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Short communication

A chaotic heat-exchanger for PEMFC cooling applications

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Abstract

High-efficiency cooling systems are key points in PEMFC transport applications, as the volume constraints force the reduction of the stack size while increasing the power density. Moreover, to ensure an optimal electrochemical reaction over the whole polymer membrane surface and hence a maximum efficiency, the temperature field in the cell must be uniform and stay in a narrow range, around 80–90 °C. This study focuses on improving the thermal performance of heat-exchangers integrated in the bipolar plates of PEMFCs. The current design of the heat-exchangers in these applications is quite simple; cooling liquid (water) flows in straight channels or serpentines in the rear of the plates. The flow regime is laminar with a Reynolds number around 200. In order to enhance convective heat transfer, we propose here to promote three-dimensional flow inside cooling channels using a novel channel geometry that generates chaotic advection flow. However, to limit the size and the electric resistance of the bipolar plates, the thickness must be severely limited. This work concentrates on developing and characterizing heat-exchangers that can be easily reduced in size while preserving high thermal performance.

Keywords: PEM fuel cell; Bipolar plates; Thermal management; Heat-exchanger; Chaotic advection

1. Introduction

A PEMFC system produces as much electrical power as thermal power. In transport applications in which the power density of the fuel cell is relatively high, much thermal energy must be evacuated from the system. For a fuel cell stack, current heat-exchanger technology involves carving channels of a depth equal to half (or one-third) of the bipolar plate in which water flows, as shown in Fig. 1. The conventional heat-exchanger design in the bipolar plate is a simple network of parallel straight channels. The objective for future generations of PEMFC stacks, however, is to increase the working temperature from 80–90 to 120 °C, thus, the heat generated will increase greatly, current heat-exchanger design will rapidly reach its limits and a new design must be found. This is the objective of the present paper.

Among the parameters that influence the PEMFC performance, the temperature homogeneity over the membrane surface is crucial. The temperature must be high enough (typically over 80 °C) to ensure a good electrochemical reaction but low enough to prevent destroying the polymer membrane. Another important parameter is the electric resistance of the collector, it must be as low as possible in order to minimize ohmic losses, so that the heat-exchanger located inside the collector must be very thin.

Hydrodynamic conditions in the heat-exchanger in bipolar plates are such that the Reynolds number of the flow is around 200, which corresponds to the laminar regime. The anticipated future size reduction of PEMFC will reduce this Reynolds number. It is well known that the laminar regime is particularly unfavorable to high convective heat transfer. With these technological constraints in mind (temperature homogeneity, low thickness and laminar regime), we propose in this paper a type of geometry that can greatly improve current heat-exchanger performance for PEMFC cooling applications.

Section 1 of this paper presents the different geometries considered and Section 2 reports the numerical procedure used to evaluate the thermal performance of the different geometries. Section 3 presents the results and discussions and Section 4 is devoted to the conclusions and future work.

2. Studied geometries

Instead of considering the whole heat-exchanger with all its cooling channels, we evaluate the thermal performance of a single channel in the network. The next step (yet to be done) will

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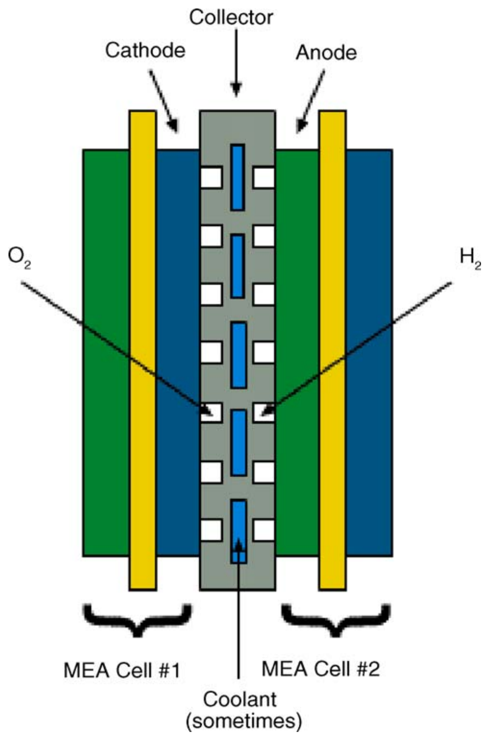

Fig. 1. Two elementary cells of a PEMFC stack with a plate heat-exchanger in the bipolar plate.

be to resolve the heat transfer problem for the entire network of several parallel channels separated by conductive materials in order to evaluate the thermal performance of the whole heat-exchanger. A study must be undertaken to find the best composition among the channels, this point is not addressed here.

Three different single channel geometries were considered, see Figs. 2–4. Fig. 2 shows the classical case of the straight channel. The cross-section is rectangular with an aspect ratio of 2, the channel height is 1 mm, the hydraulic diameter D_h is 1.33 mm. These values are typical of actual PEMFC stacks. The channel length is 8 cm.

Fig. 3 shows a zigzag channel with identical cross-section and hydraulic diameter. This channel is a succession of alternating 90° bends, so that the geometry is periodic in space with a period of two alternating bends. The unfold length of a period is equal to 18 mm. The ratio of the curvature radius to the hydraulic diameter in each bend is 5.5. With the considered Reynolds number, Re about 200, the resulting Dean number is: $D_e = Re\sqrt{\frac{D_h}{R_c}} = 96$. Under this hydrodynamic condition, a secondary flow consisting of two counter-rotating vortices, called Dean vortices, is generated in each bend under the action of centrifugal forces. This flow topology is known to lead to increased convective heat transfer over a straight channel [1,2].

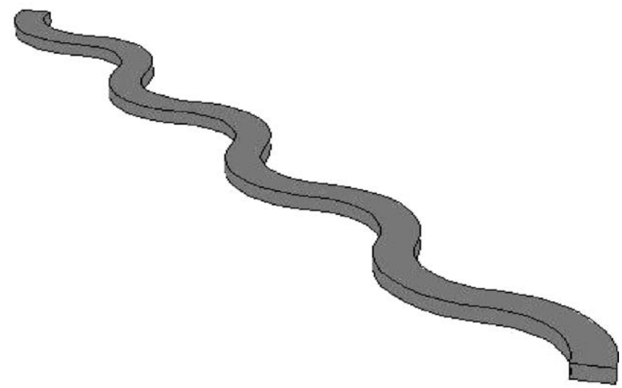

Fig. 3. Zigzag channel.

Fig. 4 shows the so-called “C-shaped” geometry, first introduced by Liu et al. [3]. This is a three-dimensional geometry contrary to the two previous ones. The term “C-shaped” is used because of the channel shape in the upper plane. Liquid flows in a straight channel in the lower plane. The channel cross-section here is rectangular with D_h = 1.33 mm, not square as in Ref. [3]. The basic geometry constituted of a “C-shaped” section in the upper plane and a straight section in the lower plane is periodically repeated in the longitudinal direction. The unfold length over one period is equal to the one for the zigzag geometry, i.e. 18 mm.

Using a Poincaré map, Ref. [4] shows that fluid particles flowing in this geometry may have chaotic trajectories. Some other parts of the flow do not exhibit this behavior because the flow is regular. The presence of chaotic trajectories in the flow greatly enhances mixing [5]. It is believed that this special feature will induce a large heat transfer coefficient, but this has not been proved so far.

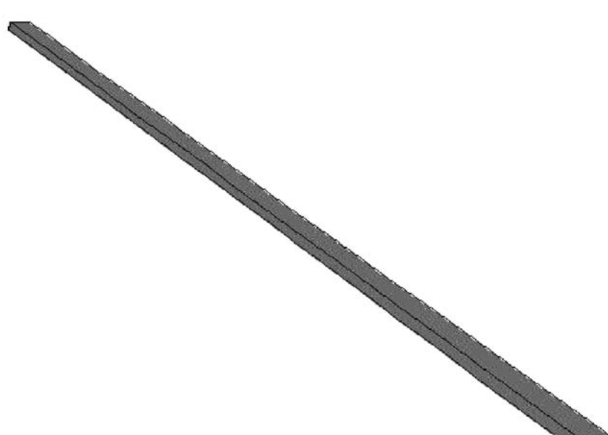

Fig. 2. Straight channel.

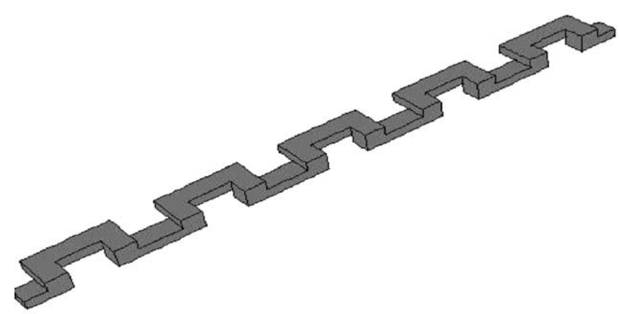

Fig. 4. “C-shaped” channel.

3. Numerical procedure

In order to evaluate the thermal performance of each geometry, numerical computations were performed using the commercial CFD code Fluent©. A uniform surface heat flux is imposed on a single channel surface. Current PEMFC stacks generate about 5000 W m^{-2}; in anticipation of future power density increases, we used $\varphi = 10{,}000$ W m^{-2} in our calculations. The inlet section of the channel is hydrodynamically fully developed. An approached fully developed velocity profile for rectangular ducts is imposed [6]. The forced convection regime is imposed and the Navier-Stokes equations are solved under laminar assumptions.

A parametric study was performed to identify the grid mesh that assures the capture of the thermal and velocity gradients near the walls. A spatial resolution of 40 × 40 meshes in the cross-section and 40 meshes in 2 mm in the z-direction was found fully satisfactory. The convergence of computations was stopped when the residues were less than 10^{-7}.

4. Results and discussion

The thermal performance of the single channel was first characterized in terms of the evolution along the curvilinear coordinate of the non-dimensional convective heat transfer coefficient, Nu, defined as:

$$Nu = \frac{\varphi}{T_m - T_w} \frac{D_h}{\lambda}$$

where T_m is the bulk mean fluid temperature over the cross-sectional area of the channel, T_w the channel wall temperature and λ is the thermal conductivity of the flowing fluid (here water, $\lambda = 0.6$ W m^{-1} K^{-1}). Fig. 5 presents the evolution of the Nusselt number with the curvilinear coordinate for these channel geometries. The Reynolds number of the flow is 200.

For the straight channel, a classical power-law decrease is found with an asymptotic value of $Nu = 3.00$. This value is consistent with the results in Ref. [7] and can be taken as a validation of the CFD computations.

A significant decrease in the Nusselt number is observed in the entrance region of the zigzag channel corresponding to a distance of about 0.005 m beyond the channel inlet. After the first period, the variation of the Nusselt number is periodic with a mean value of 8.2, a period equivalent to two alternating bends, and a maximum relative variation of about 1.1. These results are in good agreement with those in Ref. [1]. Heat transfer is higher in a curved tube than in a straight tube because of the presence of Dean vortices. Fig. 6 shows the secondary flow at the exit of the zigzag geometry.

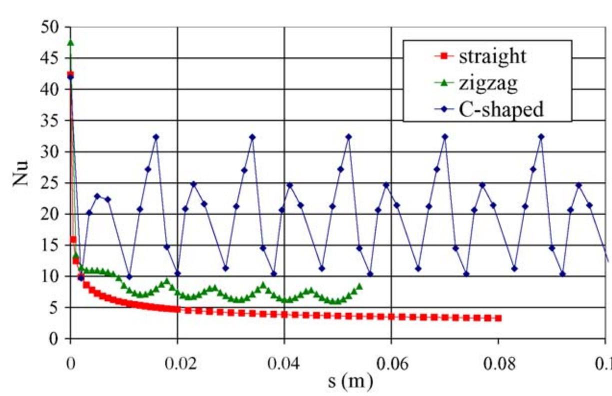

Fig. 5. Evolution of the Nusselt number with the curvilinear coordinate for the three channel geometries.

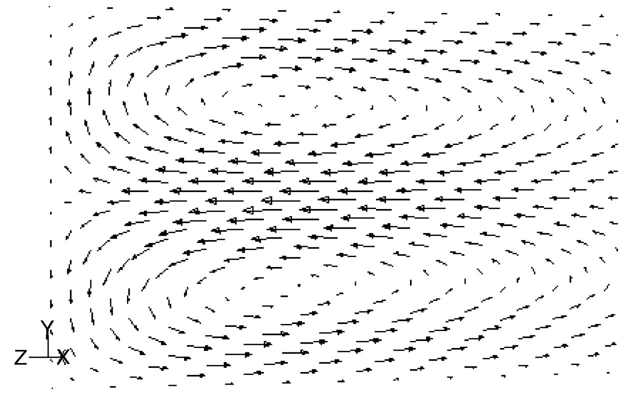

Fig. 6. Secondary flow at the exit of the zigzag channel.

Dean vortices transport cold particles from the center of the duct to the hot regions close to the wall. The secondary flow generates Nusselt number oscillations, though the amplitude and frequency of these oscillations depend on many parameters. Unlike a single curved duct, where the oscillations are attenuated in the asymptotic zone, in the zigzag geometry, the oscillations are kept constant by the zigzag perturbation. This geometry presents a clear improvement in thermal performance over the straight channel, the mean Nusselt number is about 2.5 times higher.

After the first period in the C-shaped channel, a periodic variation of the Nusselt number is observed around a mean value of 20.4. The period is equal to the spatial period of the channel and the maximum relative variation of the Nusselt number is about 10. The mean value remains constant over the computational domain and is much higher than that for the two other configurations. Even though the zigzag configuration increases heat transfer, Dean vortices divide the cross-section into two zones where the isotherms form closed curves. Fluid particles in the core of Dean vortices are prevented from approaching the hot walls. The flow is regular and no chaotic region exists. However, in the C-shaped geometry, the existence of chaotic advection regime can be shown. Actually, the chaotic advection is present in a flow when at least one of the following features exists [5]:

- strong stretching and folding of material lines and surfaces;
- sensitivity to initial conditions;
- production of transverse homoclinic and heteroclinic intersection.

The second feature was emphasized in the C-shaped geometry. Figs. 7 and 8 show the evolution of two-particle trajectories, initially close to each other and injected at the center of the

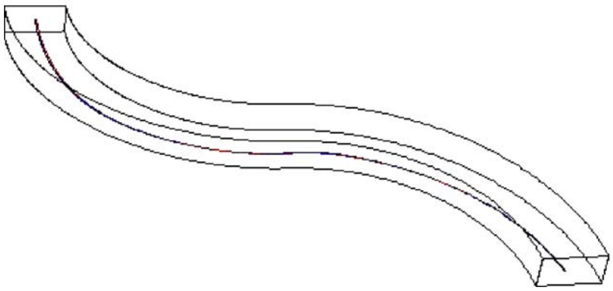

Fig. 7. Evolution of two-particle trajectories in the zigzag channel.

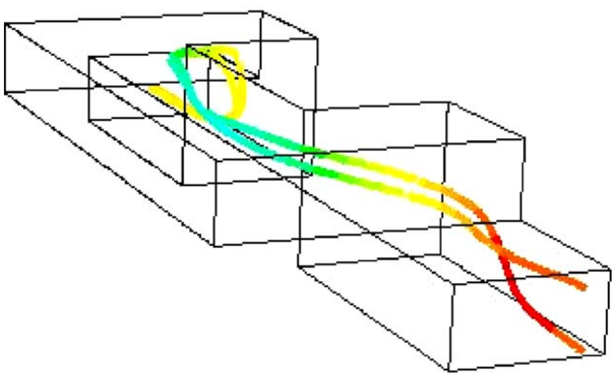

Fig. 8. Evolution of two-particle trajectories in the C-shaped channel.

inlet area of the zigzag and C-shaped channels. The distance separating the two particles is equal to the mesh size. In the zigzag channel, the flow is regular and the particle trajectories do not diverge. In the C-shaped channel, the particle trajectories can diverge rapidly. The distance separating two-particle trajectories was computed in several cross-section of the channel. The distance increases exponentially, a clear signature of chaotic advection, as in Ref. [4]. Consequently, in the C-shaped channel, the cold fluid in the center is no longer contained in the cold region. More cold particles of fluid visit hot regions close to the wall and global heat transfer is improved. Heat transfer is increased due to the kinematic effect and the mean Nusselt number retains the same value all along the geometry. These results are in good agreement with those in Ref. [8] for a similar type of chaotic geometry. Figs. 9 and 10 present the tempera-

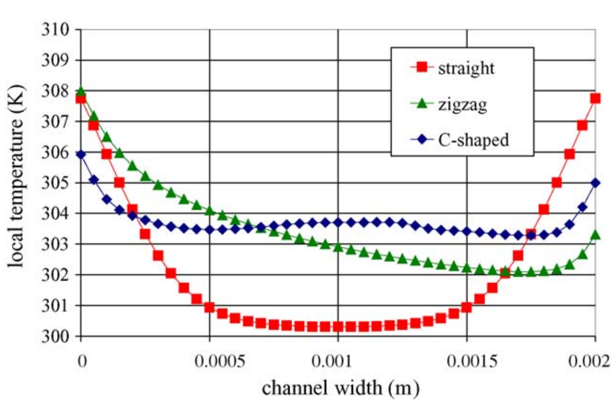

Fig. 9. Temperature profiles at 0.05 m from the inlet for the three geometries (horizontal direction).

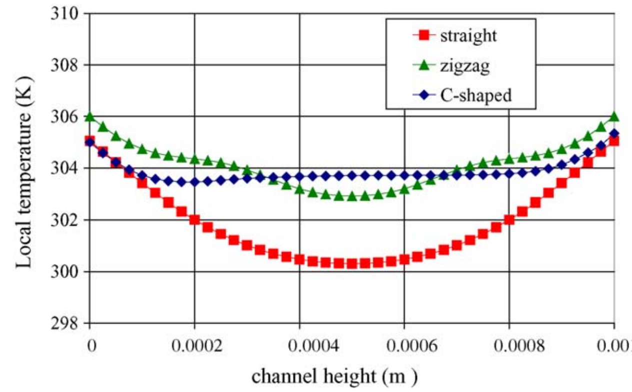

Fig. 10. Temperature profiles at 0.05 m from the inlet for the three geometries (vertical direction).

ture profile for the three geometries at 0.05 m from the inlet. In both horizontal and vertical directions, the temperature profile is more uniform for the C-shaped channel. The flatter temperature distribution in the C-shaped channel is attributed to chaotic trajectories of fluid particles due to the generation of chaotic advection regime.

Two points about the C-shaped geometry remain to be addressed: first, the pressure loss, which is undoubtedly higher than in the two other geometries, and secondly, the vertical extension, which doubles the space needed for the C-shaped geometry in the bipolar plate. The pressure loss is computed for the three geometries for a Reynolds number of 200 and is expressed in term of the friction factor defined as:

$$f = -\left(\frac{\mathrm{d}p}{\mathrm{d}s}\right)\frac{D_\mathrm{h}}{1/2\rho U_\mathrm{m}^2}$$

where ρ is the fluid density and $\mathrm{d}p/\mathrm{d}s$ is the local pressure gradient along the curvilinear coordinate of the channel. As this parameter is Reynolds dependent, it is often preferable to calculate the product $f \times Re$. Fig. 11 presents the evolution of the term *fRe* with the curvilinear coordinate for the three geometries for a Reynolds number of 200. For the straight channel, as an approached fully developed velocity profile has been imposed at the inlet boundary of the computational domain, the quantity *fRe* has an asymptotic behavior toward a value of 62. This value

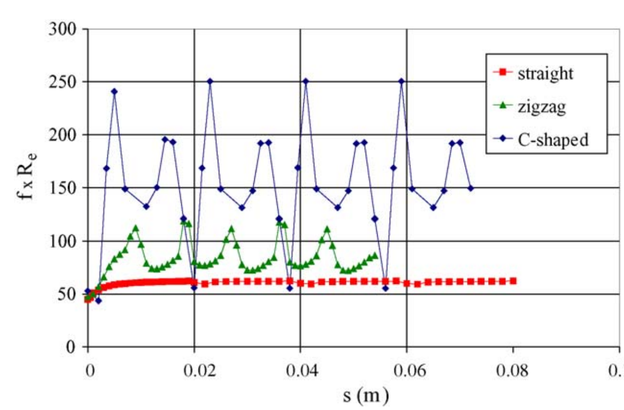

Fig. 11. Evolution of the term *fRe* with the curvilinear coordinate for the three geometries.

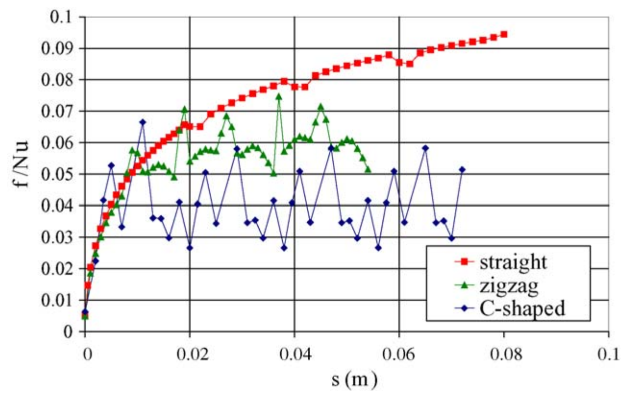

Fig. 12. Evolution of the ratio of the friction factor to the Nusselt number with the curvilinear coordinate for the three geometries.

is to be compared with the theoretical value of *fRe* for a fully developed pipe flow which is 62.2. This result gives another validation for the present computations. The presence of bends in the zigzag geometry increases the pressure loss over that in a straight geometry. After the establishment region, the mean value of the quantity *fRe* for the zigzag geometry is about 80. The sharp angles in the C-shaped geometry lead to a mean value of about 160. This value is 2.6 times higher than that in the straight geometry.

The question arises whether the higher Nusselt number for the C-shaped geometry is only due to the higher friction coefficient. In other words, is the heat transfer more intensified than the pressure loss in the C-shaped geometry? To answer this question, the ratio of the friction factor to the Nusselt number, $\frac{f}{Nu}$, is introduced. The geometry that produced the most efficient convective heat transfer is the one for which the maximum Nusselt number is coupled with the minimum friction factor, i.e. the smallest ratio $\frac{f}{Nu}$. Fig. 12 shows the evolution of this ratio along the three channels. In the first period of the zigzag and the C-shaped geometry, i.e. for $s < 0.018$ m, the ratio $\frac{f}{Nu}$ is comparable for the three considered geometries. However, for larger curvilinear coordinate, the C-shaped geometry presents a significant improvement in terms of effectiveness compared to the two others; the mean value of $\frac{f}{Nu}$ for the C-shaped geometry is less than half of that for the straight geometry. We thus see that with the C-shaped channel, more heat transfer is promoted than friction is produced.

It may be argued that the C-shaped geometry has a significant disadvantage in heat-exchangers for PEMFC applications because of its height; in the present study, it is actually twice the height of the straight or zigzag geometries. The first solution is to halve the size of the C-shaped geometry and keep the Reynolds number constant by doubling the mean velocity. If this solution is judged unsatisfactory, the Reynolds number can be reduced as much as the size. We can postulate that even if the Reynolds number is significantly reduced, the Nusselt number will still be higher than in the current straight channel because very high convective heat transfer has been obtained for a Reynolds number of 200 (about six times higher than that for the straight channel).

5. Conclusions and future work

This numerical study considered two channel geometries as alternatives to the one used in present heat-exchangers integrated in the bipolar plate of a PEMFC stack. The first alternative, the zigzag channel, leads to convective heat transfer coefficients about twice that of the straight channel, an improvement classically attributed to the presence of Dean roll cells in the flow. The second alternative is the C-shaped geometry, for which the convective heat transfer coefficient is about six times that of the straight channel. This high performance is explained as a consequence of chaotic regions in the flow. The chaotic behavior of fluid trajectories has been proven numerically. The pressure loss along the C-shaped channel is greater than in the straight channel. However, the ratio of the friction factor to the Nusselt number is lower, showing that the heat transfer enhancement is higher than the pressure loss increase.

Future work will focus on the thermal performance of the C-shaped channel at low Reynolds number and on finding alternative geometries to the C-shaped geometry in which large chaotic regions and low pressure losses are promoted.

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