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Hydraulic-thermal performance of vascularized cooling plates with semi-circular cross-section

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ABSTRACT

Analytical and numerical studies are conducted to investigate the hydraulic and thermal performance of new vascular channels with semi-circular cross-sections. An analytical model is built to minimize the flow resistance in vascularized cooling plates and then vascular designs are created by using the proposed model. The analytical results show that the dimensionless global flow resistance for the configurations with semi-circular cross-sections, both the optimized and non-optimized constructs, is significantly higher than it is for the configurations with circular cross-sections. A numerical model for three-dimensional fluid flow and heat transfer characteristics of vascularized cooling plates is also presented. Then, to validate the analytical model, the flow resistances predicted by the analytical model are compared with numerical data subject to a fixed volume and a fixed pumping power, and favorable agreements between the numerical and analytical results are obtained as system size increases. It is also shown that the thermal resistances for the first and second optimized constructs are closely competitive across all working conditions, whereas the best architecture in the non-optimized configurations is the third construct among the cooling plates.

1. Introduction

Over the last few decades, thermal management has become a crucial issues in the design of electronic or automotive components because temperature affects their performance and reliability. Especially with the decrease in size of thermal systems, it is vital to dissipate their heat. Even the traditional convective cooling of these systems may lose its performance in some cases [1] and thus should be managed.

Effective flow architectures of several types are proposed because the design of the flow architecture has a significant effect on the operating performance of engineering equipment [2]. A new research direction for global optimization is based on the constructal law [3–6]. Among the engineered flow architectures derived from the constructal law are the tree-shaped (dendritic) designs. Tree-shaped flow configurations offer maximum access between one point (inlet, or outlet) and an infinite number of points (area, volume) [7]. Numerous studies [8–19] have been conducted to examine the development of flow architecture using the constructal law. Wechsatol *et al.* [20] developed general rules to construct asymmetric trees. They also showed that asymmetric bifurcation provides lower flow resistance than symmetric bifurcation. We have characterized the thermo-hydraulic performance of vascular designs with circular cross-sections [2]. More recently, Kim *et al.* [21] showed that the main features of a steam generator can be determined based on the method of constructal design. They also showed that the total heat transfer rate increases proportionately to the length of the entire heat exchanger.

Polymer electrolyte membrane fuel cells (PEMFCs) are the typical examples where cooling is important: local hot spots due to an improperly designed cooling system can accelerate the mechanical damage of polymer electrolyte membranes, impairing the reliability of PEMFCs [22]. In reality, a temperature gradient of a magnitude of $\sim 5\,^{\circ}C$ has been found across the cathode. This temperature gradient has a significant influence on the water and heat transport in a PEMFC [23]. Thus, dealing with hot spots is one of the biggest challenges for cooling plates in a PEMFC.

Chen *et al.* [24] studied the cooling performance of several serpentine and parallel channel designs through a numerical simulation of fluid flow and heat transfer within the cooling plates. They concluded that the cooling effect of serpentine-type cooling modes could be better than that of parallel-type cooling modes in terms of temperature uniformity and maximum temperature. Similarly, Choi

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Nomenclature

A	area, m^2
C	Constant factor, m^2 s^{-1}, Eqs. (3) and (4)
d	elemental length scale, m
D_h	hydraulic diameter, m, Eq. (6) and Table 1
D_i	channel diameter of the ith channel, m
D_{1h}	hydraulic diameter of thin channels, m
D_{2h}	hydraulic diameter of thick channels, m
L_i	length of the ith channel, m
$\dot{m}$	mass flow rate, kg s^{-1}
$\dot{m}_i$	mass flow rate of the ith channel, kg s^{-1}
P	local pressure, Pa
$\boldsymbol{P}$	pumping power, W, Eq. (28)
p	perimeter, m, Eq. (5)
P_{in}	inlet pressure, Pa
P_{out}	outlet pressure, Pa
q	total heat imposed on the cooling plate, W, Eq. (27)
R	thermal resistance, K W^{-1}, Eq. (27)
Re	Reynolds number, Eq. (26)
Sv	svelteness number, Eq. (2)
T_m	mass-weighted average temperature, K, Eq. (25)
V_{in}	local mean velocity at inlet, m s^{-1}, Eq. (26)
V	element volume, m^3, Eq. (1)
V_c	total channel flow volume, m^3, Table 1
x, y, z	Cartesian coordinates
X, Y, W	inner dimensions of vascularized unit, mm, Figs. 1 and 2

Greek symbols

ΔP	pressure difference, Pa
ρ	fluid density, kg m^{-3}
ϕ	porosity, Eq. (1)
μ	fluid dynamic viscosity, kg s^{-1} m
ν	fluid kinematic viscosity, m^2 s^{-1}
ψ	non-dimensional global flow resistance, Eq. (15)

Subscripts

comp	compensation of pressure drop along the inlet and outlet channels
i	channel rank
in	inlet
max	maximum
out	outlet

Superscript

$\rightarrow$	Vector quantity

et al. [25], using computational fluid dynamics (CFD) simulations, examined the performance of cooling plates for the PEMFC by comparing the numerical results for serpentine channels with those for parallel channels. They also showed that modified serpentine-type and modified parallel-type models are more cooling than those that are unmodified. Yu *et al.* [26] studied several multi-pass serpentine flow-field designs in order to achieve better heat management by using cooling plates. They demonstrated that multi-pass serpentine flow-field designs had better cooling performance than did conventional serpentine flow-field, in terms of both the maximum temperature and temperature uniformity. Recently, Kurnia *et al.* [27] studied the thermal performance of parallel, serpentine, wavy, coiled and novel hybrid channels. Likewise, despite several studies of the hydraulic and thermal performance of cooling plates, none of these models report the optimized cooling channel architectures (or multi-scale based) based on the constructal law.

As noted above, while there have been some theoretical studies on the constructal law or on a multi-scale approach, the studies meant to validate previous research results or to optimize the geometric configurations of the cooling plates with semi-circular cross-sections for more practical applications have been limited. This research therefore uses analytical and numerical investigations to evaluate the hydraulic and thermal performance of cooling plates with semi-circular cross-sections which provide a solution to the fabrication of test sections. In general, the circular section does not represent the channel cross-sections being produced in engineering devices very well, while the stacked plates for etched passages is represented by a semi-circle. The impact of these characteristics on the relative heat transfer and pressure drop performance of such channels has not been quantified [28]. In this study, three types of design are considered on a square flat volume consisting of 10 × 10, 20 × 20, and 50 × 50 volume elements: a first-level construct (Fig. 1a), a second-level construct, and a third-level construct, respectively.

2. Analytical model and method

2.1. Geometry

The considered geometry in the study is a vascularized cooling body consisting of a square slab measuring $X \times Y$ and having thickness W, where W is the dimension of the solid body in the direction perpendicular to the plane $X \times Y$, as shown in Fig. 1. The size of the square domain is measured in terms of $N \times N$, where N is the number of small square elements counted along one side (see Fig. 1b). New vascular designs for the volumetric bathing of the smart structures with volumetric functionalities (self-healing, cooling) are optimized by using the methodology which was explored in the previous research work [7]: one channel size versus two channel sizes, increasing complexity (1st, 2nd, and 3rd

Table 1
Geometric dimensions for the constructal configurations.

System size	Complexity	V_c (m^3)	d (m)	D_h ($D_{1h} = D_{2h}$) (m)	D_{2h} (m)	D_{1h} (m)	Sv
10 × 10	1st	4×10^{-6}	10^{-2}	1.868×10^{-3}	2.856×10^{-3}	1.582×10^{-3}	6.3
	2nd	4×10^{-6}	10^{-2}	1.803×10^{-3}	2.418×10^{-3}	1.276×10^{-3}	6.3
	3rd	4×10^{-6}	10^{-2}	1.885×10^{-3}	2.251×10^{-3}	1.301×10^{-3}	6.3
20 × 20	1st	2×10^{-6}	5×10^{-3}	9.530×10^{-4}	1.833×10^{-3}	8.100×10^{-4}	7.9
	2nd	2×10^{-6}	5×10^{-3}	9.330×10^{-4}	1.566×10^{-3}	6.560×10^{-4}	7.9
	3rd	2×10^{-6}	5×10^{-3}	9.550×10^{-4}	1.417×10^{-3}	6.490×10^{-4}	7.9
50 × 50	1st	8×10^{-7}	2×10^{-3}	3.860×10^{-4}	1.029×10^{-3}	3.350×10^{-4}	10.8
	2nd	8×10^{-7}	2×10^{-3}	3.830×10^{-4}	9.000×10^{-4}	2.780×10^{-4}	10.8
	3rd	8×10^{-7}	2×10^{-3}	3.860×10^{-4}	8.010×10^{-4}	2.720×10^{-4}	10.8