International Journal of Heat and Mass Transfer 53 (2010) 1004–1014

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Combined local microchannel-scale CFD modeling and global chip scale network modeling for electronics cooling design

R. Wälchli [a,b,*], T. Brunschwiler [a], B. Michel [a], D. Poulikakos [b]

[a] IBM Research GmbH, Zurich Research Laboratory, 8803 Rüschlikon, Switzerland
[b] Laboratory of Thermodynamics in Emerging Technologies, Department of Mechanical and Process Engineering ETH Zürich, 8092 Zürich, Switzerland

ARTICLE INFO

Article history:
Received 17 December 2008
Accepted 9 October 2009
Available online 11 December 2009

Keywords:
Microchannel
Liquid cooling
Heat transfer in electronics

ABSTRACT

Microchannel cold plates enjoy increasing interest in liquid cooling of high-performance computing systems. Fast and reliable design tools are required to comply with the fluid mechanics and thermal specifications of such complex devices. In this paper, a methodology accounting for the local as well as the device length scales of the involved physics is introduced and applied to determine the performance of a microchannel cooler. A unit cell of the heat transfer microchannel system is modeled and implemented in conjugate CFD simulations. The fluidic and thermal characteristics of three different cold plate mesh designs are evaluated. Periodic boundary conditions and an iteration procedure are used to reach developed flow and thermal conditions. Subsequently, two network-like models are introduced to predict the overall pressure drop and thermal resistance of the device based on the results of the unit cell evaluations. Finally, the performance figures from the model predictions are compared to experimental data. We illustrate the cooling potential for different channel mesh porosities and compare it to the required pumping power. The agreement between simulations and experiments is within 2%. It was found that for a typical flow rate of 250 ml/min, the thermal resistance of the finest microchannel network examined is reduced by 7% and the heat transfer coefficient is increased by 25% compared to the coarsest channel network. On the other hand, an increase in pressure drop by 100% in the case of densest channel network was found.

1. Introduction

In bipolar microprocessor technology, liquid cooling of electronics applications has been ubiquitous and highly needed in order achieve high power dissipation. This situation changed rapidly when the industry switched from bipolar to *complementary metal oxide semiconductor* (CMOS) chip technology. Due to low heat fluxes of the first chip generations, CMOS microprocessors immediately dropped the need for liquid cooling. However, aligning with Moore's law a continuous increase in power dissipation of high-performance computing systems has been taking place. Furthermore, the idea of collecting excess heat from data centers for reuse are expected to leverage low resistance liquid coolers for future thermal packaging solutions [1].

Microchannel cold plates have been demonstrated for the first time by Tuckerman and Pease with a cooling performance up to 790 W/cm^2 [2]. However, manufacturing difficulties, high pressure drop, and cost issues hindered this innovative concept for being implemented in electronics applications. Since the introduction of the flip chip and C4 technology that removed the requirement for wire bonding the entire backside of the die became available for heat removal. Thermal interface materials allow for a low heat flow resistance between the chips and the heat absorber device. Recent backside liquid cooling approaches ([3,4]) have demonstrated high cooling potential. In Brunschwiler et al. [3] direct water jet impingement with 50,000 nozzles arranged in a hierarchical manner was used to cool the backside of the chip. At a flow rate of 2.5 liter per minute (lpm) and a pressure drop of 0.35 bar a thermal resistance value of 15.0 K mm^2 W^{-1} was reached. However, direct contact of fluid with the microprocessor is delicate and requires advanced sealing and assembling. In the work of Colgan et al. [4], micromachining in silicon was used to build three-dimensional heat transfer structures of the size of 20 × 20 mm^2. At a flow rate of 1.5 lpm and a pressure drop of 0.4 bar a unit thermal resistance of 15.9 K mm^2 W^{-1} was achieved. A comprehensive study of high heat flux cooling technologies can be found in [5].

High convective heat transfer and low pressure drop is desired in order to meet performance requirements mentioned above. Precise prototyping and mass manufacturing technologies are needed

---

* Corresponding author. Address: IBM Research GmbH, Zurich Research Laboratory, Saumerstreet 4, 8803 Ruschlikon, Switzerland. Tel.: +41 44 724 8272; fax: +41 44 724 8958.

E-mail address: eto@zurich.ibm.com (R. Wälchli).

0017-9310/$ - see front matter © 2009 Elsevier Ltd. All rights reserved.
doi:10.1016/j.ijheatmasstransfer.2009.11.012

Nomenclature

A	elemental unit cell width (mm), or area (mm^2)
B	elemental unit cell fin width (mm)
C	elemental unit cell fin width (mm)
CMOS	complementary metal oxide semiconductor
c_p	specific heat capacity, 4182 (J kg^{-1} K^{-1})
DBC	direct bonded copper
D_h	hydraulic diameter (m)
FS	full scale
H	measure of height (m)
h	heat transfer coefficient (W m^{-2} K^{-1})
L	measure of length (m)
$\boldsymbol{L}$	translational vector (m)
$\dot{m}$	mass flow rate (kg s^{-1})
N	integer number of entities in x-direction (–)
p	pressure (Pa)
$\dot{Q}$	heat flux (W)
R	thermal resistance (K W^{-1})
T	temperature (K)
u	velocity component (m s^{-1})
V	volume (m^3)
$\dot{V}$	volumetric flow rate (m^3 s^{-1})
V1, V2, V3	mesh design version
W	measure of width (m)
$\boldsymbol{x}$	spatial vector (m)

Greek letters

β	coefficient of thermal resistance model function (K s W^{-1} m^{-3})
γ	coefficient of pressure drop model function (Pa s m^{-3})
δ	coefficient of pressure drop model function (–)
$\mathit{\Delta}$	Laplace operator (–)
ε	coefficient of thermal resistance model function (K W^{-1})
λ	thermal conductivity (W m^{-1} K^{-1})
υ	kinematic viscosity (m^2 s^{-1})
φ	porosity (%)
ϕ	coefficient of thermal resistance model function (–)

Subscripts/superscripts

$''$	area normalized (m^{-2})
–	mean
0	zeroth node
avg	average value in the centre of the flow channel
base	bottom side of cold plate/unit cell
cd	conductive
cell	pertaining to the *unit cell*
centre	pertaining to the central part of the cold plate
cv	convective
fl	fluid
H_2O	water
inlet	pertaining to the inlet port
n	network index
tot	pertaining to the entire cold plate
void	void space

to achieve well controllable and tunable performance figures of such microchannel cold plates. The cold plates studied in this paper utilize a *direct copper bonding* (DBC) process in order to comply with the requirements mentioned earlier. Copper sheets of 200 μm thickness are patterned by means of lithography and wet etching. A subsequent chemical oxidation process to build a defined copper oxide layer is applied. Finally, the layers are aligned in a stack and are fused at a temperature above the liquidus temperature of the copper oxide but below the solidus temperature of the copper. The result is a copper only and interface free bond due to the interlayer grain growth [6]. By alternating the patterns from one layer to the next, a porous, mesh-like heat transfer cavity with percolation paths can be created. A more detailed description of this technology can be found in [7]. Depending on the geometry and aspect ratio of a *unit cell* of this mesh the devices can be designed towards a particular operation point with respect to pumping power and cooling potential requirements.

In order to allow for fast and reliable engineering of liquid cooling systems, it is important to establish reliable design rules. Hence, fluidic and thermal performance figure estimates of microchannel cold plates are required. This typically implies an optimization of such microchannel cold plates with the correct engineering trade-off between frictional losses and convective heat transfer.

This article proposes a novel and reliable approach to characterize such mesh-type microchannel heat exchangers. The method is based on first solving at the *local level* the conjugate heat transfer problem of a mesh *unit cell* stack with computational fluid dynamics. Employing periodic boundary conditions the computational domain can be reduced to one *unit cell* stack only. This allows detailed studies and the optimization of heat transfer geometries within acceptable time frames and reasonable computation power. Physical data such as heat transfer coefficients and pressure drop are extracted for a flow rate range of 50–500 ml/min. This information can be subsequently used to estimate the performance figures of the entire cold plate device. To this end, a resistor network model for pressure drop analysis and a heat advection/convection model, both at the *device level*, are proposed to calculate the overall fluidic and thermal resistance of the cold plates. The thermal model also accounts for the effect of lateral heat spreading in the base due to the heating of the fluid from inlet to outlet. The model predictions are finally compared to experimental data.

2. Cold plate geometry

Fig. 1 describes the geometry of the heat transfer mesh and the layering principle of the entire cold plate, respectively. Fig. 1a is the top view of the elemental unit of the entire heat transfer space. By mirroring and stacking this entity a unit cell stack (*unit cell*), Fig. 1b, is created. This represents the simulation domain used in this work to calculate the performance quantities like pressure drop and heat transfer coefficient. The entire cold plate assembly is depicted in Fig. 1c. Several micromachined sheets of copper foils are piled and finally bonded together to form the entire cold plate. The mesh and cold plate manifold design allow multiple flow directions in a plane, i.e. being rotation-symmetric. This is needed where oscillating/reciprocating fluid flows take place. A potential application for self-contained, oscillating liquid cooling within a confined space is described in [8]. Four opening ports are implemented with tubes attached to the sidewalls of the cold plates. In the present work we are studying steady-state conditions with a continuous flow rate of water entering two ports from one side and leaving from the opposite side. Due to vertical and horizontal direction change of the streamlines a highly curling and tortuous flow profile with a resulting minimal thermal boundary layer can