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Experimental and numerical investigation of a mini channel forced air heat sink designed by topology optimization

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ABSTRACT

This work presents a method of designing an air heat sink with forced convection by topology optimization. Both pressure drop and heat transfer performances are evaluated. To reduce computational cost, a 2D two-layer model is first developed and implemented in COMSOL Multiphysics to represent three-dimension fully conjugate heat transfer modeling. It has been shown to be accurate in temperature field prediction and able to capture trends of pressure drop variation. Through a multi-stage optimization process, a non-conventional fin structure is created. The optimized structure is then manufactured and experimentally validated. Compared to a conventional straight channel heat sink, the topology optimized heat sink can achieve lower junction temperatures with the same pumping power or requires lower pumping powers for maintaining the same junction temperature. Furthermore, full 3D numerical analysis by ANSYS Fluent is performed to study the detailed characteristics of the topology optimized heat sink. It shows that the non-conventional layout of the fins introduces strong mixing effect, continuous boundary layer interruption and local high speeds, which all contribute to heat transfer enhancement.

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1. Introduction

As the power density of electronic devises is rapidly increasing, thermal management has become a major challenge today [1]. The heat flux of modern microprocessors could easily go up to the order of 10,000 W/m^2. For example, Intel Xeon Processor E5-2600 v4 processors have to dissipate around 100 W of heat within a substrate of 52.5 × 45 mm [2]. A widely adapted solution is attaching chip packages onto heat sinks with straight fins and passively or actively cooling them using air. In research, the micro- and mini-channel heat sinks have gained its popularity as a promising and reliable technique for high density power dissipation. In the pioneering work of Tuckerman and Pease [3], a 50 μm wide and 300 μm deep microchannel heat sink was capable of dissipating a power density of 790 W/cm^2 with a junction temperature of 71 °C. However, the initial microchannel concept had several evident drawbacks, such as large pressure drop penalty and significant lateral temperature gradient. To circumvent these disadvantages and further enhance the thermal performance, many different fin and channel structures have been investigated.

As highlighted by Steinke and Kandlikar [4], channel curvature could skew the traditional parabolic velocity profile to enhance heat transfer. Following this idea, wavy channels were numerically studied by Sui et al. [5] and proved to increase the heat transfer coefficient by 153% while increasing the friction factor only by 54% at R_e = 800. Furthermore, the wavy channel concept was combined with secondary branches, which leads to a performance boost as high as 190% [6]. Similar secondary flow concept was applied in a new sectional oblique fin design proposed by Lee et al. [7]. It was shown to increase heat transfer performance by 50–60% with almost no additional pressure drop penalty compared to conventional microchannels.

Inspired by natural structures, Bejan and Errera [8] proposed the tree-shaped channel network based on constructal theories. Similar fractal-like flow network was studied by Pence [9] and Chen and Cheng [10]. Except for fin structures, there were also several reports about designing the whole heat sinks including manifolds. Sharma et al. [11] demonstrated a novel concept for energy efficiency-hotspot-targeted liquid cooling of multicore microprocessors. Through the introduction of the flow-throttling zones, higher flow rates were guaranteed over hotspots. As a result, chip temperature non-uniformities were greatly reduced. Other system level designs can also be found in [12–16].

The designs mentioned above were all initiated from intuition or experience. With the increase in computational power, it has become a popular practice to optimize designs through numerical simulations. In general, there are three levels of optimization: size,

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Nomenclature

A	area (m^2)
cp	specific heat ($J/kg \cdot K$)
dz	thickness (m)
D	diameter (m)
h	convection coefficient ($W/m^2 \cdot K$)
H	height (m)
I	current (A)
k	thermal conductivity ($W/m \cdot K$)
$\dot{m}$	mass flow rate (kg/s)
P	pressure (Pa)
$\dot{q}$	heat transfer rate (W)
$\ddot{q}$	heat flux (W/m^2)
$q_f/q_k/q_h$	convexity parameter
Q	power (W)
R	thermal resistance (°C/W)
SC	straight channel
TO	topology optimized
T	temperature (°C)
u	velocity vector
V	voltage (V)
v	velocity (m/s)
D_h	hydraulic diameter (m)
Re	Reynolds number
Nu	Nusselt number
f	friction factor
L	length (m)

Greek symbols

Δ	gradient (difference)
α	friction coefficient
μ	dynamic viscosity ($Pa \cdot s$)
ρ	density (kg/m^3)
γ	local design variable
η	fin efficiency

Subscripts

ave	average
b	base
bt	bottom layer
f	fin
fl	fluid
HS	heat sink
in	inlet
min	minimum
max	maximum
out	outlet
s	solid
top	top layer
th	thermal
wet	wet surface
sec	sectional area

shape and topology optimization. Size optimization requires well-predefined geometric parameters such as length, width and depth. It aims to find a set of dimensions that will maximize the overall performance. As in [6], various aspect ratios were examined in order to find the optimum values of the key dimensions. In shape optimization, on the other hand, geometric boundaries are normally defined by curve functions. Boundaries are adjusted as the functions change. Hilbert et al. [17] presented a successful implementation on the blade shape of a heat exchanger. Finally, in topology optimization, size, shape and topology are optimized simultaneously [18]. It does not require predefined geometric parameters and allows the formation of new boundaries. Therefore, it is the highest level of optimization. The difference between these three optimization methods is shown in Fig. 1: in the optimization of a fin and tube heat exchanger, the diameter of the tube is the parameter to be defined and it is adjusted in size optimization; in shape optimization, the tube may not remain as a circle but morph to other geometries; in topology optimization, no definition of prescribed tube geometry is required and novel geometries may be generated.

Topology optimization was first introduced by Bendsøe and Kikuchi [19] in 1988 for the design of mechanical elements that can withstand given loads. An overview of this method was provided in the monograph by Bendsøe and Sigmund [20]. In the monograph, topology optimization was described as a material distribution method in which a design domain was discretized and each element of the domain was decided to be occupied as either void or material. In 2003, this method was extended by Borrvall and Petersson [21] to include physics of fluid flow for the first time, a technique termed as topology optimization of fluids. In this method, each element of the discretized design domain was assigned a local design variable γ which varied continuously from 0 to 1. The authors then introduced a friction coefficient α, which was a function of the local design variable γ. The product of α and velocity u (αu) was then introduced to the momentum equation as an additional friction force term. For $\gamma = 0$, $\alpha \approx \infty$, resulting in a reduction of local velocity to 0; for $\gamma = 1$, $\alpha = 0$, retaining the original momentum equation. Extreme values of γ, 0 and 1, represented solid phase and fluid phase, respectively. The initial problem of optimization was then transferred to the optimization of values of γ in each element of the design domain. With minimizing pressure drop as the objective, the method was successful implemented in the design of a diffuser, a pipe bend, a rugby ball and a double pipe. Soon after this pioneering work, Gersborg-Hansen et al. [22] extended it from Stokes flow to incompressible laminar viscous flow at low-to-moderate Reynolds numbers. Several other studies also reported successful implementation of this method in designing various flow structures [23–25]. As a real application, a microfluidic mixer was designed with a 70% increase in mixing accompanied by a 2.5 times increase in the pressure drop penalty [26].

The method is also applicable to cases of heat conduction where the distribution of high conductivity material is to be optimized. As reported in [27], this method generated a leaf like conductance path for a volume to point heat dissipation problem. At the system design level, distribution of insulation material and thermal connection between a thermoelectric cooler and a structural chassis were optimized to ensure sufficient cooling for temperature sensitive electronic components in a downhole oil well intervention tool [28]. The optimized result was then validated by experiments.

In cases of natural convection where fluid flow and heat transfer are combined together, it is worth mentioning the work by Alexandersen et al. [29]. In their work, various novel heat sink structures were generated under different Grashof numbers. However, due to the three-dimensional simulation of conjugate heat transfer, the computational cost was considerably high and required a cluster for implementation.

Finally, there were also several attempts in utilizing topology optimization in heat sink design with forced convection. Oevelen and Baelmans [30] first introduced a 2D numerical model to solve

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