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Comparison of the optimized thermal performance of square and circular ammonia-cooled microchannel heat sink with genetic algorithm ☆

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ABSTRACT

Minimization of the thermal resistance and pressure drop of a microchannel heat sink is desirable for efficient heat removal which is becoming a serious challenge due to the demand for continuous miniaturization of such cooling systems with increasing high heat generation rate. However, a reduction in the thermal resistance generally leads to the increase in the pressure drop and vice versa. This paper reports the outcome of optimization of the hydraulic diameter and wall width to channel width ratio of square and circular microchannel heat sink for the simultaneous minimization of the two objectives; thermal resistance and pressure drop. The procedure was completed with multi-objective genetic algorithm (MOGA). Environmentally friendly liquid ammonia was used as the coolant and the thermophysical properties have been obtained based on the average experimental saturation temperatures measured along an ammonia-cooled 3.0 mm internal diameter horizontal microchannel rig. The optimized results showed that with the same hydraulic diameter and pumping power, circular microchannels have lower thermal resistance. Based on the same number of microchannels per square cm, the thermal resistance for the circular channels is lower by 21% at the lowest pumping power and lower by 35% at the highest pumping power than the thermal resistance for the square microchannels. Results obtained at 10 °C and 5 °C showed no significant difference probably due to the slight difference in properties at these temperatures.

1. Introduction

Since the innovation work by Tuckerman and Pease [1], the microchannel heat sink (MCHS) attracted great interest in recent years because of its ability to dissipate a large amount of heat from a small area. The excellent capability of a MCHS with many parallel microchannels machined above or below the substrate of electronic chips has been demonstrated to be effective in removing the high heat flux generated in very large-scale integration (VLSI) integrated circuits (ICs). The MCHS has made it a favorite choice for electronic cooling systems' devices in microprocessors, radars, laser diode arrays and high-energy laser mirrors [2]. The rectangular MCHS has perhaps received the most attention due to ease of machining and its proven excellent thermal performance compared to others [3–6]. Research into increasing the performance of a MCHS using different channel geometry, coolants, flow regime, structural materials, and optimization tools have been completed with different outcomes presented and discussed [7]. Lately, research has been focused on the performance of nanofluids as coolants as well [6,8,9]. Current awareness of the advantages of compact heat exchangers with their smaller evaporators and condensers, have encouraged more studies on the flow pattern and heat transfer of these microchannel cooling systems. With concerns over effects from the non-environmentally friendly coolants, investigations have focused onto the application and performance of alternatives in the available MCHS as well as in the compact refrigeration and air-conditioning industries [10–12]. This present study has been initiated to look at the aspect of thermal management of the microchannels with a new environmentally friendly coolant i.e. ammonia. Ammonia has zero ozone depleting potential (ODP) as well as zero global warming potential (GWP)

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Nomenclature

A	area (m^2)
c	specific heat (kJ kg^{-1} K^{-1})
g	acceleration due to gravity (m s^{-2})
G	volumetric flow rate (m^3 s^{-1})
h	Heat transfer coefficient (W m^{-2} K^{-1})
D_h	hydraulic diameter (m)
f	friction factor
j	geometry parameter
k	thermal conductivity (W m^{-1} K^{-1})
L	microchannel heat sink length (m)
n	number of microchannels
Nu	Nusselt number
m	number of objective function
p	pressure (kPa)
P	pumping power (kW)
q	heat flux (kW m^{-2})
R	thermal resistance (W^{-1} K)
Re	Reynolds number, $Re = \frac{\rho V D_h}{\mu}$
t	thickness (m)
$\bar{x}$	vector contain design variables
V	velocity (m s^{-1})
w	channel (wall) width (m)

Greek letters

β	ratio of wall width to channel diameter (or width)
μ	dynamic viscosity (N s m^{-2})
ρ	density (kg m^{-3})

Subscripts

avg	average
cap	capacity
cir	circular
cond	conduction
conv	convection
eff	effective
f	saturated liquid
hs	heat sink
min	minimum
sq	square
w	wall
p	constant pressure for heat capacity

and although has been around since the early twentieth century, it has only been explored as a potential coolant in microchannels recently. However, the behavior of ammonia or any coolants in macrochannels do vary from that in microchannels and thus it is important to have available properly derived properties if a MCHS performance is to be determined to enable us to identify the coolant capability under micro-size conditions.

From about 2008, researchers have looked into the global search afforded by evolutionary algorithms for faster investigations of MCHS optimized designs particularly with new coolants. Exploration of the cooling capabilities of newly developed environmentally refrigerants is better completed with genetic algorithm (GA), an evolutionary algorithm, due to its proven record in its stochastic approach search for sets of optimal solutions. In the present work, optimization of the circular and square MCHS has been completed using GA with the environmentally friendly liquid ammonia, whose properties have been obtained experimentally. GA has been shown to be able to determine the optimized performance of a rectangular MCHS with textbook data as well experimental data [13]. Past optimization of the MCHS have been based on the experimental or/and numerical methods approach [14–18]. Unlike the limited optimization outcomes with experiments and numerical simulation with discrete variation of the parameter to be optimized under specific conditions, GA is particularly useful as a fast optimization tool in the exploration of the performance of potential coolants with acceptable patterns and trends when compared to the limited available data. Because of the strong proven capabilities of GA in reliability design, transportation, and medicine, this stochastic-based optimization tool has been utilized in this study. The circular and square geometry is being investigated here to compare the performance under similar conditions, microchannel height and microchannel wall, using the thermal resistance and pressure drop model, with experimentally obtained properties of ammonia. The former geometry is preferred in the air-conditioning and refrigeration industries while the latter, the rectangular type, is the common design in a MCHS in the cooling of ICs. In the present work, the thermal resistance in combination with the pressure drop across the channel is used as a measure of the performance of the MCHS. Liu and Garimella had determined that among all the analytical models that they investigated, the thermal resistance showed excellent agreement with their numerical results [19]. This procedure has never been done specifically with experimental data of ammonia.

2. Experimental model

The experimental apparatus is illustrated in Fig. 1a [10–12]. The facility is comprised of a refrigerant loop, the water loops, and a data acquisition system. As seen in the figure, the refrigerant loop consists of a receiver, a refrigerant pump, a mass flow meter, and a preheater, the test sections and a condensing unit. To start a testing circle, the refrigerant in the receiver tank is delivered to the test section by the refrigerant pump. The flow rate of the refrigerant can be controlled by adjusting the pumping frequency in the electric motor controller and is measured by a Coriolis-type mass flow meter. The mass quality at the inlet of the test section is controlled by a preheater. Vapor phase refrigerant at the outlet of the evaporative test section is condensed into liquid phase in the condensing unit. The liquid phase refrigerant is then accumulated in the receiver to start a new loop. Temperatures of the condenser, sub-cooler and preheater are adjusted by three individual water loops.

Fig. 1b shows details of the test section. The test section is made of circular stainless steel smooth tubes with inner diameter investigated of 1.5 and 3.0 mm with the heated length of 1000 and 200 mm in the horizontal orientation, respectively. For evaporation at the test section, power was conducted from an electric transformer to the test section. The input electric voltage and current were adjusted to control the input power. The outside tube wall temperatures at the top, middle and bottom sides were measured at every 100 mm axial intervals of the heated length using T type thermocouples. To measure the local saturation pressure, Bourdon tube-type pressure gauges were set up at the inlet and outlet of the test tube. A pressure transducer was also installed to evaluate the pressure gradient of the refrigerant flowing along the test section. Two sight glasses with the same inner diameter as the test section and a length of 200 mm were installed to visualize the flow and to enhance the flow stability of the fluid when entering the test section. To reduce the heat loss, test sections were well insulated with rubber and foam. The temperature, pressure and mass flow rate data were all recorded by using the data acquisition system.