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Computational analysis of new microchannel heat sink configurations

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ABSTRACT

Computational experiments were carried out on flow and heat transfer in four new microchannel heat sink (MCHS) configurations to compare their performance with the conventional MCHS. The new microchannel heat sinks simulated consist of four compartments with separate coolant inlet and outlet plenum for each compartment. The presence of several regions of developing flow in new designs result in higher Nusselt number and heat transfer rates. The substrate temperature gradients in new configurations are lower than that in conventional MCHS due to better distribution of coolant and recirculation. At the same pumping power, thermal resistances in the new designs are lower than the thermal resistances in conventional MCHS. The design may be further optimized by varying channel dimensions.

1. Introduction

Thermal management is important for effective functioning of devices comprising electronic circuits. The placement of a large number of electronic circuits in a device platform leads to generation of high heat fluxes. The failure to remove such high heat flux is likely to result in malfunction of circuits, which along with increased substrate temperature may lead to failure of the device. In 1981, Tuckerman and Pease [1] introduced the concept of microchannel heat sink (MCHS) for removing the heat generated in electronic circuits. Microchannels are closed channels for fluid flow, with hydraulic diameter ranging from a few tens to hundreds of micrometer. Due to smaller hydraulic diameters of microchannels, higher heat transfer coefficients can be achieved. Microchannel heat sinks offer several advantages such as higher heat transfer area per unit volume and lower coolant requirement in comparison with millimeter sized and conventional channels [1]. With the use of high-aspect ratio channels, heat transfer in microchannel heat sinks could be improved further [2,3]. Numerical investigations are normally performed to optimize channel dimensions and design, before verifying the same through heat transfer experiments. The theories developed for conventional channels can be utilized to predict hydrodynamics of microchannels also, provided the losses due to coolant entry from the inlet plenum to the microchannels and coolant exit from microchannels to outlet plenum are considered [4].

The need to improve heat transfer performance of conventional microchannel heat sink consisting of parallel microchannels has been realized. This has led to the development of heat sinks with micro-structured traverse flow [5], interruptions promoting thermally developing flow [6], modified coolant inlets and outlets [7], channels comprising bends & T-sections [8], increased fin volume [9], zig-zag microchannels [10], trapezoidal and triangular shaped channels [11], wavy microchannels [12], channels with reentrant cavities [13], etc., with better heat transfer performance. Brinda et al. [14] have reported 20% reduction in thermal resistance while utilizing a microchannel heat sink with ladder type connections of channels. A microchannel heat sink with straight microtubes with multiple inlets attached tangentially to the microtube was found to reduce the substrate temperature gradient [15], while the sink with bifurcated tree like microchannels too provided enhanced heat transfer [16]. The numerical investigations of double-layered MCHS have revealed their superior heat transfer performance, in comparison to that of single layer microchannel heat sink [17–19]. These attempts have resulted in increased heat transfer rates leading to lower thermal resistance.

The uniformity of substrate temperature is also an important factor to be considered in the design of MCHS. This could either be achieved by better distribution of coolant [20], or by distribution of heat flux [21] or through use of coolants with better transport properties such as nanofluids [22]. While using water as coolant and in circumstances where distribution of heat flux is impractical, the availability of low temperature coolant to various parts of sink is essential to maintain substrate temperature

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Nomenclature

Symbol	Meaning
A	channel flow area, m^2
C_p	specific heat, J/kg K
D_h	hydraulic diameter, μm
h	heat transfer coefficient, W/m^2 K
H	height, μm
k	thermal conductivity, W/m K
L	length, m
Nu	Nusselt number
p	channel wetted perimeter, μm
P	pressure, kPa
Pr	Prandtl number
q	heat flux at microchannel heat sink bottom plate, W/cm^2
Q	volumetric flow rate, m^3/s
Re	Reynolds number
R_{th}	total thermal resistance, K m^2/W
T	temperature, K
$T_{f,avg}$	volume averaged temperature of the fluid
$T_{w,avg}$	surface averaged temperature at the fluid solid interface
$\vec{V}$	fluid velocity, m/s
v	axial velocity, m/s
W	width, μm
x	thermally developing length, m
Greek	
θ	the maximum temperature difference on a heating surface per unit of heat applied, K m^2/W
μ	viscosity, kg m/s
ρ	density, kg/m^3
Subscript	
avg	average
b	microchannel heat sink bottom plate or substrate
c	channel
f	fluid
in	inlet
max	maximum
min	minimum
out	outlet
s	solid
w	wall

uniformity. This may be achieved with engineered design of single layer microchannel heat sink with provisions for better coolant distribution. However this aspect of coolant distribution to minimize the substrate temperature gradient has not been widely investigated. The presence of thermally developing regions and coolant mixing are expected to contribute to improved heat transfer. Hence in the context of present status of microchannel heat sinks, scope for improvement of performance of single layer microchannel heat sinks in terms of substrate temperature gradient is aplenty. The present work is an attempt in this direction.

2. New microchannel heat sink designs

In the present study, new perspective of dividing the heat sink into four quadrants with separate inlet and outlet as shown in Fig. 1, is likely to provide better thermal management in terms of substrate temperature gradient. The I-type MCHS design reported by Chein and Chen. [7] has been modified with respect to microchannel length and used as conventional heat sink in the present work is also shown in Fig. 1. In the work of Chein and Chen [7], the microchannel length was 10 mm, while the inlet and outlet plenums were 3 mm long. However in the present study, to make effective use of heat sink area, the length of inlet and outlet plenums was fixed at lower value (1 mm), which in turn led to increase of channel length by 4 mm (2 mm each near inlet and outlet plenum).

The overall length and width of the heat sink were 18 mm and 6.2 mm respectively. Channel width, channel height and fin width were fixed at 200 μm, 400 μm and 200 μm for all the four new heat sink configurations and the conventional heat sink. Each quadrant of the new MCHS consists of

(i) An inlet plenum for supply of coolant to microchannels located in that region.
(ii) An entrance region comprising of a coolant entry from the inlet plenum to parallel microchannels.
(iii) An exit region comprising of coolant exit from microchannels to the outlet plenum.
(iv) An outlet plenum to collect coolant leaving the microchannels for recycling.

The arrangement of microchannels in configuration A and configuration C are similar. However, the designs of inlet and outlet plenums in configuration A and configuration C are different. Similarly, while the arrangement of microchannels in configuration B and configuration D are similar, the designs of inlet and outlet plenums of these two configurations (B & D) are different. The performance of each of the four designs has been evaluated numerically to compare the designs on the basis of substrate temperature gradient, total thermal resistance and pumping power.

3. Model assumptions & governing equations

The following assumptions have been made in the present study:

(1) Both fluid flow and heat transfer are steady and three-dimensional.
(2) The fluid is single phase, incompressible and the flow is laminar.
(3) The properties of fluid and heat sink material are independent of temperature.
(4) All the surfaces of heat sink exposed to surroundings are assumed to be insulated except the bottom plate of heat sink where a constant heat flux boundary condition is imposed simulating the heat generation.

The continuity, momentum and energy balance equations for the current problem can be written as follows [23]:

Fluid flow:

$$\nabla \cdot \vec{V} = 0 \tag{1}$$

$$\rho(\vec{V} \cdot \nabla\vec{V}) = -\nabla P + \mu\nabla^2\vec{V} \tag{2}$$

Energy in fluid flow

$$\rho C_P(\vec{V} \cdot \nabla T) = k\nabla^2 T \tag{3}$$

Energy in solid part of heat sink

$$K_s\nabla^2 T_s = 0 \tag{4}$$

The boundary conditions are as follows: