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Numerical modeling and thermal optimization of a single-phase flow manifold-microchannel plate heat exchanger

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ABSTRACT

Manifold-microchannel technology has demonstrated substantial promise for superior performance over state of the art heat exchangers, with potential to reduce pressure drop considerably while maintaining the same or higher heat transfer capacity compared to conventional microchannel designs. However, optimum design of heat exchangers based on this technology requires careful selection of several critical geometrical and flow parameters. The present research focuses on the numerical modeling and optimization of a manifold-microchannel plate heat exchanger to determine the design parameters that yield the optimum performance for the heat exchanger. A hybrid method that requires significantly shorter computational time than the full Computational Fluid Dynamic (CFD) model was developed to calculate the coefficient of performance and heat transfer rates of the heat exchanger. The results from the hybrid method were successfully verified with the results obtained from a full CFD simulation and experimental work. A corresponding multi-objective optimization of the heat exchanger was conducted utilizing an approximation-based optimization technique. The optimized manifold-microchannel plate heat exchanger showed superior heat transfer performance over chevron plate heat exchanger designs.

1. Introduction

There is current interest among various industries for the development of advanced ultra-compact heat exchangers that can effectively exchange heat between two fluid streams having a low temperature difference. Plate heat exchangers are widely used for such purposes and are particularly useful for heat recovery and energy conversion-based applications. As noted in [1], one of the earliest references to a plate heat exchanger can be found in a U.S. patent by Drache in Germany in 1878. However, the first commercially successful design has been attributed to Richard Seligman in 1923 [2]. Since then considerable experimental and theoretical research on plate heat exchangers has been conducted [3–10]. Recent progress in plate heat exchanger technology, including thermal and hydrodynamic characterization, two-phase performance, and fouling and corrosion issues, has been summarized by Abu-Khader [11].

Most plate heat exchanger designs that yield a good thermal performance are disadvantaged by the high pumping power required for their operation. This is due to the long and restricted flow paths in these designs that result in a large pressure drop. Among the various existing designs, manifold-microchannel technology has yielded promising results that favor its development in next-generation plate heat exchangers.

Manifold-microchannel technology can be utilized on a plate heat exchanger by adding a set of manifolds over the microchannels as shown in Fig. 1. The fluid flow enters via manifold channels, is distributed across microchannels where the major heat transfer will occur, and then travels back to the manifold channels. The short flow path through the microchannels maintains the flow in the developing regime, which has better heat transfer than that of fully developed flow, and reduces pressure drop. It has been shown that manifold-microchannel designs can reduce single-phase flow pressure drop by a factor of n^2, where n is the number of channel divisions [12]. A summary of manifold-microchannel technology can be found in [13].

In order to increase the effectiveness of a manifold-microchannel heat exchanger, a multi-pass design can be implemented. In this design the heat exchanger is composed of several manifold-microchannel segments. The flow is forced to pass through the

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Nomenclature

A	cross sectional area [m^2]
a	constant described in Eq. (9) [s/m]
b	constant described in Eq. (9) [–]
C_1, C_2, C_3	constants described in Eq. (16) and (17) [–]
COP	coefficient of performance [–]
D	hydraulics diameter [m]
F	flow maldistribution factor [–]
f	Fanning friction factor [–]
K_1	constant derived in Eq. (11) [–]
K_2	constant derived in Eq. (12) [–]
K_3	constant derived in Eq. (13) [–]
k	thermal conductivity [W/mK]
H	height [m]
L	length [m]
m	constants described in Eq. (16) [–]
$\dot{m}$	mass flow rate [kg/s]
MAE	mean absolute error [%]
N_x	number of passes [–]
N_y	number of manifold channels [–]
N_z	number of stacks [–]
Nu	Nusselt number [–]
n	total number of microchannels in a single pass [–]
P	pumping power [W]
p	pressure [Pa], constant described in Eq. (17) [–]
p'	pressure, normalized $\left(p' = \frac{p}{\rho v_{in}^2}\right)$ [–]
Δp	pressure drop [Pa]
$\Delta p'$	pressure drop, normalized [–]
Q	heat transfer rate [W]
$Q/(V\Delta T)$	heat transfer density [W/(m^{3}K)]
Re	Reynolds number [–]
T	temperature [K]
ΔT	base and inlet channel temperature difference [K]
t	thickness [m]
V	volume [m^3]
v	velocity [m/s]
v'	velocity, normalized ($v' = \frac{v}{v_{in}}$) [–]
W	width [m]
x,y,z	Cartesian coordinate system [–]
x	distance at x-axis direction [m]
x'	distance at x-axis direction, normalized ($x' = \frac{x}{L_{mnd}}$) [–]

Greek symbols

α	W_{chn} to t_{fin} ratio [–]
β	chevron angle (corrugation inclination angle to the direction of the main flow) [°]
Γ	perimeter [m]
μ	dynamic viscosity [N s/m^2]
ρ	density [kg/m^3]
φ	enlargement factor [–]

Subscripts

1	manifold 1 (inlet)
2	manifold 2 (outlet)
base	microgroove base
chn	microchannel
fin	fin
in	microchannel inlet
out	microchannel outlet
mnd	manifold
spsm	single pass single manifold

microchannels multiple times and to come into contact with the heat transfer surface successively. A schematic drawing of a multi-pass heat exchanger design is shown in Fig. 1 (a).

One of earliest works on manifold-microchannel technology was first reported by Harpole and Eninger in 1991 [14]. Since then, there have been a few published studies on manifold-microchannel heat exchangers, such as experimental investigations by Cetegen [12], Kim et al. [15] and Kermani et al. [16], Boyea et al. [17], Jha et al. [18,19]. To numerically model a manifold-microchannel plate, both the manifold and microchannel sections must be considered, due to the interdependency of the fluid flow in both sections. In addition, it is necessary to determine the pressure drop across the manifold in order to calculate the total pumping power of the system. It is a challenge to model the entire heat exchanger, particularly designs with a large number of microchannels, on account of the long modeling and computational times associated with such models. For that reason, most of the previous numerical modeling-based studies, including those by Cetegen [12], Arie et al. [20], Copeland et al. [21], Ng and Poh [22,23], and Ryu et al. [24], have considered only a single microchannel segment without considering the pressure drop on the manifold channel. Wang et al. [25] and Boteler et al. [26] have simulated the entire heat exchanger, considering both the microchannels and manifold channel. However, their designs included a small number of microchannels, typically between 2 and 30 channels per design. Those models are not applicable for manifold-microchannel plate heat exchangers, which usually consist of hundreds of microchannels. In response to this difficulty, Escher et al. [27] reported a simplified approach, wherein multiple microchannels were collectively modeled as a porous medium, at the cost of losing accuracy. A summary of our literature survey on manifold-microchannel heat exchangers is presented in Table 1.

The present work proposes a novel approach for numerical modeling of the fluid flow and heat transfer characteristics in a manifold-microchannel plate heat exchanger. A hybrid approach was developed, wherein the transport phenomena in the microchannels were modeled by a CFD formulation, while those in the manifold were governed by an ordinary differential equation for one-dimensional flow. The manifold modeling method is an expansion of the work by Maharudrayya et al. [28]. The advantages of this approach compared to others are that the pressure drop calculation considers both the pressure drop in the manifold channel and the microchannels, the approach can be applied to systems with a high number of microchannels, and the computational time is much shorter than modeling the entire heat exchanger.

Heat exchanger design based on manifold-microchannel technology requires selection of several geometrical and flow parameters. In order to take advantage of the full potential of this technology, these parameters must be optimized properly. The current paper discusses a multi-objective optimization method of the manifold-microchannel heat exchanger to determine the parameters that yield maximum heat transfer and minimum pumping power. Numerous heat exchanger optimization efforts have been reported in the literature, such as those by Ryu et al. [29], Gopinath et al. [30], Sharma et al. [31], and Türkakar et al. [32]. However, multi-objective optimization can be very time consuming, considering the number of simulations that must be performed in conjunction with CFD. An approximation-based