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Experimental investigation on thermal and hydraulic performance of microchannels with interlaced configuration

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ABSTRACT

In this study, a novel interlaced microchannel with a "cold water-hot water-cold water" counterflow arrangement was designed. The influences of microchannel configurations on the thermal and hydraulic performance were studied by comparing the proposed microchannel configuration with parallel and traditional spiral configurations. The results showed that the effective heat transfer area of the interlaced microchannel was 6.4 and 8.4 times that of the parallel and spiral configurations, respectively. For interlaced microchannels, the maximum temperature difference between the cross sections was 0.07 °C, and the temperature rise along the flow direction was only 6 °C. When the Reynolds number was 492, the Nusselt number of the interlaced microchannels was 2 and 10 times that of the parallel and spiral microchannels, respectively. The heat transfer performance of interlaced microchannels was improved by 83.46% compared with that in the literature. The influence of microchannel configurations on the pressure drop and the entrance length were negligible. The interlaced microchannel exhibited its lowest thermal resistance of 0.015 °C/W and lowest entropy production of 22.6 W/ °C at a Reynolds number of 492. The heat transfer enhancement coefficient of the interlaced microchannel and parallel microchannel were 5 and 2.8 times that of the traditional spiral microchannel, respectively. The maximum heat load of loop heat pipe was enhanced by 4 times with the integration of interlaced microchannel as the condenser.

1. Introduction

The demand for heat dissipation in high heat flux microelectronics such as processors and high-power Light Emitting Diodes (LEDs) has become a bottleneck in the development and reliable operation of electronics with the rapid development of micro-nano technology [1]. Microchannels have been widely used in the fields of microelectronics, aerospace, and the chemical industry because of their advantages of small volume, high heat transfer performance, and low manufacturing cost. However, the temperature rise along the flow direction produces local hot spots on the chip surface, which severely shortens the service life. Therefore, extensive research has been carried out to improve the thermal performance of microchannels, such as optimizing the microchannel geometric design and improving the thermophysical properties of the working fluid.

Optimizing the microchannel cross-sectional structure is the most effective method. The heat transfer area is increased to strengthen the heat transfer and reduce the flow resistance with a trapezoidal microchannel [2]. The application of a Ω-shaped microchannel effectively increases the heat transfer and hydraulic performance [3]. Some special microchannel configurations such as sinusoidal microchannels [4], convergent microchannels [5], fractal microchannels [6], and Y-shaped microchannels [7] have been proposed to redevelop the thermal boundary layer and enhance the heat transfer performance.

By adding pin fins to the microchannel, the disturbance is enhanced and the heat transfer performance is improved [8]. A microchannel with zigzag [1] or semicircular [9] side walls breaks the thermal boundary layer and increases the heat transfer area. In addition, the heat transfer area and disturbance of the liquid are increased by filling the porous material into the microchannel [10]. Although the addition of microfin pins, special side wall structures, and porous material strengthen the heat transfer performance, the pressure drop increases. Therefore, the concept of a double-layered microchannel has been proposed [11], which not only maintains high heat transfer performance but also reduces the pressure drop along the flow direction [12]. In order to further decrease the pressure drop, methods such as optimizing the entrance to the microchannel using a genetic algorithm [13], changing the inlet number, and decreasing the flow length of the

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Nomenclature

A	area, m^2
AR	area ratio
D	diameter, m
H	height, m
K	pressure loss coefficient
L	length, m
N	microchannel number
Nu	Nusselt number
ΔP	pressure drop, Pa
Po	Poiseuille number
Pr	Prandtl number
Re	Reynolds number
R	thermal resistance, °C/w
S	entropy roduction, W/°C
T	temperature, °C
ΔT	temperature difference, °C
W	width, m^3
V	velocity, s
Z	ratio of width to depth
c_p	constant pressure specific heat, J/(Kg·K)
c_r	correction factor of radius
c_t	correction factor of temperature
f	fanning friction factor
h	heat transfer coefficient, W/(m^2·K)
k	heat conductivity coefficient, W/(m·K)
m	fin parameters
$\dot{m}$	mass flux, kg/m^2·s
q	heat load, W
q''	heat flux, W/m^2
r	radius, m
t	thickness, m
y	longitudinal separation, m

Subscripts

abs	absorb
b	base
c	cold
cf	cold fluid
ch	channel
con	contraction
$cond$	conductive
$conv$	convective
cap	capacitive
cs	cross sectional
enl	enlargement
e	entrance
eff	effective
f	fluid
fri	friction
h	hydraulic
ht	heat
hf	hot fluid
im	inlet manifold
om	outlet manifold
l	large
m	manifold
$rele$	release
r	reference
s	small
t	total
w	wall

Greek symbols

μ	dynamic viscosity, Pa·s
ρ	density, kg/m^3
η	coefficient
ν	specific volume, m^3/kg
σ	cross sectional area ratio

Abbreviation

IM	interlaced microchannel
PM	parallel microchannel
SM	spiral microchannel
HFM	heat flux method
EFM	empirical formula method
$EGHT$	entropy generation due to heat transfer
$EGPD$	entropy generation due to pressure drop
$C\text{-}H\text{-}C$	cold water-hot water-cold water
LHP	loop heat pipe

working fluid [14,15] have been proposed by homogenizing the distribution of the fluid.

The optimization of the thermophysical properties of working fluids has attracted significant interest. Owing to the excellent electrical properties of dielectric fluid, the thickness of liquid film is reduced and the two-phase heat transfer performance is strengthened [16]. Liquid metal with a low boiling point and high thermal conductivity is a promising working fluid for the cooling of electronic chips [17]. The thermal conductivity of the fluid is effectively improved by adding nanoparticles such as CuO [18], Al_2O_3 [19,20], and TiO_2 [21], and the thermal boundary layer was broken to enhance heat transfer between the fluid and the wall. However, the deposition of nanoparticles limits the application of nanofluids [22]; therefore, it became of great interest to study the influence of nanoparticle concentration on heat transfer performance [23,24].

Although the microchannel has been widely investigated, studies on the interlaced arrangement of channels have rarely been reported. In this study, a novel interlaced microchannel (IM) with hot- and cold-fluid channels located on the same side of the microchannel substrate was designed. For comparison, a parallel microchannel (PM) and traditional spiral microchannel (SM) were manufactured. The effects of the microchannel configurations on the thermal and hydraulic performance were studied.

2. Experimental

2.1. Microchannel design

In this study, three kinds of microchannels with IM, PM, and SM configurations were manufactured, as shown in Fig. 1. The IM and PM configurations were rectangular microchannels. The depth-width ratio of the IM was 3:1, and hot and cold water were counterflowed on the same side of the microchannel substrate in a “C-H-C” arrangement. The hot and cold water flow direction and heat transfer direction are shown in the left view of Fig. 1(a). The heat transfer mainly occurred on the two side walls of the microchannel to increase the heat transfer area. The inlet and outlet of the hot and cold water were designed with a 60° inclination to reduce the local pressure loss. In this work, an embedded sealing grooves was designed. The sealing grooves were completely coupled with the microchannel shape as designed on the top cover, and then were embedded approximately 0.2 mm deep in the microchannel substrate. Sealant was applied inside the grooves during installation to