NUMERICAL STUDY OF LONGITUDINAL VORTEX AND CHAOTIC FLOW ON HEAT TRANSFER CHARACTERISTIC IN MICRO-CHANNELS

by

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In this study, the flow characteristics and heat transfer mechanism of the chaotic flow induced by a micro-channel containing staggered herringbone micro-structures (CM), longitudinal vortex induced by a micro-channel containing inclined ribs (SLM), and a smooth micro-channel (SM) were comprehensively analyzed through numerical simulations at Reynolds number of 164 to 965. The results demonstrated that the primary enhanced heat transfer mechanism of the single longitudinal vortex was the disruption of the thermal boundary-layer, whereas the chaotic flow enhanced heat transfer by facilitating fluid mixing. Furthermore, the longitudinal vortex decay is slower than that of the chaotic flow, resulting in SLM having a superior heat transfer performance at lower pressure drops compared with the CM. Further results showed that the SLM had the highest Nusselt number (Nu = 28), the best comprehensive evaluation factor (PEC = 2.1), and the lowest thermal resistance ($R_T = 0.6 \text{ K/W}$).

Key words: micro-channels, single longitudinal vortex, chaotic flow, heat transfer, flow analysis

Introduction

The micro-channel heat sink was first proposed by Tuckerman and Pease [1]. It has been widely used in microelectronic mechanical systems, chemical industry, energy, aerospace, and other fields owing to its high heat transfer efficiency and compact structure [2-4]. With the development of high performance, integrated, and miniaturized electronic devices, the heat flux in traditional integrated circuits has already exceeded 100 W/cm², and reached 10³ W/cm² or even more in some ultra-large-scale integrated circuits. If the heat flux cannot be effectively removed, a rapid increase in temperature will reduce the reliability of electronic devices. Therefore, the pursuit of highly efficient heat transfer heat sinks has emerged as a primary research focus for numerous scholars.

In recent years, scholars have conducted numerous studies on micro-channels to further improve their heat transfer performance. In general, technologies to improve the heat transfer performance of micro-channels can be divided into active heat transfer enhancement technology and passive heat transfer enhancement technology. Active heat transfer enhancement technology requires external energy, such as external magnetic fields [5], electric fields [6], and surface vibrations [7], to achieve heat transfer enhancement. Passive heat transfer enhance-

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ment technology improves the heat transfer performance by changing the structure of the micro-channels, using porous materials, or changing the coolant physical properties. Passive heat transfer enhancement technologies are widely used to enhance the heat transfer performance of micro-channels, owing to their high reliability, economics, and applicability. With the benefit of this technology, many new micro-channels have been developed in recent years, including divergent-convergent [8], wavy [9], and cavity-containing micro-channels [10]. However, these micro-channels have several shortcomings. For instance, the stagnation region may appear at low Reynolds numbers, which will have an unfavorable effect on heat transfer enhancement. To address these shortcomings, researchers have proposed the design of micro-channels capable of inducing longitudinal vortices, chaotic flows, or other flow patterns to enhance heat transfer. For instance, Promvonge *et al.* [11] found that the presence of longitudinal vortexes induced by inclined ribs resulted in a maximum PEC of approximately 2.2. Apart from the increase in the Reynolds number, the significant increase in heat transfer was due to impingement jets induced by a longitudinal vortex pair of flow appearing on the upper, lower, and rib trailing end side walls. Feng et al. [12] demonstrated that the generation of longitudinal vortexes induced by wire coils enhanced the heat transfer performance in mini-channels effectively. Yuan et al. [13] observed that interconnected grooves effectively changed the flow transitioning from transverse vortices to longitudinal vortices, leading to higher heat transfer performance.

In the study of chaotic flows, researchers initially applied this flow to solve the problem of fluid mixing. However, it was discovered that chaotic flow also played a significant role in enhancing heat transfer. For instance, Greiner et al. [14] found that inducing chaotic advection in a system was one potential passive method for enhancing heat transfer. Stroock et al. [15] first comprehensively elucidated the enhancing mixing mechanism of the chaotic flow generated by staggered herringbone micro-structures in their pioneering work. In addition, they indicated that chaotic flow also enhanced the rates of heat transfer because the mass transfer mechanism was closely related to that of the heat transfer. However, they did not analyze the pressure drop of the chaotic flow generated by staggered herringbone micro-structures. Besides. researchers [16, 17] also observed that chaotic flow occurred in various other configurations, such as wavy-walled micro-channels and twisted patterns, and they concluded that they resulted in enhanced rates of mixing and heat transfer. Ghaedamini et al. [16] found that wavy walled micro-channels were capable of inducing chaotic advection and strong chaotic advection was observed at high pressure drop. However, they found that strong chaotic advection may be achieved with smaller pressure drops in certain situations. Castelain et al. [17] observed completely regular chaotic flow (from the Eulerian point of view) experimentally in a twisted duct flow through a laser Doppler velocimeter. From the literature review, it is clear that both longitudinal vortices and chaotic flow can effectively improve thermal performance and comprehensive performance. However, these two types of flow are based on different mechanisms and exhibit different flow forms. As the fluid-flow has a significant impact on the heat transfer performance, it is essential to investigate the influence of these two flows on the heat transfer and their fundamental differences.

In this study, staggered herringbones and inclined rib micro-structures were selected to induce chaotic flow and longitudinal vortices. The thermal-hydraulic characteristics of the micro-channels containing these two micro-structures were numerically investigated. Thus, the effects of the longitudinal vortex and chaotic flow on the local thermal hydraulic characteristics (including distributions of local pressure drop, temperature fields, and dimensionless secondary flow intensity) and average thermal-hydraulic characteristics (including friction resistance coefficient, Nusselt number, and comprehensive performance under different Reynolds numbers) were analyzed thoroughly. This study will serve as a valuable guide for micro-channel design to enhance heat transfer, particularly in the context of heat sink and transport phenomena.

Numerical simulation

Physical model

Figure 1(a) shows the schematic diagram of the CM, with overall dimensions of 37 mm in length, L, 2.5 mm in width, W, and 1.25 mm in height, H. The fluid inlet section area of the CM was 1.5 mm in width, W_c , and 1.25 mm in height, H_c . The CM comprised three sets of micro-structures, with each set containing six staggered herringbones, and the spacing, L_{S1} , between each set was 2 mm. The sizes of the staggered herringbone micro-structures are 0.5 mm for the width, W_r , and 0.25 mm for the height, H_r . The distance, L_{S2} , between the two adjacent micro-structures was 1 mm. The distance, L_{S2} , between two adjacent micro-structures was 1 mm. The distance, L_{S2} , between two adjacent micro-structures was 1 mm. The distance include the micro-structures was consistently 45°, as indicated by the green arrow. Figure 1(b) shows a inclined ribs micro-channel (SLM). Except the pattern of the SLM differs from that of the CM, the overall dimensions, fluid inlet section area, arrangement of the micro-structures, and fluid-flow direction remain consistent with those of the CM. Smooth rectangular micro-channels were used for comparison with SLM and CM.



Figure 1. Schematic diagrams of; (a) CM and (b) SLM micro-channel

Numerical model

The numerical model can be assumed to be a 3-D steady-state incompressible laminar flow process, and the effects of volume force, gravity, thermal radiation and viscous dissipation can be ignored [12]. Under these assumptions, the governing equations were cited from [12]. Copper and deionized water were chosen as the materials for the micro-channel and coolant, respectively. The viscosity of deionized water was assumed to change in accordance with temperature fluctuations, and the specific expression was cited from [18].

The inlet flow rate of the coolant u_{in} was 0.12~0.84 m/s, and the temperature, T_{in} , was fixed at 298 K. A constant heat flux of $q_w = 10^5$ W/m² was applied on the bottom of the micro-channel. The remaining walls were assigned adiabatic walls and without velocity slip. Fluent CFD software was used to solve the model, with the convergence residual set as 10^{-5} .

Grid independence

Figures 2(a) and 2(b) show the grid structures of the SLM and CM. The SLM was used to determine the final grid number. As the deviations of pressure drops ΔP in the SLM corresponding to 0.8 and 1.8 million grids from that corresponding to 1.5 million grids were 7.2% and 2.3%, and the deviations of Nusselt number of the SLM micro-channel corresponding to 0.8 and 1.8 million grids from that corresponding to 1.5 million grids were 5.3% and 1.5%, respectively. The 1.5 million grid number was selected to divide all the micro-channel geometric models. The discrepancy between the experimental and numerical mean Nusselt number and ΔP of the SLM was validated in section *Vallidation with experimental data*.



Figure 2. Schematic diagrams of 3-D grid structure of; (a) SLM and (b) CM

Data reduction

The formulas of Reynolds number, the hydraulic diameter of the micro-channel, D_h , values of average Nusselt number, the average Darcy friction resistance coefficient, f, the pressure drop, ΔP , between the inlet and outlet, the PEC and the thermal resistance (RT), are cited from [9].

To evaluate the intensity of the vortex in the cross-section of the fluid, we defined the dimensionless secondary flow intensity, *Se*:

$$Se = \frac{\rho_f D_h^2 \frac{1}{A(x)} \int \int_{A(x)} |\omega^n| dA}{\mu_f}$$
(1)

where ω^n and A(x) are the vorticity and the cross-section of the fluid in the mainstream direction, respectively. The ΔSe is the difference between the Se and the Se_{plain}, where Se and Se_{plain} stands for the dimensionless secondary flow intensity in micro-channels with and without micro-structures, respectively:

$$\Delta Se = Se - Se_{\text{plain}} \tag{2}$$

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Results and discussion

Validation with experimental data

The discrepancy between the experimental and numerical mean Nusselt number and ΔP values of the SLM was validated. The experimental set-up was described in [17]. The experimental conditions (cooling inlet T_{in} , flow rate u_{in} , and heat flux, q_w , were consistent with the simulated boundary conditions. The differences between the simulated results and tested data were within 5% and 6%, respectively, indicating that the numerical solution here was reliable.

Heat transfer characteristics

The variations in Nusselt number and PEC for different types of micro-channels at different Reynolds number are given in figs. 3(a) and 3(b), respectively. Figure 3(a) shows that the Nusselt number values of both the CM and SLM are significantly higher than those of the SM under the same Reynolds number, and the heat transfer performance of the SLM is better than that of the CM. Figure 3(b) illustrates that the PEC values of both the CM and SLM are greater than 1. Furthermore, the PEC of the SLM surpasses that of the CM, indicating that the presence of staggered herringbones and inclined ribs has enhanced overall performance, with inclined ribs outperforming staggered herringbones.



Figure 3. Heat transfer characteristics of different types of micro-channels, variations; (a) Nusselt number and (b) PEC with different Reynolds number values

Wall temperature analysis

Figure 4(a) depicts the T_{wx} curves of the SM, CM, and SLM in the *x*-direction. It is evident from the figure that the T_{wx} of the CM and SLM exhibits a relatively gradual change trend, while the T_{wx} of the SM demonstrates a more pronounced variation compared to that of the CM and SLM. The temperature difference, ΔT_{wx} , at the inlet and outlet for the SM is 12.7 K, while it does not exceed 1.7 K for the CM and SLM. This suggests that the temperature uniformity of the CM and SLM is superior to that of the SM. Figure 4(b) shows the thermal resistance RT values of the SM, CM and SLM at different Reynolds number values. The RT of the CM and SLM are much lower than that of SLM, which indicates that the heat transfer capabilities of the CM and SLM are stronger than that of the SM. The conclusion drawn from fig. 4(b) is consistent with that of fig. 4(a).



Figure 5 presents the wall temperature T_w contours on the heating wall for various micro-channels. Specifically, fig. 5 (a) contrasts the T_w of the SLM and CM. Figure 5(a) demonstrates that T_w on the heating surface of the SLM distributed more evenly than that of the CM. Figure 5(b) provides the wall temperature T_w contour on the heating wall of the SM. The wall temperature difference between the inlet and outlet of the SM is significantly greater than that of the SLM and CM. This indicates that the heat dissipation capabilities of the SLM and CM are far greater than that of the SM. The analytical results of the wall temperature T_w in fig. 5(a) and 5(b) are in accordance with the conclusions drawn from the T_{wx} in fig. 4(a) and the thermal resistance R_t in fig. 4(b).



Figure 5. Wall temperature T_w contours of; (a) SLM, CM and (b) SM, at $u_{in} = 0.6$ m/s

Analysis of heat transfer mechanism

The temperature gradient dT/dz of various types of micro-channels thermal boundaries is analyzed to elucidate the heat transfer mechanisms in this section. A symmetric straight line crossing the solid domain, boundary-layer and fluid domain is considered. Therefore, the line formed by point 1 (x = 10 mm, y = 0.8 mm, z = 0 mm) and Point 2 (x = 10 mm, y = 0.8 mm,

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z = 2.5 mm) are selected for this analysis, as shown in fig. 6. It can be seen that the dT/dz values of all micro-channels in both the solid and fluid regions are significantly smaller than those at the solid-fluid interface. The dT/dz in the solid region is equal to zero, because of its high thermal conductivity and uniform material. Similarly, all the dT/dz values in the fluid region are much lower than those at the solid-fluid interface. Meanwhile, the dT/dz values of the CM and SLM fluctuate, which is different from that of the SM. This is because the internal structures of the micro-channels have an impact on flow and heat transfer. By contrast, the absence of micro-structures in the SM ensures that there is no disturbance to the fluid, thereby maintaining a smooth dT/dz within the SM. Furthermore, it is observed that the fluctuation in dT/dz is more pronounced in the CM than in the SLM. This is attributed to the greater turbulence effect produced by the staggered herringbones compared with the inclined ribs. It also can be seen that the turbulence caused by the inclined ribs of the SLM predominantly impacts the solid-liquid interface, where a significant temperature gradient exists. In contrast, the turbulence generated by the staggered herringbones of the CM primarily affects the fluid region, where dT/dz is small. This leads to an increased heat transfer resistance and reduced heat transfer efficiency in the CM.

Figure 6 illustrates that the maximum dT/dz values for the SM, CM, and SLM are located at the solid-liquid interface, with values of approximately $1.8 \cdot 10^5$ K/m, $1.4 \cdot 10^5$ K/m, and $0.4 \cdot 10^5$ K/m, respectively. The maximum dT/dz values for the SM and CM are five and 3.5 times greater than that of the SLM, respectively. Additionally, it is observed that the boundary-layer of the SM is the thickest, followed by that of the CM, with the SLM having the smallest boundary-layer (as indicated by the black elliptic dotted line – 1).

The previous analysis clearly demonstrates that the disturbance effect of the SLM is primarily focused at the solid-liquid interface, resulting in a high heat transfer efficiency. On the other hand, the disturbance effect of the CM not only



Figure 6. Temperature gradients dT/dzof different types of micro-channels at $u_{in} = 0.72$ m/s

impacts the solid-liquid interface, but also extends to the fluid region. Consequently, the CM requires a greater pressure drop to enhance the heat transfer capacity.

Flow analysis

Figure 7 shows the pressure drop, ΔP , and local pressure, P_x , as functions of Reynolds number for various micro-channels. As shown in fig. 7(a), the pressure drop ΔP values of the SLM and CM were significantly higher than those of the SM at the same Reynolds number. Compared with the SM, the ΔP of the CM and SLM increased more significantly with an increase in Reynolds number. This is attributed to the generation of chaotic flow and longitudinal vortices induced by the staggered herringbones and inclined ribs, which lead to additional mechanical energy consumption and increased fluid-flow resistance. In addition, the ΔP values of the CM with staggered herringbones were found to be higher than those of the SLM with inclined ribs, indicating that the CM exhibited a higher flow resistance characteristic compared to the SLM. This is consistent with results shown in fig. 6.





To better understand the influence of micro-structures on the flow resistance, the axial variations in the P_x have been studied. As depicted in fig. 7(b), it is evident that the P_x gradually decreases overall along the axial flow direction x. However, there are fluctuations in P_x in some local fluid regions. For instance, the P_x suddenly drops at the inlet due to the entrance effect. Additionally, the P_x also fluctuates near the micro-structures because of the sudden decrease in fluid velocity when impacting the micro-structures. This leads to a slight increase in P_x as a result of the Bernoulli effect. It can be concluded that micro-structure obviously affects the flow behavior of fluids and leads to pressure fluctuations. Meanwhile, when the fluid-flows along the micro-channels, the p_x decreases smoothly to zero at the outlet owing to the weakening and eventual disappearance of the vortex flow induced by the micro-structures. Furthermore, it is evident that both the values and fluctuations of the P_x for the CM are greater than those of the SLM, indicating that a greater pressure drop is required for the fluid to,flow through the CM.

Figure 8(a) depicts the changes in the f of various types of micro-channels with increasing Reynolds number. It is observed that both the CM and the SLM exhibit a gradual decrease in f as the Reynolds number increase, but the f of the CM remains higher than that of the SLM. Because f is directly proportional to ΔP , it can be concluded that ΔP for CM is greater than that for SLM at the same Reynolds number, resulting in a higher f value for the CM com-



Figure 8. Flow characteristics of different types of micro-channels; (a) f with different Reynolds number and (b) dimensionless secondary flow intensity ΔSe at Re = 833, u_{in} = 0.72 m/s

pared to the SLM. This suggests that the geometry of staggered herringbones is more intricate than that of the inclined ribs, leading to a more significant obstruction of fluid-flow by the CM compared to the SLM.

Figure 8(b) illustrates the variation in the dimensionless secondary flow intensity ΔSe with respect to the x-direction for both the SLM and CM. It is evident that the ΔSe of the CM is higher than that of the SLM, indicating that a greater pressure drop is required for fluid-flow in the CM. The finding here agrees with that of the pressure drop analysis. In comparison the CM, the variation of ΔSe in the SLM is more gradual, indicating that the longitudinal vortex induced by the SLM decays at a slower rate than that of the CM. This flow characteristic of the SLM enables it to achieve higher heat transfer performance at lower pressure drops.

Conclusions

In this study, the flow and heat transfer characteristics of a chaotic micro-channel CM and a single longitudinal vortex micro-channel SLM were investigated. The essential differences in the heat transfer mechanisms between the two micro-channels were revealed. The main conclusions are as follows.

- The heat transfer resistance is mainly concentrated at the solid-fluid interface. Therefore, enhancing the disturbance of micro-structures at the solid-liquid interface has a more pronounced effect than enhancing convection in the fluid region.
- The enhanced heat transfer mechanism of the SLM is inducing the fluid to impact the solid-fluid interface, rather than improving the convection of the central fluid, so as to obtain superior heat transfer performance.
- The longitudinal vortex decays more slowly than the chaotic flow, resulting in a higher heat transfer performance and lower pressure drop.

Declaration of competing interest

All authors declare no competing financial interests to influence the work reported in this paper.

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