Parameterization Investigation on the Microchannel Heat Sink with Slant Rectangular Ribs by numerical simulation

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Abstract: The slant rectangular ribs (SRRs) arranged in a microchannel can enhance heat transfer efficiency due to the flow disturbance, the interruption of boundary layer and increase of heat transfer area. In order to fully understand the mechanism of heat transfer enhancement, the flow and heat transfer in the microchannel with single- and double-row slant rectangular ribs are numerically simulated at Reynolds number ranging from 62.5-625. Numerical results show us that, great difference of flow pattern between microchannel heat sinks (MCHS) with single- and double-row slant rectangular ribs exists, and enormous influence of parameters of SRRs on heat transfer presents in various conditions. The flow disturbance of SRRs is the crucial factor to enhance heat transfer. Thereafter, the parameters such as attack angle, height, length, width and pitch of SRRs are evaluated by the average Nusselts number with identical pump power. Solid foundation for design of MCHS with SRRs can be provided.

Key words: Slant rectangular ribs (SRRs), Microchannel heat sink (MCHS), Parameterization investigation, Average Nusselts number, Numerical simulation.

1 Introduction

Microchannel heat sink has become most important cooling approach with the rapid increase in power density (even more than 100 W/cm^2) owing to miniaturization of electronic packages. The microchannel heat sink was proposed initiatively in 80's by Tuckeman[1] to take place of traditional heat exchanger for the well satisfaction of cooling requirement of integrated circuit (IC). One important way to enhance heat transfer capability of MCHS is to alter the shape of microchannel [2,3], the width-depth ratio of rectangular microchannel [4,5], or to employ complex manifold geometries [6,7], double/multi-layers microchannel [8-10], interrupted microchannel with various ribs in the transverse microchambers [11-13], and so on. Another way to improve the heat transfer efficiency of MCHS is to interrupt the thermal boundary by various lugs or/and ruts periodic located on the side wall or bottom wall [14-20].

Different baffles tilted at an angle to main flow were utilized usually in macroscopic channel (or tube) to enhance heat transfer. Arslan et al [21] experimentally investigated heat transfer and pressure drop in rectangular channel located baffles with various attack angle on the bottom at high Reynolds number ($Re=1000-10000$). The experimental results show us greater heat transfer enhancement and pressure drop with the increase of attack angle. Periodic located baffles (or ribs) with various shapes are usually applied in microchannel to enhance heat transfer. Enormous augmentation of heat transfer was found by Nasiruddin when MCHS with slant baffles located on the sidewalls, slant baffles located on the bottom wall may further enhance heat transfer due to the extra heat transfer area [22]. Numerical simulation of stacked microchannel heat sink with mixing-enhanced passive structure was proposed by Cheng [23]. The heat resistance drops with the increase of rib -height, but the pressure drop didn't discuss in that paper. Laminar periodic flow and

heat transfer in square channel with 45° inline baffles on two opposite walls were simulated numerically by Promvonge[24], the numerical result shows great enhancement of heat transfer in such a MCHS. As a result, to improve the mixing of hot fluids and cool fluids by changing the flow structure in microchannel is an effective approach to enhance heat transfer in MCHS. The structure of passive micromixer can be referred to the design of MCHS. An example presented by Fu et. al.[25] displays enormous enhancement of mixing due to the chaotic advection in the microchannel caused by staggered oriented ridges. In analogous, crosswise ridge leading to split and recombination helical flows carried out by Chen[26] is another example to intensify mixing as well. In addition, connected-groove in microchannels can improve fluids mixing in microdevices [27]. Ihsan Ali Ghani[28] reviewed many published result and concluded that, flow disruption for enhancing flow mixing, interruption of boundary layer and increasing heat transfer area are three important mechanisms to improve the performance of MCHS. In practice, periodic located lugs or/and ruts are frequently used to enhance heat transfer in MCHS. However, slant ribs are seldom employed in MCHS due to the complicated fabrication process. H.Salma [29] conducted optimization of thermal performances and pressure drop of microchannel heat sink using genetic algorithm on the basis of derivation of the total thermal resistance and pressure drop in a rectangular microchannel[30]. If there is no empirical formula for total thermal resistance and pressure drop, the computation burden by direct numerical simulation would be very huge.

On the basis of above review, MCHS with periodic located SRRs can interrupt boundary layer, increase the heat transfer area and enhance the flow mixing. The motivation of this work is to clarify the mechanism of heat transfer enhancement in MCHS with SRRs by numerical simulation. Furthermore, parameterized investigations are performed for high-performance of MCHS, and the geometric parameters, such as attack angle, width, height, length and pitch of SRRs, are optimized.

1 Geometric model and numerical model

1.1 Geometric model

Two MCHSs with single-row ribs and double-row ribs were designed for numerical investigation on the basis of analysis and reviews in last section. Single-row ribs channel (SRC) with rectangular slant ribs only on the bottom wall and double-row ribs channel (DRC) with rectangular slant ribs on both bottom and top walls are shown in Figure 1. Conjugate heat transfer of solid and fluid regions are under consideration. The axial heat conduction of solid region is considerable owing to the good thermal conductivity of solid material (silicon wafer in this work) [31].

The inlet dimension in both SRC and DRC is $H \times W = 300\mu\text{m} \times 300\mu\text{m}$ (cf. Fig 1). A constant heat flux $q = 100\text{W}/\text{cm}^2$ is loaded on the bottom wall, and $q = 0$ on other walls (i.e. thermal insulation). The thickness of bottom wall both in SRC and DRC are $80\mu\text{m}$. Note that, the thickness of top and side walls is neglected and don't display in Figure 1 and 2 because $q = 0$. The length of microchannel L_f is $2400\mu\text{m}$, while the channel length with constant heat flux L_s is $1800\mu\text{m}$. The geometric parameters of SRRs, including attack angle (the angle between main flow direction and slant ribs), height h , length l , width t and pitch d are shown in Figure 2. It is worth pointing out that, the rib height in DRC is only half of that in SRC.

Figure 1 Geometry of SRC and DRC

Figure 2 Geometric parameters of SRRs

1.2 Numerical model

In allusion to the MCHS composing of silicon channel and filled water as working fluid, following assumptions can be made, 1) incompressible, 2) no consideration of source term, 3) no viscous dissipation, 4) steady flow. Hence, the control equations for the fluid in MCHS read as:

1) the continuity equation:

$$\nabla \cdot \vec{V} = 0 \quad (1)$$

where, $\vec{V}$ denotes flow velocity.

2) the momentum equations:

$$\rho \left[\frac{\partial \vec{V}}{\partial t} + (\vec{V} \cdot \nabla) \vec{V} \right] = -\nabla p + \mu \nabla^2 \vec{V} \quad (2)$$

where, ρ, p, μ are density of fluid, pressure and dynamic viscosity, respectively.

3) the energy equation:

$$\rho_f c_f \left[\frac{\partial T}{\partial t} + \vec{V} \cdot \nabla T \right] = \nabla \cdot k_f \nabla T \quad (3)$$

Where, k_f, c_f, T is thermal conductivity, specific heat and temperature, respectively. Note that, the unsteady terms in equation (2) and (3) equal to zero in present work.

The control equation for solid silicon reads as:

$$k_s \nabla^2 T = 0 \quad (4)$$

The physical property of silicon and water are listed in Table 1. The density, conductivity and specific heat of silicon and water are set to be constants at any temperature because they change less than 10% when temperature increasing from 293K to 333K. Instead, the viscosity of water will decrease more than 50% when temperature increasing from 293K to 333K[32]. Hence, the value of fluid viscosity obtained by linear interpolation in each 10 Kelvin interval should be updated at every iterative computation.

Table 1 Thermal physical property [32]

	Density (kg/m ³)	Conductivity (W/m·K)	Specific heat (J/kg·K)	Viscosity (10 ⁻³ Pa·s)			
				293K	303K	313K	323K
Water	988.2	0.6	4182	1.0040	0.8015	0.6533	0.549
Silicon	2328.3	148	700				

Enforced no-slip boundary conditions at all solid-fluid interfaces mean no velocity-slip and temperature-jump. Velocity of fluid v_{in} is uniform at inlet, outflow is set at outlet. The initial temperature for fluid and solid both are 293K. Constant heat flux is assumed, so Riemann

boundary condition like $-k_s \frac{\partial T}{\partial y} = q$ should be exerted on wall, where $q=100\text{W/cm}^2$ on the

bottom wall, and $q=0$ on other walls including side walls. Actually said, the heat flux through the side walls aren't exactly zero in most case. Some numerical simulation trials for the entirety of MCHS were carried out to explore how many heats can be transported through side walls. Fortunately, the heat through side walls is very limited (less than 0.02% of that from bottom wall) and can be neglected.

The hybrid meshes in whole simulation region should be generated by T-grid because of the complex geometrical structure. In the meantime, the verification of grid independence was conducted for the grid number ranging from 200k to 2000k. It can be concluded that, 800-1000k grids are suitable to present work.

Average Nusselt number $\bar{Nu}$ to denote heat transfer capability can be written as [33]:

$$\bar{Nu} = \frac{\bar{h}D_h}{k_f} \quad (5)$$

where, $D_h = \frac{4A}{P}$ is hydrodynamic radius, $A = W \times H$ is area of inlet, P is wetted perimeter.

Average heat transfer coefficient $\bar{h}$ can be defined as:

$$\bar{h} = \frac{qA_q}{A_c(T_c - T_f)} \quad (6)$$

where, A_q is heated area, A_c is conjugated area (i.e. solid-fluid interface area on bottom wall)

$T_c = \frac{\int TdA}{\int dA}$ is the average temperature in conjugated area, $T_f = \frac{\int T\rho_f dV}{\int \rho_f dV}$ is average temperature of

fluid in channel.

Local Nusselts number at a fixed x can be read as:

$$Nu_x = \frac{qA_q D_h}{k_f A_c (T_{wx} - T_{fx})} \quad (7)$$

where, $T_{wx} = \frac{\int T dz}{\int dz}$ is average temperature at a fixed x on heated solid-fluid interface,

$T_{fx} = \frac{\int T \rho_f (V \cdot dA)}{\int \rho_f (V \cdot dA)}$ is average temperature of fluid at a fixed x.

Friction coefficient denoting resistance force reads as:

$$f = \frac{2\Delta p D_h}{L \rho_f v_{in}^2} \quad (8)$$

where, Δp is pressure drop at heating length L_s , v_{in} is inlet velocity.

Pump power of MCHS for steady flow can be defined as

$$P_p = \Delta p v_{in} A \quad (9)$$

1.3 Validity of numerical model

Roache[34] suggested that, the numerical model should be verified normally before applied in practical engineering. For this purpose, comparison between numerical results of a simplified model without ribs and experimental results is carried out. The simplified model is a rectangle channel with the same dimensions as that in Ref.[19]. More than 1.2 million rectangular grids are meshed for the computational domain of MCHS. No-slip boundary conditions, velocity-inlet, outflow are enforced at wall, at inlet, at outlet, respectively. Riemann boundary conditions

$$-k_s \frac{\partial T}{\partial y} = 0 \text{ at inlet and at walls except bottom wall are exerted, } -k_s \frac{\partial T}{\partial y} = q \text{ at bottom}$$

wall. The initial temperature for fluid and solid both are 293K. Finite volume method, second order upwind discretization and SIMPLEX algorithm are used to solve the equations. Figure 3 shows us that, the numerical results agree good with the experimental results (Figure 5 in Ref [19] for friction coefficient, Figure 10 in Ref [19] for Nusselts number). The validity of numerical model is verified.

In addition, some literature indicated that, viscous dissipation may not be neglected. Results in Table 2 show us that, viscous dissipation can be neglected because of the little influence on the numerical results.

Table 2 The influence of viscous dissipation on numerical results

	with viscous dissipation	no viscous dissipation
$\overline{Nu}$	48.508	48.0766
f	1.04424	1.03862

Figure 3 Validity of numerical model

	with viscous dissipation	no viscous dissipation
$\overline{Nu}$	48.508	48.0766
f	1.04424	1.03862

2

Results

and discussion

Chaotic advection in microchannel and unceasing interruption of thermal boundary are two effective approaches to improve the performance of MCHS. Periodic arranged transverse ribs or other similar structures in a microchannel can certainly interrupt boundary layer. Furthermore, the transverse ribs will influence flow pattern and increase heat transfer area. In order to elaborate the influence of flow patterns in microchannel on the performance of MCHS, numerical simulations of flow and heat transfer in MCHS with SRRs (SRC and DRC) were conducted for Reynolds ranging from 62.5 to 625. It is known to us, the Reynolds number for MCHS normally ranges from 10-1000, much lower Reynolds number is unacceptable because $q=100\text{W}/\text{cm}^2$. On the contrary, much higher Reynolds number can result in the disorder of flow pattern. It is sure after some trials that, stable flow pattern determinately exists only when Reynolds number less than ~ 700 , because the motivation of this work is to clarify the mechanism of heat transfer enhancement in MCHS with SRRs

2.1 Flow pattern

To understand the influence mechanism of transverse flow(secondary flow) on heat transfer in microchannel, the overall stream-lines in fluid region and flow patterns in the cross-section normal to main flow are analyzed in this section.

2.1.1 Overall 3-D flow pattern

Numerical simulations of MCHS with SRC and DRC were conducted when the attack angle is set to be 30,45,60 degree (the other parameters: $Re=625$, $h/H=0.5$, $l/W=0.8$, $d/W=1$, $t=30\mu m$). Figure 4a-f show us stream-lines in whole region. It can be seen that, nearly all of the stream-lines in DRC are helical. Helical stream-lines indicate that, the fluids on the bottom and the fluids on the top can exchange each other to enhance heat transfer efficiency. However, some relatively flat stream-lines on the top of SRC result in recession of heat transfer enhancement. Moreover, the helix angle decreases with the increase of attack angle, that means the fluids on the bottom and the fluids on the top will exchange more frequently, hence more stretching and folding of fluids lead to more heat transfer enhancement.

Figure 4 Stream lines in the whole region ($Re=625$, $h/H=0.5$, $l/W=0.8$, $d/W=1$, $t=30\mu m$)

(a)SRC $\theta=30^\circ$, (b)DRC $\theta=30^\circ$, (c)SRC $\theta=45^\circ$, (d)DRC $\theta=45^\circ$, (e)SRC $\theta=60^\circ$, (f)DRC $\theta=60^\circ$

2.1.2 Flow patterns in cross sections

1) Flow patterns in SRC

Figure 5 are flow patterns in cross section at various x when $Re=625$, $\theta=45^\circ$, $h/H=0.5$, $l/W=0.8$, $d/W=1$ and $t=30\mu m$. It is shown that, the contrarotations of majority of fluids lead to heat transfer enhancement because of the exchange between upgoing fluids and downgoing fluids.

Temperature distributions and velocity vectors diagrams in the region of certain SRR can help us to understand the effect of SRRs on flow pattern. It can be seen from Figure 6 that, the

vortex is right at the position of hot fluids region both in the front (Figure 6a) and rear (Figure 6h) of SRR. Somewhat different is more homogeneous temperature distributions and more widely spread of hot fluids regions in the rear of SRR than that in front of SRR. In addition, Figure 6b-6d show us one part of hot fluids come from bottom to center of the channel following the vortex. And Figure 6e-6g show us the considerable increase of heat dissipation areas due to the arranged SRRs in microchannel, besides the shifts of flow patterns result in mixing acceleration of hot and cool fluids.

Another point worthy noting is how the hot fluids move to the center of channel from the vicinity of SRR? Figure 6b show us the fluids will be split right and left because of the coming rib. The fluids near the right of rib will flow into the vortex (Figure 6c), and the fluids near the left of rib will flow into another small vortex newly generating in the region between wall and left side of rib(Figure 6d). Hereafter, the vortex will cross over the top of rib by compulsion because the rib occupies the original center position of vortex, and swallow the newly generating vortex successively (Figure 6e and 6f). A more widely spread vortex is formed and can take the heats away from the regions near the rib and the bottom wall to the channel center(Figure 6g).

Figure 5 Flow patterns in cross section at various x in SRC($\text{Re}=625$, 45° , $h/H=0.5$, $l/W=0.8$, $d/W=1$, $t=30\mu\text{m}$)

Figure 6 Temperature and velocity distributions in the region of SRR at various x in SRC ($\text{Re}=625$, 45° , $h/H=0.5$, $l/W=0.8$, $d/W=1$, $t=30\mu\text{m}$)

2) Flow patterns in DRC

Figure 7 are flow patterns in cross section at various x when $Re=625$, $\theta=45^\circ$, $h/H=0.5$, $l/W=0.8$, $d/W=1$ and $t=30\mu m$. It is shown that, the contrarotations of majority of fluids like that in SRC lead to heat transfer enhancement owing to the exchange between upgoing fluids and downgoing fluids. There exists a vortex-pair in DRC, but only one vortex in SRC(cf. Figure 5). The center-pair of flow patterns shifts violently during the fluids flow every rib-pair (rib on bottom wall and rib on top wall). The shifts of the vortex-pair result in blinking flow which exists deterministic chaos[35]. It can be deduced from above analysis, more enormous enhancement of heat transfer in DRC can be obtained than that in SRC.

Figure 7 Flow patterns in cross section at various x in DRC($Re=625$, $45^\circ h/H=0.5$, $l/W=0.8$, $d/W=1$, $t=30\mu m$)

Temperature distributions and velocity vectors diagrams in the region of SRR can help us to understand the effect of rib-pair on flow pattern. It can be seen from Figure 8 that, the vortex-pair is right at the position of hot fluids both in the front(Figure 8a) and rear (Figure 8f) of rib-pair. Somewhat different is more homogeneous temperature distributions and more widely spread of hot fluids in the rear of rib-pair than that in front of rib-pair. In addition, Figure 8b-8c show us one part of hot fluids come from bottom to center of the channel following the vortex. And Figure 8e show us somewhat increase of heat dissipation areas due to located rib-pairs whose height being half of that in SRC, besides the shift of flow patterns can result in the mixing acceleration of hot and cool fluids.

(e)x=1100 μ m

(f)x=1200 μ m

Figure 8 Temperature and velocity distributions in the region of SRRs at various x in DRC

(Re=625, $\theta=45^\circ$, h/H=0.5, l/W=0.8, d/W=1, t=30 μ m)

How the fluids move to the center of channel from the vicinity of bottom-rib(the rib of rib-pair on the bottom wall)? Figure 8b show us the fluids will be split right and left because of the coming rib-pair. The fluids near the right of bottom-rib will flow into the right vortex(Figure 6c) , and the fluids near the left of bottom-rib will flow into another small vortex-pair newly generating in the region between wall and left side of bottom-rib(Figure 8d-8e). The same status can be seen in the top half of microchannel. Hereafter, the contrarotated original vortex-pair will swallow the newly generating vortex-pair because the original vortex-pair rotate slowly than the new one. A more widely spread vortex-pair are formed and can take the heats away from the regions near the bottom-rib and the bottom wall to the channel center (Figure 6f).

2.1.3 Temperature and Nusselts number

1) Temperature contours

The temperature contours at various cross-sections(y-z plane) normal to the main flow can reveal the temperature variation along main flow of SRC and DRC. Comparison of the temperature contours of SRC and DRC in Figure 9 shows us that, the fluids near the bottom wall and bottom-rib in SRC and DRC will be pushed to the channel-center by vortex (or vortex-pair)owing to the SRRs. Nevertheless, more heats will be carried from the bottom region in DRC than that in SRC, and more widely spread and more homogeneous distribution of temperature can be seen in DRC than that in SRC. It can be concluded better heat transfer performance in DRC than that in SRC, this is coincident with the conclusion drawn from the analysis of flow pattern in last section.

Something interesting is distinct discrepancy of the temperature distributions in the plane(x-y plane) near the bottom wall in SRC and DRC. It can be seen from Figure 9, high-temperature regions in the rear of the first three ribs are larger in SRC than that in DRC. The likely reasons are: 1) the velocity of the fluids in the rear of rib is lower in SRC than that in DRC, and fluids have enough time to absorb the heats from the bottom wall. 2) the fluids with lower temperature from the top-half of the microchannel can be easily driven to the bottom of the microchannel by top-ribs in DRC. Reversely, high-temperature regions in the rear of next three ribs are smaller in SRC than that in DRC, because the fluids with higher temperature moved to the top of the channel early will be driven to the bottom again by the top-ribs in DRC.

Figure 9 Temperature contours in SRC and DRC
($Re=375$, 45° , $h/H=0.5$, $l/W=0.8$, $d/W=1$, $t=30\mu m$)

2) Nusselts number

It is known to all, the variation of Nusselts number along the stream can be utilized to voice heat transfer performance in MCHS. The Nusselts numbers periodically fluctuate always both in SRC and DRC(see in Figure 10) due to the periodic located ribs(or rib-pairs) unlike that in straight microchannel (SC). The reason is that, the heat will be taken away from bottom to top by the bottom ribs, while the fluids from the mid and top of channel can be moved to the bottom to enhance the heat transfer in the front of the bottom ribs. Hence, an increasing Nusselts number can be resulted in. Thereafter, the Nusselts number will decrease gradually to a lowest point when the cool fluids absorb more and more heats. The fluctuation of Nusselts number will be repeated every time during flow in the region of rib (or rib-pair).

Note that, the tendencies of Nusselts number variation in DRC is unlike that in SRC. The Nusselts number in SRC always increases oscillatorily when $x=600-2000\mu m$, because the hot fluids are driven from the bottom, and cool fluids from the top replenish the bottom region unceasingly. However, The Nusselts number increases oscillatorily when $x=600-1200\mu m$ (i.e. the first three rib-pairs), and then flattens or decrease gradually when x exceeds $1200\mu m$ because the fluids with higher temperature driven from the bottom previously are pushed back to the bottom by the top-ribs(cf. Figure 4). It could predict, the Nusselts number both in DRC and in SRC will oscillatorily decrease in a longer channel(or experiencing more ribs/rib-pairs) like that in a straight channel because more and more heats are absorbed by the fluids. In a words, the reasons of heat transfer enhancement in SRC and DRC are: 1) periodic interruption of the boundary layer, 2) chaotic advections in microchannel, 3)more cooling areas attributed to the bottom ribs.

Figure 10 Nusselts number at various x

($Re=375$, 45° , $h/H=0.5$, $l/W=0.8$, $d/W=1$, $t=30\mu m$)

3.2 Optimization of geometric parameters of SRR

It is known from the above analysis, the geometric parameters of the SRRs and Reynolds number influence significantly on the flow pattern and heat transfer. Active roles of the SRRs are summarized in the end of section 3.1, passive role of SRRs is much greater pressure drop. Therefore, pump power may be a suitable index to measure heat transfer enhancement, in other words, average Nusselts number under the identical pump power is a good option to evaluate thermal performance of MCHS.

3.2.1 Attack angle

Enormous influence of attack angle of SRRs will be exerted on the heat transfer and pressure drop because of the transverse flow caused by SRRs. Numerical simulations for various attack angles ($30^\circ, 37.5^\circ, 45^\circ, 52.5^\circ, 60^\circ$) were carried out to calculate average Nusselts number (Nu_0) and friction coefficient (f_0) at fixed parameters $h/H=0.5$, $l/W=0.8$, $d/W=1$, $t=30\mu m$. Figure 11a shows us that, more effective heat transfer exists in SRC than in DRC at lower Reynolds number (less than 125) when the attack angle being relative small. Only half of the height of the bottom-rib in DRC comparing to that in SRC results in less cooling area, smaller vortex near rib, less fluids mixing. Nevertheless, a stable vortex-pair structure can be generated in DRC even in lower Reynolds number when the attack angle increases. Frequent shift of the vortex-pair structure during the flow can induce blinking flow to promote heat transfer efficiency.

(a) Re vs Nu/Nu_0 , (b) Re vs f/f_0 , (c) Pump power vs Nu , (d) Partial zoom of (c)

Figure 11 Influence of attacks on the heat transfer ($h/H=0.5$, $l/W=0.8$, $d/W=1$, $t=30\mu m$)

It is easily to understand, the pressure drop will increase with the increase of attack angle (see in Figure 11b). What needs to be explained is that, the friction coefficient of SRC is bigger than that of DRC. The reason is that, one half of the high-speed flow is warded off when the rib height is $H/2$, while only two $H/4$ of low-speed flow near the wall are warded off in DRC. After the comprehensive consideration of heat transfer and pressure drop, pump power is prompted to measure the performance of MCHS. Figure 11c shows us that, the Nusselts number increases logarithmically with the pump power. Maximum Nusselts number for SRC at $\theta=52.5^\circ$ means the optimization value of attack angle. Nusselts numbers are very close when attack angle bigger than

45°, but $\theta=45^\circ$ can be intended to be an optimization value for DRC from Figure 11d.

3.2.2 Height

Intuitional comprehend tells us that, higher rib can result in more cooling areas, more fluids in vortex/vortex-pair, more heat exchange, hence more effective heat transfer like that indicated in Figure 12a. However, much higher rib enlargers flow resistance, suppresses the formation of vortices, and destroys the vortex structure. As a result, optimization value of rib height exists. Figure 12c shows us, the highest Nusselts numbers exist both in SRC and DRC when $h/H=0.3$. Moreover, the Nusselts number in SRC and DRC is much bigger than that in straight channel ($h/H=0$).

Figure12 Effect of rib-height on heat transfer

Figure 13 Effect of rib pitch on heat transfer (6 ribs/rib-pairs,

SRC: $\theta=52.5^\circ$, $h/H=0.3$, $l/W=0.9$, $t=30\mu m$,

DRC: $\theta=45^\circ$, $h/H=0.5$, $l/W=0.9$, $t=30\mu m$),

(a) Re vs Nu/Nu_0 , (b) Re vs f/f_0 , (c) Pump power vs Nu

(DRC: $\theta=45^\circ$, SRC: $\theta=52.5^\circ$, $l/W=1$, $d/W=1$, $t=30\mu\text{m}$)

(a) Re vs Nu/Nu0 (b) Re vs f/f_0 (c) Pump power vs Nu

3.2.3 Rib pitch

It is known from the stream lines in Figure 4, the helical stream lines will be influenced by three important factors, attack angle, flow velocity and rib-pitch. A larger pitch needs to be allocated for smaller attack angle and larger flow velocity, and vice versa. In allusion to the optimization attack angle, Nusselts number and friction coefficient at various Reynolds numbers were calculated (Figure 13 a,b). Figure 13c shows us the optimization value of rib pitch both in SRC and DRC is $d/w=6:6$.

3.2.4 Length and width

The length and width will affect transverse flow, cooling areas, pitch, and so on. Figure 14 shows us that, the optimization value of rib length are $l/W=0.8$ and Max (penetrating cross the channel) for DRC and SRC respectively, and the optimization value of rib width are $t/W=4/30$ and $t/W=3/30$ for DRC and SRC respectively.

3.3 Optimization results.

In conclusion, the optimization parameters for SRC are: $\theta=52.5^\circ$, $h/H=0.3$, $d/W=1$, $t/W=3/30$, penetrating length, respectively. For DRC, $\theta=45^\circ$, $h/H=0.3$, $d/W=1$, $t/W=4/30$, $l/W=0.8$,

Figure 14 Effect of rib length on heat transfer

Figure 15 Effect of rib width on heat transfer

(6 ribs/rib-pairs, $\theta=45^\circ$, $h/H=0.3$, $d/W=1$, $t=30\mu\text{m}$)

(6 ribs/rib-pairs, $\theta=45^\circ$, $h/H=0.3$, $l/W=1$, $d/W=1$)

4 Conclusions

Numerical simulations for the flow, pressure drop, temperature were carried out to analyze the flow pattern and temperature distribution in SRC and DRC. The mechanism to influence the performance of MCHS by the SRRs is clarified. And geometric parameters of SRRs were optimized. The following conclusions can be drawn:

- 1) Three important factors for enhancement of heat transfer in MCHS are interruption of

thermal boundary, increase of heat transfer area, generation of chaotic advection by arranged SRRs.

2) Stable single-vortex pattern which can prompt to transport the fluids near the bottom and bottom-ribs upwards to the channel center exists in SRC, and every small vortex generating near the bottom-ribs by the SRRs will be combined by relative larger vortex from upstream. Stable vortex-pairs pattern which prompt to transport the fluids near the bottom and bottom-ribs upwards to the top half of channel exists in DRC except at lower Reynolds and smaller attack angle. Similarly, the vortex-pair from upstream will swallow two new-generating smaller vortices near the bottom ribs, and forms a pair of new vortex-pair. Blinking flow induced in this process by the shifts of the center of flow pattern can result in chaotic advection. Both SRC and DRC can enhance the heat transfer by the increase area of the hot and cool fluids due to the stretching and folding.

3) The performance of MCHS is better in DRC than that in SRC because a stable vortex-pair can generate in DRC except when the smaller attack angle and lower Reynolds number being assumed.

4) Comprehensive consideration of heat transfer and pressure drop can be taken to measure the performance of MCHS by the Nusselts number at identical pump power. The optimization parameters for SRC are: $\theta=52.5^\circ$, $h/H=0.3$, $d/W=1$, $t/W=3/30$, penetrating length, respectively, while $\theta=45^\circ$, $h/H=0.3$, $d/W=1$, $t/W=4/30$, $l/W=0.8$, respectively for DRC.

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Nomenclatures

A / cm^2	Area	$\bar{Nu}$	Average Nusselts number
A_q / cm^2	Heated area	Nu_x	Local Nusselts number
A_c / cm^2	Conjugated area	$q / \text{W} \cdot \text{cm}^2$	Heat density
A_{in} / cm^2	Area of inlet	θ / degree	Attack angle of rib
$c_f / \text{J}(\text{kg} \cdot \text{K})^{-1}$	Specific heat	P / μ	Wetted perimeter
$d / \mu\text{m}$	Pitch of rib	p / kPa	Pressure
D_h	1. Hydrodynamic radius	$\rho / \text{kg} \cdot \text{m}^{-3}$	Density
$\Delta p / \text{Pa}$	Pressure drop	Re	Reynolds number
f	Friction coefficient	$t / \mu\text{m}$	Thickness of rib
$k_f / \text{W}(\text{m} \cdot \text{K})^{-1}$	Thermal conductivity of fluid	T / K	Temperature
$K_s / \text{W}(\text{m} \cdot \text{K})^{-1}$	Thermal conductivity of solid	T_c / K	Conjugated temperature
$h / \mu\text{m}$	Height of rib	T_f / K	Fluid temperature
$H / \mu\text{m}$	Height of channel	$\mu / 10^{-3} \text{Pa} \cdot \text{S}$	Viscosity
$\bar{h} / \text{W} \cdot \text{K}^{-1} \text{cm}^{-2}$	Average heat transfer coefficient	$V / \text{m} \cdot \text{s}^{-1}$	Velocity of fluid
$L / \mu\text{m}$	Length of rib	$v_{in} / \text{m} \cdot \text{s}^{-1}$	Inlet velocity
$L_f / \mu\text{m}$	Length of fluid-filled channel	$W / \mu\text{m}$	Width of channel
$L_s / \mu\text{m}$	Length of heated channel	$x, y, z / \mu\text{m}$	Coordinate

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Highlights

The slant rectangular ribs(SRRs) are arranged in a microchannel to enhance heat transfer.

MCHS in single- and double-row SRRs are numerically simulated at $Re=62.5-625$.

The most important factor for heat transfer enhancement is chaotic advection caused by SRRs.

The parameters of SRRs are optimized for heat transfer enhancement.