International Communications in Heat and Mass Transfer 74 (2016) 40–47

Contents lists available at ScienceDirect

International Communications in Heat and Mass Transfer

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

Heat transfer enhancement in a micro-channel cooling system using cylindrical vortex generators☆

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ARTICLE INFO

Available online 17 March 2016

Keywords:
Vortex generators
Micro-channel
Hydraulic thermal performance
Micro scale cooling system
Heat transfer enhancement

ABSTRACT

Three-dimensional conjugate heat transfer under laminar flow conditions within a micro-channel is analysed numerically to explore the impact of a new design of vortex generator positioned at intervals along the base of the channel. The vortex generators are cylindrical with quarter-circle and half-circle cross sections, with variants spanning the whole width of the channel or parts of the channel. Micro-channels with Reynolds number ranging from 100 to 2300 are subjected to a uniform heat flux relevant to microelectronics cooling. To ensure the accuracy of the results, validations against previous microchannel studies were conducted and found to be in good agreement, before the new vortex generators with radii up to 400 μm were analysed. Using a thermal–hydraulic performance parameter expressed in a new way, the VGs described here are shown to offer significant potential in combatting the challenges of heat transfer in the technological drive towards lower weight/smaller volume electrical and electronic devices.

1. Introduction

Recently, developments in electronic and electrical devices have led to the reduction in their volumes and weights, and managing the heat generated is becoming a real challenge to thermal system researchers [1]. Therefore there is a need to improve cooling systems by decreasing their size and weight to micro- and mini-scale systems, such as micro-channel heat exchangers and heat sinks [2], whilst simultaneously increasing the efficiency to meet this development [3]. Mini- and micro-channels are different from traditional channels, and can be classified according to their associated hydraulic diameters, D_h, [4–6] as presented in Table 1.

The term 'micro-channel' first appeared in 1981 [7]. The concept had a great influence in thermal science as it decreased the hydraulic diameter and enhanced the heat transfer. In the revolution of advanced manufacturing processes, many experimental and numerical studies investigated the heat transfer and fluid flow performance of various modified geometries such as micro-channels with grooves and ribs [8–11]. The effect of vortex generators (VGs) on heat transfer and fluid flow characteristics were investigated experimentally in 1969 [12]. They can take various forms such as protrusions, wings, inclined blocks, winglets, fins, and ribs [13,14], and have been used to enhance heat transfer in different geometries such as circular and non-circular ducts under turbulent flow [15–17]. They have also been used in laminar flow [18], with flat plate-fins in rectangular channels [19–21], tube heat exchangers [22], heat sinks [18,23] and rectangular narrow channels [24,25].

The rectangular micro-channel was the best geometry based on the numerical investigation of Xia et al. [26], who considered various microchannel shapes. They also investigated the distribution of flow through a collection of 30 microchannels forming a heat sink, considering different header chamber shapes and inlet/outlet positions.

Ebrahimi et al. [13] used a finite volume based numerical analysis to study the impact of using vortex generators (VGs) with various orientations on laminar fluid flow and heat transfer regimes in a micro-channel. The results showed that the Nusselt number rose from 2 to 25% when the Reynolds number ranged from 100 to 1100, whilst the maximum increase of friction factor was 30% when using the VGs. Various other recent investigations have also indicated potential benefits of using VGs with laminar flow at different Reynolds number [24,27,28]. However, there remains a need for deeper understanding of VG performance over a wider range of laminar flow, for example to minimise the pressure drop penalty resulting from the introduction of VGs.

Cheraghi et al. [3] studied a 2D numerical smooth channel system with fixed heat flux applied to the wall sides and considered the effect of adding an adiabatic cylinder aligned perpendicular to the laminar flow direction and at different positions across the channel. The Reynolds number was 100 and the Prandtl number ranged from 0.1 to 1, and it was found that the maximum heat transfer enhancement

☆ Communicated by W.J. Minkowycz.

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http://dx.doi.org/10.1016/j.icheatmasstransfer.2016.03.002
0735-1933/

Nomenclatures

A_s	Surface area of the whole heat sink (μm^2)
CFD	Computational fluid dynamics
C_p	Specific heat, J/kg.K
D	Diameter, μm
FEM	Finite element method
FVM	Finite volume method
K	Thermal conductivity, W/m.K
L	Channel length, μm
VGs	Vortex generators
P	Pressure, N/m^2
q	Uniform heat flux, W/cm^2
Re	Reynolds number
T	Temperature, K
X	Axial distance, μm

Greek Symbols

μ	Viscosity, kg/ms
Θ	Thermal resistance
ρ	Densities, kg/m^3

Subscript

ave	Average
In	Inlet
max	Maximum
Out	Outlet
S	Surface
L	Liquid

occurred when the cylinder was fixed in the halfway position. The results also showed that the low Prandtl number had a positive effect on heat transfer enhancement.

A modified channel having cylindrical vortex generators inside a uniform channel under turbulent flow with Reynolds number of 3745 has also been investigated numerically [29]. It was found that utilizing a cylindrical vortex generator enhanced the heat transfer by 1.18 times compared to the uniform channel.

Based on the published literature and to the authors' knowledge, no previous study has investigated the influence of the diameter of half-cylindrical vortex generators placed along the base of the channel. This study therefore proposes and explores new geometry designs not previously considered: specifically VGs based on cylinders with half-circular and quarter-circular cross-sections are introduced into rectangular micro-channel heat sinks. Several lateral variations of these VGs are also considered, namely full-span cylinders, shorter centred cylinders, and split/separated cylinders. The optimal radii of the VGs are also established. Section 2 of this paper describes the geometry in more detail, after which Section 3 discusses the mathematical model and Section 4 discusses the numerical approach and validation process.

Table 1
Channel classification by Mehendale et al. [5] and Kandlikar and Grande [6].

Mehendale et al. [5].		Kandlikar and Grande [6].	
Conventional channels	$D_h > 6$ mm	Conventional channels	$D_h > 3$ mm
Compact passages	1 mm $< D_h \le 6$ mm	Minichannels	1 μm $< D_h \le 3$ mm
Meso-channels	100 mm $< D_h \le 1$ mm	Microchannels	10 μm $< D_h \le 200$ μm
Micro-channels	1 μm $< D_h \le 100$ μm	Transitional channels	0.1 μm $< D_h \le 10$ μm
		Molecular nanochannels	$D_h \le 0.1$ μm

The main results are discussed in Section 5, and conclusions are drawn in Section 6.

2. Geometry description

The base geometry considered is a single micro-channel with rectangular cross-section, as shown in Fig. 1(a). Such channels are common in the heat sinks designed for CPUs (Fig. 1b), where they form the gaps between the parallel fins of the heat sink, and that is the application considered here, with the base area of the heat sink taken as $A_s = 6.27 \times 10^8\ \mu m^2$. Within the channel, a number of cylindrical vortex generators are equally distributed along the base. These have cross-sections that are either a quarter-circle or a half-circle, as shown in Fig. 1(c) and (d) respectively, and a variable radius, *r*, ranging up to 400 μm. The micro-channel dimensions are given in Table 2. The table also shows the three different spanwise configurations considered, namely: 'full-span', where the VG occupies the full width of the micro-channel; 'centred', where the VG is shorter than the channel width and is centred; and 'split', which is the same as the centred configuration except that the VG is split into two equal parts, with a gap of 50–100 μm between them. Table 2 includes a view of the VGs looking along the channel from the inlet.

3. Mathematical modelling

3.1. Governing equations and key parameters

The water flow in the micro-channel is considered to be laminar, steady, incompressible and Newtonian, with gravitational and viscous dissipation effects neglected. With $\boldsymbol{u} = (u, v, w)$ representing the liquid velocity in (x, y, z) Cartesian coordinates, and p, ρ and μ denoting the liquid pressure, density and viscosity respectively, the (dimensional) governing equations for the flow are the usual continuity and Navier–Stokes equations:

$$\nabla \cdot \boldsymbol{u} = 0 \tag{1}$$

$$\rho(\boldsymbol{u} \cdot \nabla)\boldsymbol{u} = -\nabla p + \mu \nabla^2 \boldsymbol{u}. \tag{2}$$

The energy equation for the liquid phase in the micro-channel is

$$\rho C_p \boldsymbol{u} \cdot \nabla T_L = k \nabla^2 T_L \tag{3}$$

where C_p, T_L, and k are respectively the specific heat, temperature, and thermal conductivity of the liquid. Conduction in the solid is captured by

$$\nabla \cdot (k_S \nabla T_S) = 0 \tag{4}$$

where T_S and k_S are respectively the temperature and thermal conductivity of the solid.

The Reynolds number is here defined in terms of the inlet velocity, u_{in}, and hydraulic diameter as

$$Re = \frac{\rho u_{in} D_h}{\mu}. \tag{5}$$

The heat transfer performance is quantified by the thermal resistance, defined as

$$\Theta = \frac{T_{ave} - T_{in}}{A_s q}, \tag{6}$$

where T_{ave} is the average temperature of the base, T_{in} is the inlet temperature, and q is the heat flux through the base of the heat sink. To give a balanced assessment of the effective heat transfer enhancement provided by VGs, taking into account the penalty paid in terms of the pressure