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A "2.5-D" modeling approach for single-phase flow and heat transfer in manifold microchannels

Raphael Mandel, Amir Shooshtari, Michael Ohadi *

Small and Smart Thermal Systems Laboratory, Department of Mechanical Engineering, University of Maryland, College Park, MD 20742, USA

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ABSTRACT

A reduced-order "2.5-D" computational fluid dynamics (CFD) modeling approach for single-phase flow and heat transfer in manifold-microchannel heat exchangers was developed, and found to exhibit an order-of-magnitude reduced computational cost compared to a full 3-D simulation. Unlike previous approaches that neglect the convective terms in the momentum equations and assume fully developed flow, in the present work, the inertial terms in the momentum equations were retained, and a user-defined-scalar was used to calculate flow distance so that developing flow could be assumed. The 2.5-D model was then compared to a full 3-D CFD simulation, and was shown to be accurate as long as inertia is low enough to prevent the onset of secondary flows. The governing dimensionless parameters were defined, and the effect of each dimensionless parameter was investigated via parametric studies. Finally, a multi-dimensional parametric study was performed to determine the dimensionless parameter that governs the accuracy of the 2.5-D approach. In the end, it was determined that as long as dimensionless length is above 0.1, pressure drop can be predicted to within an average error of ~7% for any fluid, and heat transfer can be predicted to within an average error of 6% for water and air.

1. Introduction

The ubiquity of heat exchangers and their potential to affect system efficiency has made heat exchanger design of critical interest. Due to their high surface area to volume ratios, microchannel heat exchangers are capable of transferring a given amount of heat in a compact and lightweight design. However, their small hydraulic diameters create large pressure drops and require high pumping power, which can reduce system efficiency.

One way to minimize this effect is to divide the microgrooves into a system of parallel microchannels, thereby reducing both the flow length and the flow rate through each channel. Such a system, known as a manifold-microchannel system, is shown in Fig. 1. Due to the simultaneous reduction in both flow rate and flow length with each division, the pressure drop and pumping power tend to decrease proportional to the number of divisions squared [1]. Thus, microchannels with smaller hydraulic diameters can achieve the same pressure drop and pumping power as minichannels as long as the number of divisions is increased accordingly. In addition, due to the short flow lengths, manifold-microchannels can take advantage of thermally-developing flow, where a thin boundary layer results in a higher local heat transfer coefficient.

Manifold-microchannels have been extensively simulated in the literature, beginning with Harpole and Eninger [2]. They created a 2-D computational fluid dynamics (CFD) model, that neglected inertia and assumed fully developed flow and Nusselt numbers to simulate the effect of friction and heat transfer in the third dimension. They also simulated conjugate conduction using similarly defined source terms in the energy equation. Copeland et al. [3] used 3-D CFD to parametrically analyze the effects of the various geometric variables associated with manifold-microchannels. They neglected the effects of conjugate conduction (i.e. the solid domain), assuming instead an isothermal or isoflux boundary condition on the solid–liquid interface. Since then, numerous three-dimensional numerical studies have been conducted [4–6], including multi-objective optimization studies [1,7–10].

However, since no correlations exist to predict the pressure drop and heat transfer in manifold-microchannels, CFD is required to predict their performance. While conventional heat exchangers can be designed in a matter of hours using widely available correlations, manifold-microchannel heat exchangers require days to run the necessary CFD. Thus, the primary objective of this work was to develop a computationally-efficient, reduced-order model

* Corresponding author at: Center for Environmental Energy Engineering (CEEE), 4164C Glenn L. Martin Hall, Department of Mechanical Engineering, University of Maryland, College Park, MD, 20742, USA.

E-mail address: ohadi@umd.edu (M. Ohadi).

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Nomenclature

A	area, [m^2]
AR	aspect ratio (h_{ch}/w_{ch}), [–]
C_{fRe}	Poiseuille number ratio, [–]
C_{Nu}	Nusselt number ratio, [–]
C_p	specific heat, [J/kg K]
C_r	heat capacity rate ratio ($C_{min}/C_{\max}$), [–]
$\mathcal{D}$	parallel plates hydraulic diameter ($2w_{ch}$), [m]
D_h	hydraulic diameter $\left(\frac{2w_{ch}h_{ch}}{w_{ch}+h_{ch}}\right)$, [m]
$\vec{F}$	source term vector in momentum equations, [N/m^3]
fRe	Poiseuille number, [–]
h	heat transfer coefficient, [W/m^2-K]
IR	inlet ratio (L_{in}/L_{ch}), [–]
k	thermal conductivity of the fluid, [W/m K]
L^+	hydrodynamic dimensionless length, [–]
L^*	thermal dimensionless length, [–]
L_{ch}	length of microchannel, [m]
L_{in}	length of manifold inlet, [m]
L_{man}	length of manifold wall, [m]
L_{out}	length of manifold outlet, [m]
h_{ch}	height of channel, [m]
$\dot{m}$	mass flow rate, [kg/s]
N	number of nodes, [–]
Nu	Nusselt number, [–]
NTU	number of transfer units, [–]
P	pressure, [Pa]
Pr	Prandtl number, [–]
q''	wall heat flux, [W/m^2]
q'''	source term in energy equation, [W/m^3]
Q	total heat, [W]
Q_{max}	maximum possible heat, [W]
$\mathcal{R}e$	inlet, parallel plates Reynolds number, [$\rho V_{in}\mathcal{D}/\mu$]
Re_{ch}	microchannel Reynolds number, [$\rho V_{ch}D_h/\mu$]
S	flow length, [m]
T	fluid temperature, [K]
T_{wall}	wall temperature, [K]
U	velocity in x-direction, [m/s]
$\underline{V}$	velocity in y-direction, [m/s]
V	velocity vector, [m/s]
VR	velocity ratio, [–]
W	velocity in z-direction, [m/s]
w_{ch}	width of channel, [m]
x	coordinate direction, [m]
y	coordinate direction, [m]
z	coordinate direction, [m]

Greek symbols

ε	effectiveness, [–]
π	scalar, [–]
μ	dynamic viscosity, [Pa s]
ρ	density, [kg/m^3]
τ	wall shear stress, [Pa]

Subscripts/superscripts

app	apparent
ave	area- or mass-average
ch	microchannel
fd	fully developed
in	microchannel inlet
out	microchannel outlet
w	microchannel fin wall

capable of simulating single-phase, laminar flow and heat transfer in manifold-microchannels accurately, such that numerous simulations can be performed quickly. In addition, the model also provides insights into governing physical phenomena in manifold-microchannels by allowing physical phenomena to be isolated, resulting in an improved understanding of manifold-microchannel flow phenomena.

2. 2.5-D model

A "2.5-D" model of the manifold-microchannel flow configuration was created. Unlike Harpole and Eninger's model [2], which neglects inertia and assumes fully developed flow, the present model includes the effects of inertia and models developing flow by assuming that the flow develops along a streamline as if it were in a straight channel. The assumption of developing flow is equivalent to assuming a boundary layer profile in the third dimension—hence, our coining of the term "2.5-D".

2.1. Domain

Due to the symmetries present in manifold microchannel arrays, the domain for the manifold-microchannel simulations can be simplified to the unit-cell shown in Fig. 2(a) [1,7–14]. The definitions of the geometric variables are given in Fig. 2(b). The flow path is shown in Fig. 2(a). Flow enters from the top left in the velocity-inlet. The flow then impinges on the top of the microchannel fin and enters the microchannel, where the fluid absorbs heat. The fluid then turns upward and flows around the fin tip, and leaves through the pressure outlet.

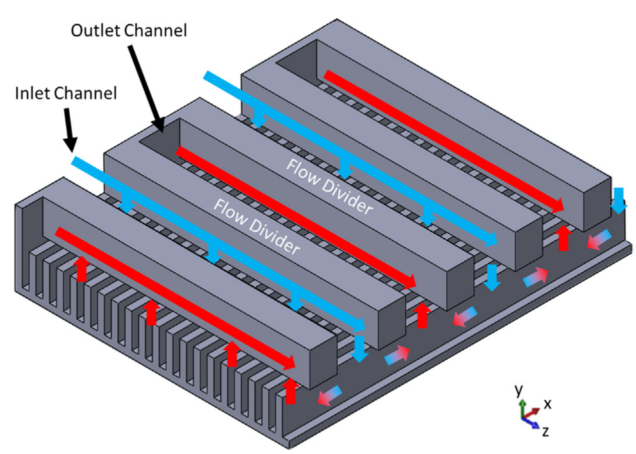

Fig. 1. Manifold-microchannel system.

2.2. Assumptions

The following assumptions and simplifications were made in the 2.5-D model:

(1) Steady-state, laminar, incompressible flow, with negligible effects of gravity on momentum and viscous dissipation on temperature
(2) Constant fluid properties
(3) Constant wall temperature