined combustion is possible within the double-wall device, but only for quite rich mixtures. When the equivalence ratio is equal to 1, the flame is mostly outside the chamber and is a little unstable. The flame plume's length varies randomly in time. The average length of the flame plume is about 4 cm. Usually, the flame rushes outside the chamber and extinguishes. With increasing equivalence ratio, the flame reaches stable conditions with occasional perturbations. Additionally, the flame plume length increases, but the outer wall temperature remains low, close to room temperature. Fig. 8 shows a *n*-heptane/air flame burning in the double-wall combustor at an air flow rate equal to 17.8 l/min, and a fuel flow rate equal to 281 cc/h, corresponding to an equivalence ratio equal to 2. The flame appears stable. The images show both swirling properties and plume characteristics. The swirling seems very strong and the flame is very thin. The plume is about 12 cm long and up to 2 cm in diameter near the exit of the chamber. This is evidence that substantial fuel combustion is occurring outside the combustor. The external wall, however, can be maintained at low temperature.

Unfortunately, as with the original single-wall design, the double-wall combustor does not allow good combustion when the mixture is lean (i.e. $\Phi < 1$). Attempts in burning *n*-heptane at equivalence ratios between 0.8 and 0.95 showed ignition difficulties. Even when ignition was accomplished, the flame would quench almost immediately. The

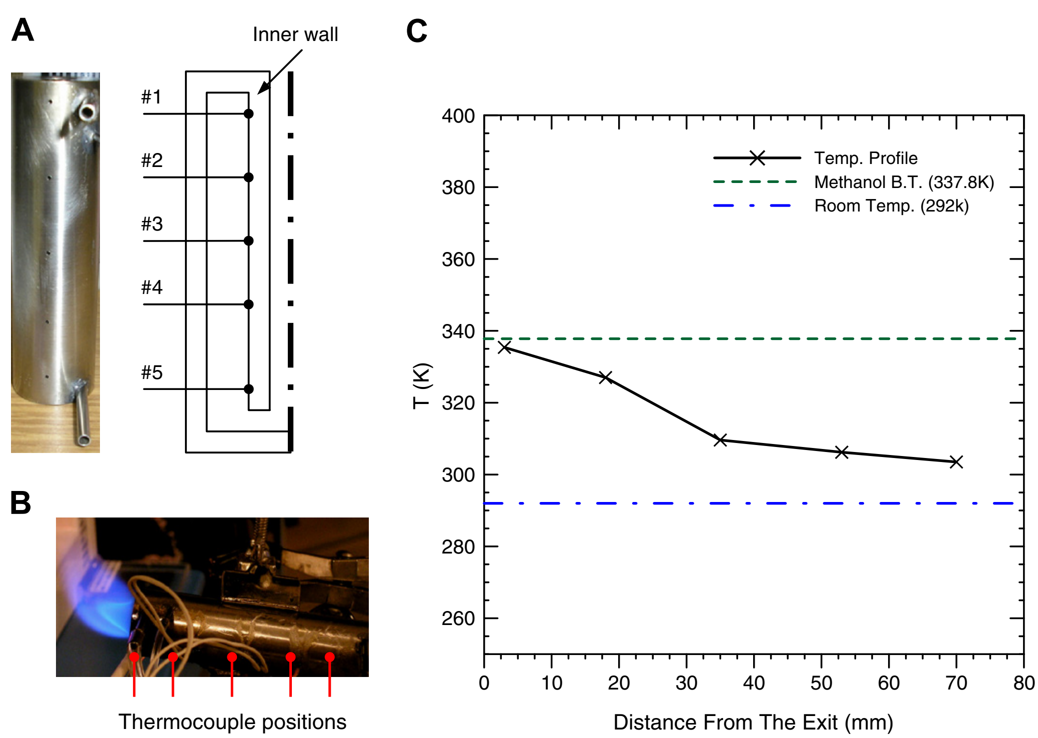

Fig. 7. (A) Position of the thermocouples along the combustor; (B) methanol flame structure; (C) temperature measurement along the combustor at air flow rate = 1.45 l/min and methanol flow rate = 71.5 cc/h.

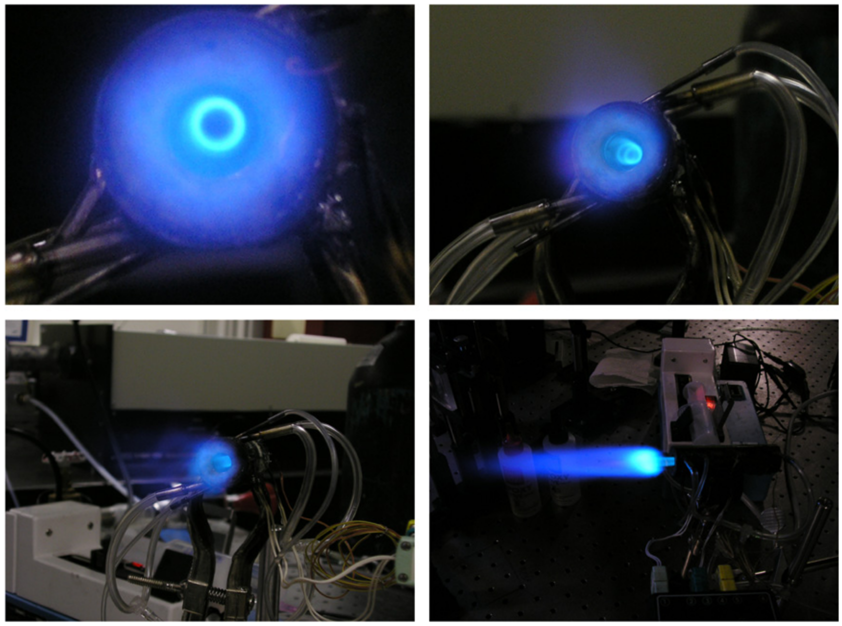

Fig. 8. (A) Thickness of the flame and swirl properties; (B) and (C) images of the swirling flame; (D) Plume's characteristics.

reason for this behavior is that the chamber is too long and narrow to allow the ignition flame to reach the bottom. For mixtures very fuel rich (i.e. $\Phi > 2$) a flame is easily created and maintained. For equivalence ratios close to $\Phi = 3$, however, liquid fuel escapes from the exit. In this condition the plume is very large and sooty.

In order to survey combustion pulsation in the double-wall combustor, the temperature outside the inner wall was monitored. Thermocouples were used to measure the temperature outside the inner wall in five different locations of the combustor, as shown in Fig. 7A. Location #1 is the one closest to the chamber exit and to the first swirling air extremity, and location #5 is the one down at the bottom of the combustor and close to the second swirling air extremity. Tests were run for different equivalence ratios and different flow rates. Fig. 9A and B indicate, for example, the temperature outside the inner wall at different air flow rates and different equivalence ratios at locations #3 and #5. When the equivalence ratio is one, the temperature outside the inner wall is mostly above the *n*-heptane's boiling temperature in both locations, except at high flow rate. Hence, the flame is confined inside the chamber and heat is absorbed by the adjacent wall to dry out the fuel-film on the surface. Therefore, the liquid film can vaporize before it enters the main chamber, and then premixed gases burn inside the main chamber. Although the double chamber design can efficiently increase the residence time, the flame inside the chamber offers more heat than the liquid fuel evaporation requires, triggering combustion pulsation. Moreover, without a geometric auxiliary (i.e., a flame holder) for stabilizing the flame inside the chamber it is sensitive to the environment and to the experimental apparatus.

Once the equivalence ratio is up to two, the fuel-film can successfully cover the entire surface of the inner tube at some of the tested air flow rates. Fig. 10 shows the temperature in five positions along the chamber. Outside the inner wall, the temperature is below *n*-heptane's boiling temperature as the air flow rate is equal to 8.89 l/min. Combustion is stable with occasional combustion pulsations. The flame exists both inside and outside the chamber, and its length is much longer than that in stoichiometric conditions. Moreover, the temperature outside the outer wall approaches room temperature. It appears, therefore, that at rich equivalence ratios the pulsations tend to be less frequent than at stoichiometric conditions. However, at rich conditions, the portion of the flame that is outside the chamber is large, and so combustion is more incomplete which reduces internal combustion efficiency. To reduce the pulsation effects at stoichiometric conditions, increasing the fuel flow rate is a possible solution, since more fuel can be available to absorb the heat, and the film protection can last longer, preventing the wall from becoming uncoated.

2.5. Discussion

All results, obtained at different flow rates and equivalence ratios, suggest that the combustor should be designed to operate near stoichiometric conditions to have a short plume outside the chamber, which would mean fewer losses from the exit of the combustor, higher efficiency, and more power generation. Unfortunately, both theoretical analysis and experimental measurements on simple steel cylinders suggest that the combustor must run rich to have fuel-film protection on the walls and lower heat losses. One design solution to this apparent conflict could be to run the combustor at both low equivalence ratio and high fuel flow rate, to maintain both fuel-film coverage on the wall and a short plume outside. This is quite promising because

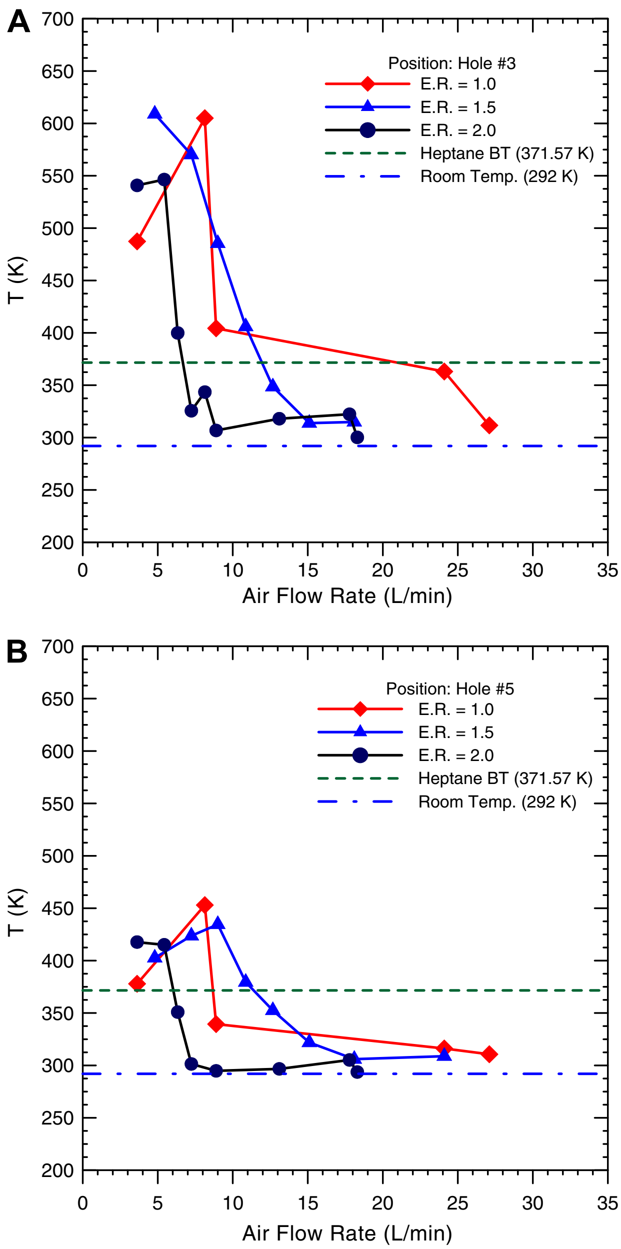

Fig. 9. (A) Temperature outside the inner wall at different air flow rates in hole #3; (B) temperature outside the inner wall at different air flow rates in hole #5.

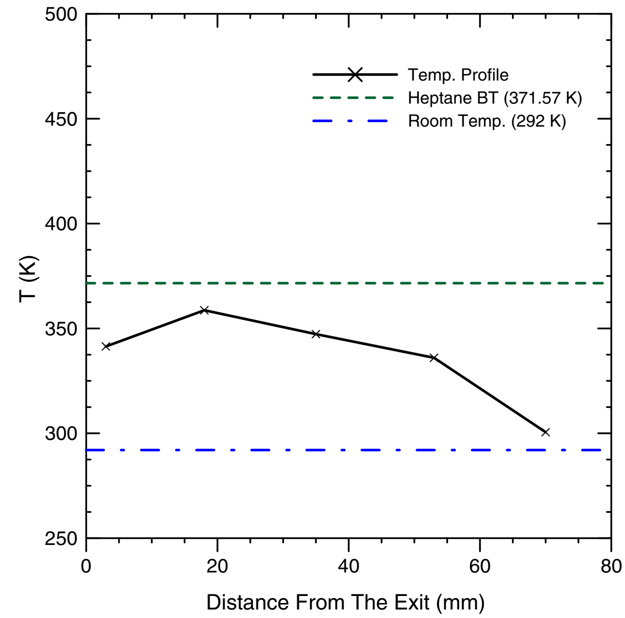

Fig. 10. Temperature outside the inner wall along the wall surface at air flow rate = 8.89 l/min and heptane fuel flow rate = 140.5 cc/h.

the plume's length increases with flow rate much less than it decreases when decreasing the equivalence ratio. Even though low equivalence ratios tend to create instabilities, high flow rates tend to re-stabilize the combustion. Finally, just as with the single-wall design, there is a large difference in behavior between *n*-heptane and methanol. Methanol is better for creating the fuel-film on the wall but worse for combustion performance. The double chamber combustor remains a promising design concept but its geometry must be adjusted to accomplish the above requirements.

An alternative approach to solving the problems identified with fuel-film coverage and external combustion in the single-wall and double chamber film designs is to allow a more conventional hot wall boundary and to instead flow the fuel through a central inlet that might also act as a bluff body flame stabilizer. This view led to the central porous fuel inlet design discussed next.

3. Miniature liquid film combustors with central porous fuel inlets

3.1. Challenge II: long plume flame outside the chamber

As described above, the double chamber combustor reduced the wall temperature successfully, but the flame burned mostly outside the chamber. It is similar to the single-wall combustor in that a long plume flame exists outside the chamber due to unburned fuel and the lack of an internal flame stabilization mechanism. Tangential air injection can offer a recirculation zone for flame stabilization, but at these small scales the anchor of the recirculation zone is sensitive to the chamber's configuration and fluid flow conditions. Hence, in order to further confine the flame inside the chamber and also to preheat the liquid fuel, a central porous fuel inlet combustor was designed.

3.2. Concept

As described earlier, when burning liquid fuels in miniature combustors, the optimum choice is to inject all or a portion of the fuel directly as a film on the solid surfaces where high heat transfer from the combustion products occurs. One major purpose of designing a combustor is so that liquid can be sprayed onto a chosen surface or injected through an orifice or porous material. Then, the liquid can spread over the surface by its own momentum and surface tension, by the friction forces from the neighboring flowing gases, and by the gravitational field. On the other hand, the interface between the liquid fuel and chamber wall is supposed to offer the liquid fuel sufficient energy for latent and sensible heat immediately and efficiently by any form of heat transfer. In general, porous media have a surface structure that spreads fuel over the wall uniformly and completely. In addition, porous media can provide excellent heat recuperation by radiation and conduction. Applying a porous cap in the core of a miniature combustor meets the requirement of forming a fuel-film and aids in directly vaporizing the liquid film from the metal-porous surface. In order to enhance the likelihood of combustion in a confined chamber at high flow rate, the central porous miniature combustor design can be further combined with other well-known strategies, such as swirl generators and vortex generators, to enhance vaporization and mixing rates.

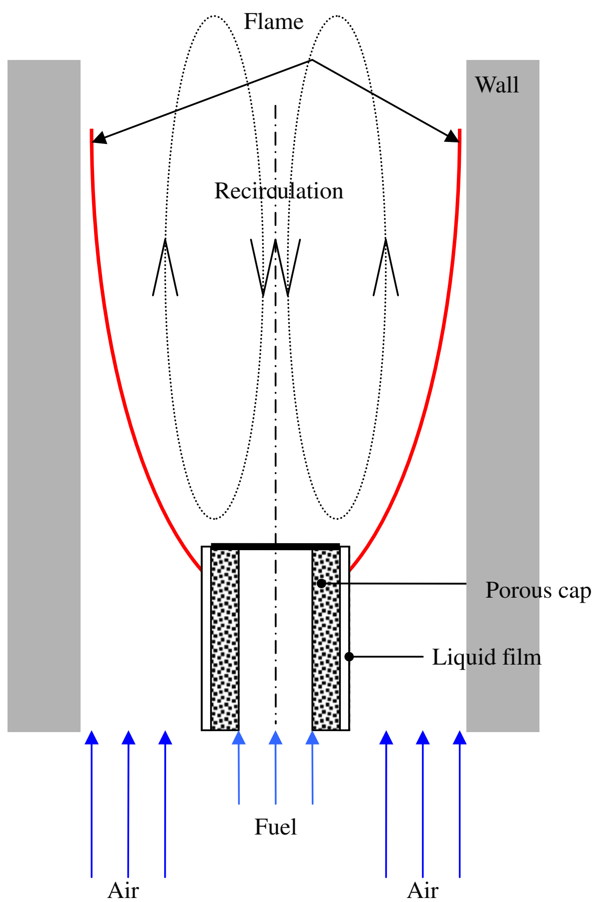

Fig. 11. Sketch of the configuration of the central porous fuel inlet miniature combustor.

Fig. 11 illustrates the basic concept of a central porous miniature liquid film combustion chamber. Swirling air entering the cylindrical chamber draws in liquid fuel and generates a liquid film on the porous surface. The swirl also provides a recirculation mechanism whereby the flame can be stabilized. The porous cap is presumed to have two functions: to supply a liquid film-surface area large enough to produce the necessary fuel vaporization rates and to hold the flame in the chamber. To demonstrate the phenomena involved within the miniature combustor, the primary evaporation and combustion mechanism is shown in Fig. 12. Liquid fuel vaporizes from the surface of the porous cap, wherein the liquid fuel is supplied. Evaporation of the liquid fuel and preheating of the fuel vapor on or inside the porous cap require latent heat and sensible heat, which are supported by convective heat transfer from solid to liquid/vapor via solid–liquid/vapor interactions and by thermal radiation absorption by the liquid fuel and fuel vapor. Chemical reaction takes place outside the porous cap, where the preheated liquid fuel vapor meets and mixes with the swirling air to form a homogeneous gas mixture followed by combustion. Thermal radiation and convective heat transfer from the flame to the porous cap drives evaporation.

3.3. Analysis

The purpose of the central porous fuel inlet combustor is to avoid the surface area limitations identified in the double-wall film burner and to confine the flame inside the chamber. The liquid fuel is kept on the porous medium. The heat provided to the fuel-film for vaporization is

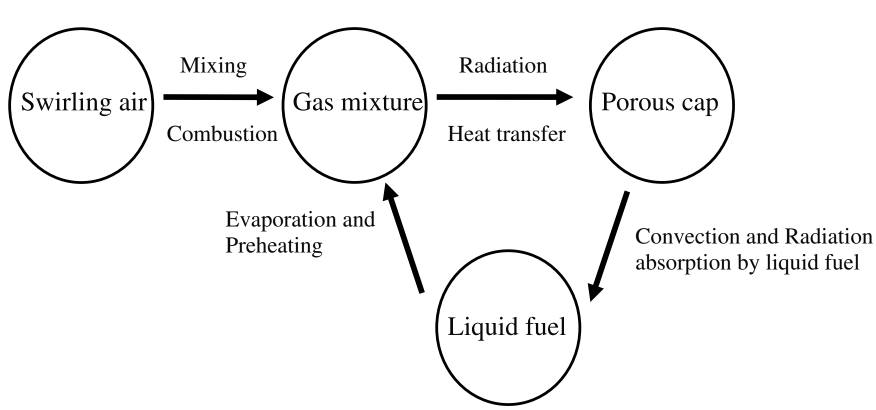

Fig. 12. Combustion and vaporization mechanism of the central porous fuel inlet miniature combustor.

extracted from the flame and absorbed by the fuel and the porous medium. Therefore, the temperature of the porous medium is a significant indicator of efficient combustion in this design. The heat transfer in the central porous fuel inlet combustor was studied using the simplified model shown in Fig. 13. Firstly, the model assumes that the flame exists inside the chamber and provides the porous media with heat due to convection. The porous medium absorbs the heat and preheats the fuel-film by heat conduction. Once the fuel-film overcomes the requirement of latent heat, gaseous fuel premixes with the air and combusts inside the chamber. Under the assumptions of the previously described double chamber combustor, the same heat transfer equations and numerical processes can be utilized for the porous cap design except that the effective thermal conductivity of the porous medium filled with fluid is different from that of the solid medium [20]:

$$k = k_f^{\phi} k_s^{1-\phi}$$

The effective thermal conductivity k was used in the heat transfer equations. After iterative calculation, the temperature of the porous medium as a function of different combustor diameter and porous medium's porosity in stoichiometric conditions was obtained. Fig. 14 displays the temperature of porous medium with different combustor diameters and different porous medium materials (bronze and stainless steel), having the same porosity (35.6%). When the combustor's diameter increases, the temperature of the bronze porous medium approaches heptane's boiling temperature and reaches it when the combustor's diameter reaches 12 mm. The temperature variation with combustor diameter of the stainless steel porous medium follows similar trend with that of the bronze but with a higher temperature profile. In general, stainless steel has higher volume specific heat capacity than does bronze porous media. When the combustor's diameter increases (to larger than 12 mm), with a fixed overall flow rate, the velocity decreases so that the heat can be stored inside the porous medium by reducing thermal convection. The temperature of the porous medium increases slightly and then remains flat by balancing with available liquid fuel. If the porous medium's temperature can be maintained near heptane's boiling temperature, the heat from the porous medium can continuously preheat the fuel-film regardless of flame fluctuations. Fig. 15 shows the temperature of the porous medium with different porosities in different media. The

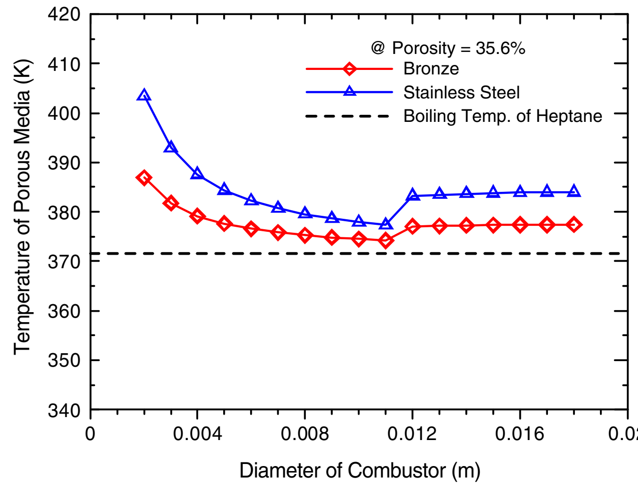

Fig. 14. Temperature of different material porous media at different combustor diameter.

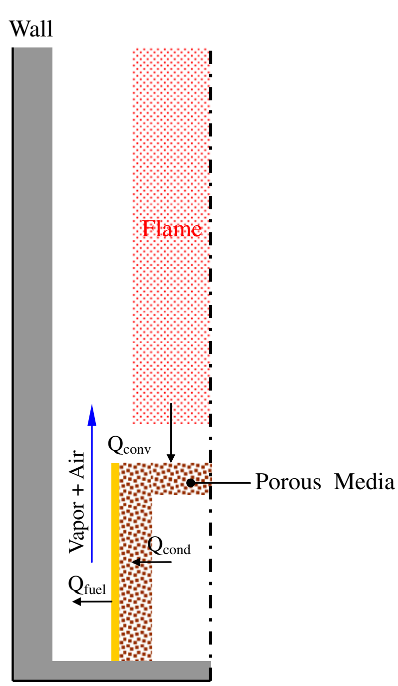

Fig. 13. Heat transfer model of the central porous fuel inlet miniature combustor.

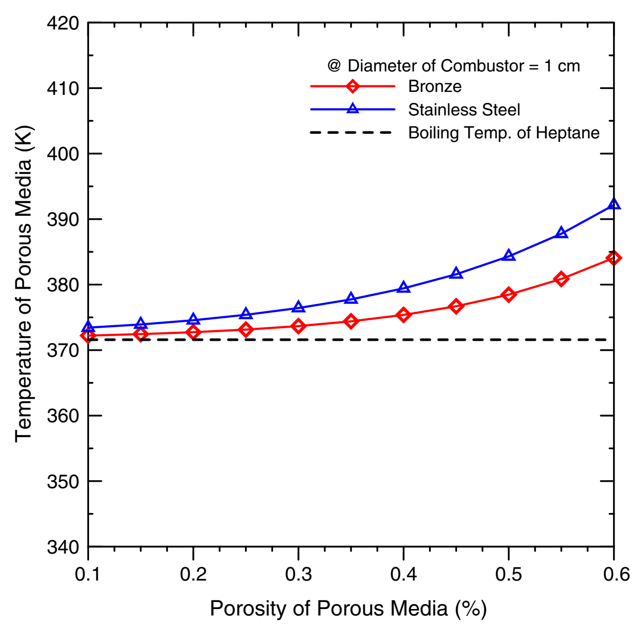

Fig. 15. Temperature of different material porous media at different porosity.

diameter of the chamber is fixed at 1 cm. However, changes do not appear significant and the temperature of both porous media is above the boiling temperature. This dries out the fuel-film on the porous medium and contributes to combustion pulsation. The challenge, therefore, is to have good preheating without over-heating.

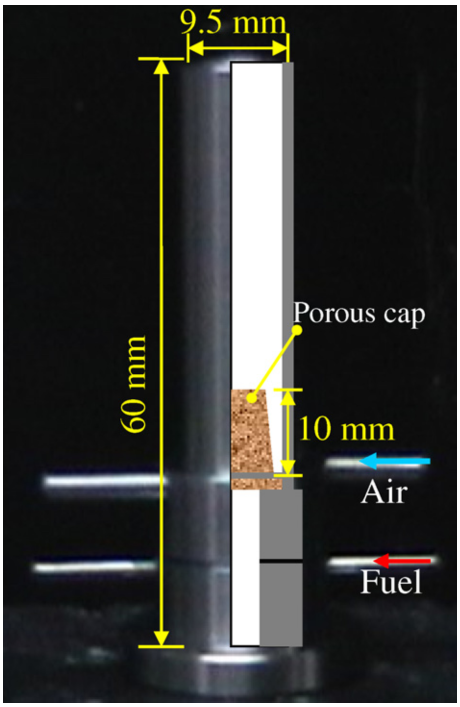

Fig. 16. Configuration of the central porous fuel inlet miniature combustor.

3.4. Experimental performance

According to previous theoretical analyses, the combustor's diameter was chosen to be about 10 mm and the porosity for the porous medium was chosen near 30%. The combustion chamber is made of a stainless steel tube (or a quartz tube for optical access) of 60 mm in length and 9.5 mm in diameter (see Fig. 16). *n*-Heptane fuel is injected from two inlet ports into the bottom of the chamber using a syringe pump. Air, metered by an electronic flowmeter, is injected tangentially below the fuel ports at flow rates between 4.5 and 9.5 l/min, resulting in mean inlet flow velocities ranging from 1.04 to 2.4 m/s with corresponding non-reacting Reynolds numbers of 580–1360. Overall equivalence ratios vary between 1.9 and 0.9.

Fig. 17 shows the flame and CH^* chemiluminescence images of heptane/air flames of different equivalence ratios from 1.9 to 0.9 in the miniature combustor. The porous-core metal combustor exhibits a wide range of flame characteristics depending on the airflow rate through the system. Fig. 17 shows this range as the airflow increases from 4.5 to 9.5 l/min while holding the liquid fuel flow rate constant at 57.9 cc/h. In low swirling flow (e.g., the first two cases in Fig. 17), the flame may be located in two different positions: on the top of porous cap and on the rim of the chamber exit, respectively. The first flame (inner flame) anchoring on the porous cap was stabilized by the

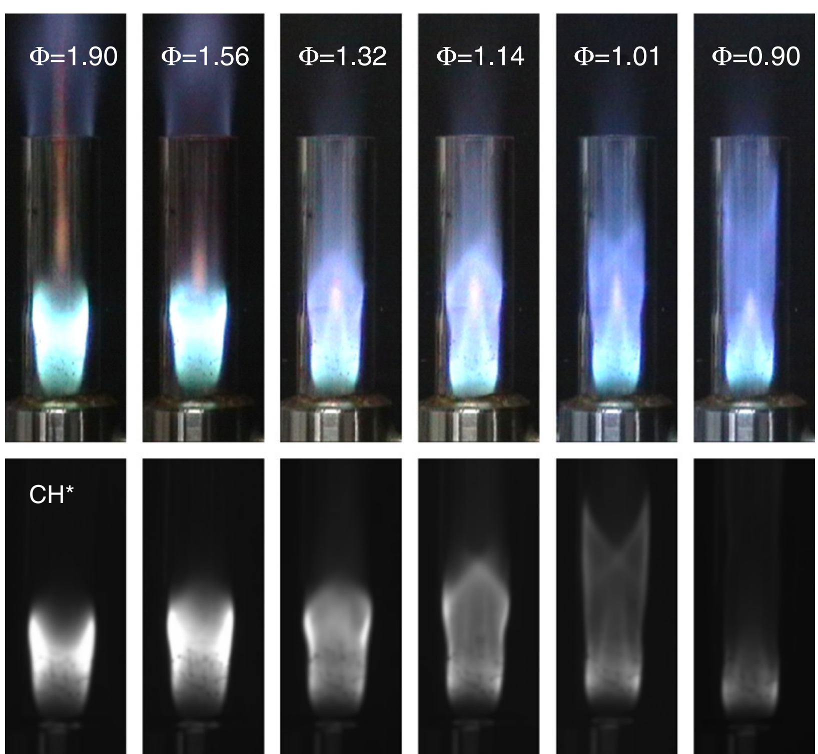

Fig. 17. Flame structure and CH^* images.

recirculation zone. The second flame (outer flame) anchored on the chamber rim because the residence time was too short for complete reaction. As the swirling air increases, the flame length shortens and its anchoring position moves upstream. Strong swirl (e.g., for the last four cases in Fig. 17) can force the flame into the chamber where it reacts more completely. In order to observe the main reacting zone of the flame, spontaneous CH^* chemiluminescence can be used to observe the flame location and flame shape. The main combustion zone in the low swirling flow cases was compressed in the bottom of the chamber and became elongated as the swirling air went up. Strong swirling air can enhance fuel–air mixing by increasing the swirl entrainment in the chamber, making the mixture more uniform so that it can react completely in the chamber. In the meantime, the wall temperature also rises with increasing swirling air. Therefore, the porous cap also heats up via heat conduction and thermal radiation from the wall. Overheated porous media can cause instability and oscillation in the combustion process. First, the heated metal-porous medium enhances vaporization of the liquid fuel until dry-out of the fuel-film on the porous medium. However, the excessive gaseous heptane which rapidly vaporizes within the hot metal pores will decrease the effective permeability of the porous medium and impede the liquid fuel supply to the porous injector. Following the vapor lock, cold liquid fuel rushes in and flushes the hot porous medium, reducing dramatically the temperature of the porous cap. In this process, the flame becomes unstable and pulsates back and forth in the chamber. The cycle of dry-out and replenishment of the liquid fuel is the main cause of flame oscillation and instability, and it depends on the characteristic time required for energy storage and fuel evaporation in the porous medium. Fig. 18 shows images taken from the top exit demonstrating the flame oscillating cycle. The images show the process of combustion oscillation from first image to fourth image. It is evident that the inner flame elongates first and then blows off; the outer swirl flame accelerates and strengthens while the combustion oscillation occurs. After flame pulsation, the inner flame re-ignites and holds on the porous medium and the outer swirl flame draws back to its original position.

3.4.1. The effect of the porous medium

The porous medium plays a critical role in the central plug combustor design. It spreads the liquid fuel as a thin film and vaporizes the liquid fuel by conducted and convected heat from the flame. The porosity and the conductivity of the porous cap are therefore essential parameters in the efficiency of combustion. In the theoretical analysis, bronze and stainless steel porous media were considered. The analysis predicted that because of its low conductivity the temperature of the stainless steel is above the liquid fuel's boiling temperature, while the expected behavior is opposite for bronze. However, the cycle of dry-out and replenishment of the liquid fuel is also related to the porous medium's temperature. For verifying if the medium's material would influence the combustion behavior, two kinds of porous media with the same porosity were tested. Fig 19 shows the limit of the operating range for the central porous fuel inlet combustor. Above the limit line the combustor can run stably. Below the limit line the flame extinguishes. When the air flow rate goes up, the flame is compressed toward the chamber wall by swirl flow. Most combustion occurs inside the chamber, and most of the heat transfer occurs towards the porous medium. The frequency of the dry-out-and-replenishment cycle also increases. For stable and continuous combustion, excess liquid fuel is required for cooling down the porous medium. Fig 19A shows that a combustor with stainless steel porous media only runs stably in the fuel rich regime. The stainless steel porous cap, having high specific heat stores thermal energy from the flame, and produces combustion instability. The instability expands in the low

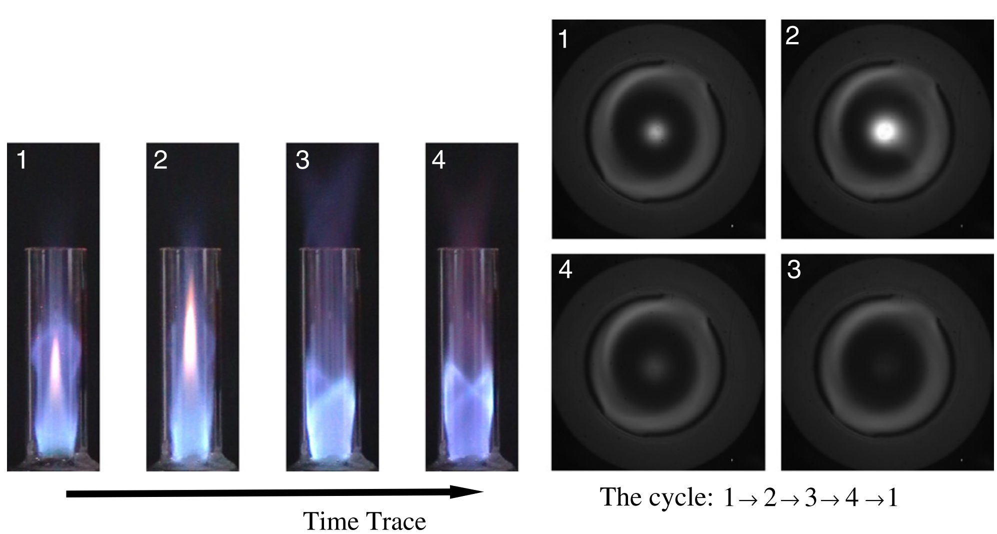

Fig. 18. Combustion oscillation and instability (cycle: 1 → 2 → 3 → 4 → 1).

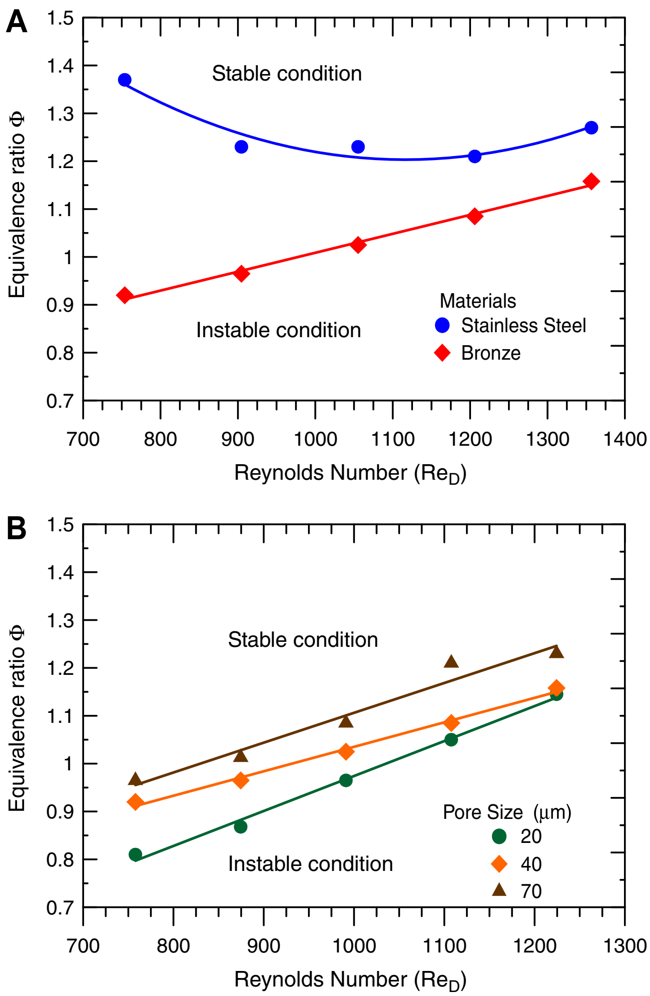

Fig. 19. Stability limits (the regime is stable above and unstable below the line) of heptane/air combustion in different (A) cap materials and (B) pore-sizes of the porous cap combustor.

Reynolds number regime due to superheating the porous cap as there is little ambient air cooling. The flame is easy to blow out during combustion oscillation, and more liquid fuel is needed to cool down the over-heated porous medium. Moreover, the porous medium temperature is affected by heat transfer from the walls and the flame. To avoid over-heating, an excessive amount of liquid fuel is necessary for the combustor with stainless steel in all conditions; hence, similarly to the double chamber combustor, it only runs stably in the fuel rich regime. The bronze porous medium, however, has relatively high thermal conductivity and low specific heat. This implies that it should be easy to conduct the heat throughout the porous medium quickly, preventing heat accumulation. The opportunity for combustion oscillation in the bronze porous medium is then reduced as compared to stainless steel. In low Reynolds number flow conditions, it is even possible to combust in the fuel-lean regime with a bronze porous cap. The stability limit of a combustor with bronze porous media tends toward the fuel-rich regime as the air flow rate increases. Fig. 19B shows the stability limits of heptane/air combustion in a different bead size of porous cap combustor. The pore-sizes are 20, 40 and 70 μm. The results show that the large pore-size porous medium operates only in fuel-rich regimes. The large pore-size porous medium has relatively low thermal conductivity according to the Katz and Thompson's formula [21]: $\phi = A_c(r_u/r_s)^{2-d_f}$, where $A_c \sim 1$, and r_u and r_s are, respectively, the pore-sizes characteristic of the upper and lower limits of the self-similar regime and $2 < d_f < 3$ is the fractal dimension. In addition, it is relatively sensitive to the environment due to its large pore-size. Surrounding perturbations easily influence the fuel-film. Small pore-size porous media have relative high thermal conductivity, and they sustain a more uniform temperature distribution. At low air flow rates or low Reynolds numbers, the flame combusts in two locations: the rim of the combustor's exit and the side of the porous medium. The porous medium accumulates heat from the flame attached to it, and combustion instability occurs. The small pore-size porous medium has high thermal conductivity and it can carry the flame heat away, but the large pore-size porous medium requires an excess of liquid fuel to cool it down. Once the air flow rate goes up, this situation becomes more serious. At high Reynolds numbers, the central porous fuel inlet combustor only runs stably in the fuel-rich regime no matter what the pore-size.

3.5. Discussion

The metal-porous medium can increase the contact surface and conduction heat transfer for liquid fuel evaporation as well as inhibit flame quench. Although high airflow rates are necessary to generate swirl for film and flame stabilization, this leads to natural oscillation of the flame. Within the range of stable airflow rates, steady injection of the fuel is necessary to prevent oscillations in the flame. As with the other film combustors, operation of the combustor's fuel supply depends on the challenge that sufficient fuel flow is required to keep the porous medium wet and cool to prevent start of flame oscillation but excessive fuel flow is to be prevented to maintain the liquid fuel-film in the operation range. In the experiments, these requirements mean that the combustor must primarily be operated in the overall fuel rich range. However, under the appropriate conditions, the implementation of the metal-porous injector and the swirling air flow produces complete combustion within the combustor and relatively low wall temperature. The role of the swirler and the porous cap as flame holders, and whether they can provide higher swirl and better mixing for improved combustor performance, requires further verification.

4. Conclusions

Two new liquid fuel-film combustors were designed to overcome limitations of earlier designs. The feasibility of the two combustors was confirmed. The double chamber combustor can reduce the outer wall temperature and combust stably, but the flame burns mostly outside the chamber and the combustor runs in the fuel-rich regime. On the other hand, the central porous fuel inlet combustor can run in the fuel-lean range and can confine most of the flame inside the chamber, but the overheating porous medium induces combustion instability. To further understand the combustion and heat transfer mechanism of liquid fueled miniature combustors it is helpful to understand more fully the flame stabilization mechanism. In the previous single-wall film combustor studies [17,18], a triple flame structure was identified as an important stabilization mechanism. The fact that the current flames also require fuel rich conditions suggests that a similar mechanism may be operating. Using chemiluminescence measurements and image observations to confirm a triple flame structure in these two alternative film combustion systems will be one important goal of future work.

References

[1] D. Dunn-Rankin, E.M. Leal, D.C. Walther, Personal power system, Prog. Energy Combust. Sci. 31 (2005) 422–465.

[2] I.A. Waitz, G. Gauba, Y.-S. Tzeng, Combustors for micro-gas turbine engines, J. Fluid Eng.: Trans. ASME 120 (1) (1998) 109–117.

[3] W. Lindsay, D. Teasdale, V. Milanovic, K. Pister, A.C. Fernandez-Pello, Thrust and electrical power from solid propellant microrockets, in: Proceedings of the 14th IEEE International Conference on Micro Electro Mechanical System, IEEE, Piscataway, NJ, 2001, pp. 606–610.

[4] C.M. Spadaccini, J. Peck, I.A. Waitz, Catalytic combustion systems for micro-scale gas turbine engines, ASME GT2005-68382.

[5] L. Sitzki, K. Borer, E. Schuster, P.D. Ronney, S. Wussow, Combustion in microscale heat-recirculating burners, in: Proceedings of the Third Asia-Pacific Conference on Combustion, Seoul, Korea, June 24–27, 2001.

[6] J. Ahn, C. Eastwood, L. Sitzki, P.D. Ronney, Gas-phase and catalytic combustion in heat-recalculating burners, Proc. Combust. Inst. 30 (2005) 2463–2472.

[7] A.C. Fernandez-Pello, Micro-power generation using combustion: issues and approaches, in: Proceedings of the 29th International Symposium on Combustion, Proc. Combust. Inst., vol. 29, 2002, pp. 883–889.

[8] A.H. Epstein, Millimeter-scale, micro-electro-mechanical system gas turbine engines, J. Eng. Gas Turbine Power 126 (2004) 205–226.

[9] S. Yuasa, K. Oshimi, H. Nose, Y. Tennichi, Concept and combustion characteristics of ultra-micro combustors with premixed flame, Proc. Combust. Inst. 30 (2005) 2455–2462.

[10] W.A. Sirignano, T.K. Pham, D. Dunn-Rankin, Miniature scale liquid-fuel film combustor, in: Proceedings of the 29th International Symposium on Combustion, Proc. Combust. Inst., vol. 29, 2002, pp. 925–931.

[11] W.A. Sirignano, S. Stanchi, R. Imaoka, Linear analysis of a liquid-film combustor, J. Prop. Power 21 (2005) 1075–1091.

[12] Y.C. Chao, G.B. Chen, T.S. Leu, C.Y. Wu, T.S. Cheng, Operational characteristics of catalytic combustion in a platinum microtube, Combust. Sci. Tech. 176 (2004) 1755–1777.

[13] D.C. Kyritsis, B. Coriton, F. Faure, S. Roychoudhury, A. Gomez, Optimization of a catalytic combustor using electrosprayed liquid hydrocarbons for mesoscale power generation, Combus. Flame 139 (1–2) (2004) 77–89.

[14] Y. Wang, V. Yang, R.A. Yetter, Combustion in meso-scale combustors. II: Numerical simulation, AIAA Paper 2004-0981.

[15] T.K. Pham, C. Mehring, D. Dunn-Rankin, W.A. Sirignano, Numerical study of a miniature liquid-fuel-film combustor, in: Proceedings of the Western States Section/Combustion Institute Spring Technical Meeting, Davis, CA, USA, March 29–30, 2004.

[16] T.K. Pham, N. Sarzi-Amade', D. Dunn-Rankin, W.A. Sirignano, Liquid film combustion in small cylindrical chambers, in: Proceedings of Fourth Joint Meeting of the US Sections of the Combustion Institute, Philadelphia, PA, March 20–23, 2005.

[17] T.K. Pham, Flame structure and stabilization in miniature liquid film combustors, Ph.D. Dissertation, UC Irvine, CA, USA, 2006.

[18] T.K. Pham, D. Dunn-Rankin, W.A. Sirignano, Flame structure in small-scale liquid film combustors, in: Proceedings of the 31st International Symposium on Combustion, Proc. Combust. Inst., vol. 31, 2007, pp. 3269–3275.

[19] N. Sarzi-Amade, Double wall miniature combustor with liquid fuel film, Tesi di Laurea, Politecnico di Milano (2005).

[20] D.A. Nield, Estimation of the stagnant thermal conductivity of saturated porous media, Int. J. Heat Transfer 34 (1991) 1575–1576.

[21] A.J. Katz, A.H. Thompson, Fractal sandstone pores: implication for pore conductivity and pore formation, Phys. Rev. Lett. 54 (1985) 1325–1328.