

Numerical Study on Heat Transfer Through In-Phase, Out-Phase and Straight Microchannel



Zahoor Bhat, Yogesh M. Nimdeo, Harish Pothukuchi, and Divya Haridas

Abstract Thermal cooling is very important for the smooth functioning of electronic devices. The simulations were performed to obtain the pressure, velocity and temperature distributions for different flow conditions. Further, the surface Nusselt number is calculated to estimate and compare the thermal performance of straight and wavy microchannels. Water is used as a cooling medium and the inlet Reynolds number varies from 50–600 by varying the coolant inlet velocity. It was observed that as compared to the straight channel, in-phase and out-phase channel shows 49.53% and 45% enhancement in heat transfer, respectively. The comparison is made on the basis of the average Nusselt number obtained at $Re = 300$. However, the lower pressure drop is obtained in a straight channel followed by in-phase and out-phase wavy channels, respectively.

Keywords Wavy microchannel · Thermal cooling · In-phase channel · Out-phase channel · Microchannel heat sinks

1 Introduction

To design and enhance the thermal performance of microfluidic heat transfer devices, understanding the heat transfer characteristics in microchannels is important. Because of continuous functioning, the heat generated from such electronic components increases. The generation of high heat fluxes from miniature electronic equipment is a significant hurdle in the development of new-generation electronic gadgets. The performance of electronic elements is highly contingent on their working temperatures, and thus adequate heat dissipation is extremely important to detour their failure and to assure invariant performance. There exist numerous techniques embraced

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for the thermal management of electronic parts among which fluid flow via small dimensional channels has been ascertained to be an effective way of heat dissipation. Several researchers have pursued their work in this direction to enhance the thermal performance of such ongoing operating electronic gadgets by dispersing heat using microchannels. The heat transfer rates acquired from mini and microchannels are significantly higher as compared to conventional cooling techniques, viz. use of fins, fans, blowers, etc.

2 Literature Review and Objective

The era of microchannels started in 1981 when Tuckerman and Pease [1] used microchannels as high-performance heat sinks for the cooling of electronic devices. During the earlier phase of microchannel evolution, conventional microchannels were used. These were in the form of straight channels in which streamlines of the flowing fluid were nearly straight. The poor mixing of the fluid resulted in inefficient heat transfer. It is, therefore, very important to alter the structure of microchannels. When fluid flows through curved surfaces, there occurs a proper mixing of coolant and formation of vortices which inhibits the continuous growth of boundary layer and hence enhances heat transfer performance. Owing to the enhanced thermal performance of microchannels, many researchers have studied the heat transfer through wavy channels by both experimental and numerical approaches. Comini et al. [2] have performed numerical analysis to estimate heat transfer through a three-dimensional wavy channel. They noticed that decreasing the aspect ratio enhances the Nusselt Number. Ahmed et al. [3] numerically studied the heat transfer through wavy channel using nanofluids. The result showed that heat transfer improves due to nanoparticle volume fraction, Reynolds number and amplitude rather than the wavelength of channel. Sui et al. [4, 5] reported that the heat transfer performance of wavy channels is much superior to the straight channels through their experiments and numerical simulations. Jixiang et al. [6] studied the influence of wavy phase shifts on heat transfer through corrugated wavy channels. They noticed that the Nusselt number and friction factor decreases by increasing phase shifts. Rush et al. [7] experimentally observed the flow and heat transfer in sinusoidal wavy passages. They observed that relative amplitude is important in the stability of wavy channels at low Reynolds number. An amplitude larger than the related amplitude delayed the initial onset of instability. At higher Reynolds number, the effect of relative amplitude diminishes. Shuo et al. [8] numerically investigated the heat transfer enhancement of symmetric and parallel wavy microchannel heat sinks with a secondary branch design. As a result of the study, the in-phase configuration has a slightly higher Nusselt number in low Reynold ranges. But in case of moderate and high Reynold ranges, the heat transfer performance of the in-phase configuration is substantially less than that of the out-phase configuration. Lin et al. [9] investigated that thermal performance of wavy microchannels can be improved by changing amplitude and wavelength. Naphon et al. [10] studied the effect of various geometric configurations on flow and

temperature distribution in wavy plate channels. They found that wavy plate with sharp edges significantly effect the heat transfer enhancement especially in V-shaped wavy channels.

According to the literature cited above, the geometry of the channel has a significant impact on heat transfer characteristics. Wavy microchannels possess a significant heat transfer advantage over straight microchannels. But in the wavy microchannels, the comparative thermal performance of in-phase and out-phase geometry of the wavy microchannels has not been addressed in detail as per our knowledge. Therefore, in this study, the performance characteristics of wavy sinusoidal channels (in-phase/out-phase) are compared to those of a straight channel. The analysis is carried out using velocity contours and variation of temperature, pressure, average Nusselt number and surface heat transfer coefficient as a function of distance along the axial length of the microchannel to get the thermal performances.

For all the simulations performed, water is used as the cooling medium and the inlet flow rate is varied to achieve different Reynolds numbers ... ranging from 50–600.

3 Materials and Methods

The numerical study has been conducted on two wavy channels of different phase shifts and a straight rectangular reference channel. In case of a wavy channel, the phase differences are introduced by varying the position of the top plate with that of the bottom plate. All the three channels are of rectangular cross section with specific dimensions (Length = 107 mm, Height = 2 mm, Width = 5 mm). The bottom plate made of copper is maintained at constant heat input of 5 W. Top plate which is also made of copper is maintained at isothermal condition. Rest two side walls are maintained at adiabatic condition. Figure 1 shows the schematic diagram of all the three channels used in the study. The inlet water is maintained at a temperature of 27 °C. Table 1 shows the description of variables used the numerical study.

Numerical Schemes:

The numerical simulations are performed using a finite volume based commercial solver ANSYS Fluent. The computational domain is discretized and SIMPLE Scheme is opted to solve pressure–velocity coupling. The momentum and energy

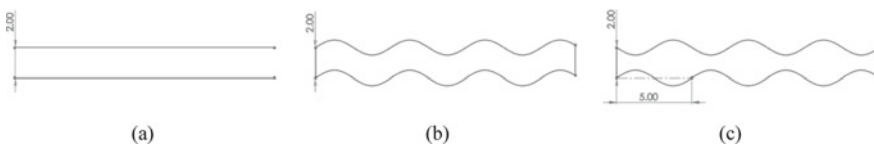


Fig. 1 Schematics of **a** straight rectangular channel (the reference for comparison), **b** in-phase and **c** out-phase wavy microchannel

Table 1 Numerical variables

Parameters	Values
Pitch of wavy channel (P)	5 mm
Amplitude of wavy channel (α)	0.5 mm
Inlet water temperature (T_{in})	27 °C
Heat input (Q)	5 W
Reynolds Range (Re)	50–600

equations solved by first order upwind scheme. The convergence criteria are chosen as residuals reduced to the order of 10^{-6} .

All simulations are conducted using hexahedra mapped meshes with 0.15 mm element size. The domain as a whole is discretized into 335,907 elements. Figure 2 illustrates how the chosen mesh distribution is organized.

Governing Equations:

The steady-state mass, momentum and energy conservation equations for the fluid are given as follows:

$$\text{Mass conservation : } \nabla \cdot (\rho U) = 0 \tag{1}$$

$$\text{Momentum conservation : } (U \cdot \nabla)\rho U = -\nabla P + \mu \nabla^2 U \tag{2}$$

$$\text{Energy conservation : } U \cdot \nabla T = \frac{k}{\rho c_p} \nabla^2 T \tag{3}$$

The heat transfer coefficient is measured by using the following formula,

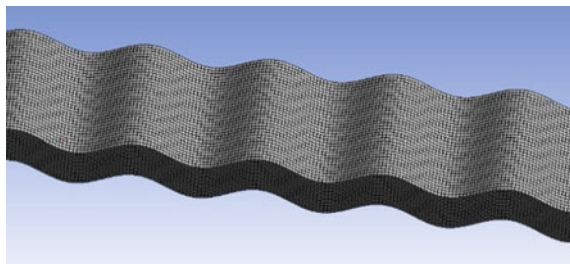
$$h = \frac{Q'}{T_w - T_m} \tag{4}$$

where Q' = applied wall heat flux

T_w = Area weighted average wall temperature.

T_m = Bulk fluid temperature ($\frac{T_i+T_o}{2}$).

Fig. 2 Schematic of mesh distribution for in-phase channel



T_i = inlet fluid temperature and T_0 = outlet fluid temperature.

Using the generated heat transfer coefficient data, the Nusselt number is calculated as

$$Nu = \frac{hD_h}{K} \tag{5}$$

Here, $D_h = \frac{4(H \times W)}{2(H+W)}$ is the hydraulic diameter of the channel and ‘k’ is the thermal conductivity of the fluid.

Boundary Conditions:

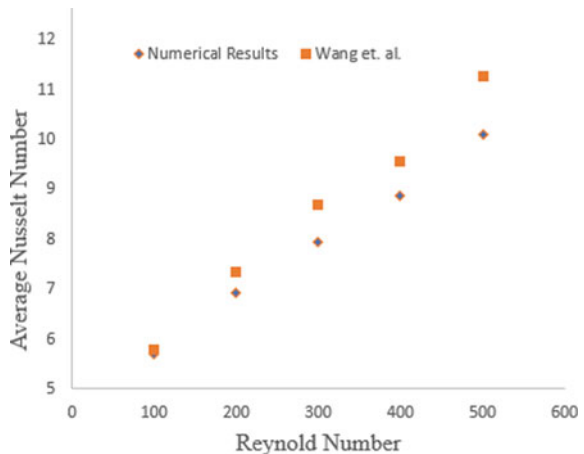
The following inlet and outlet boundary conditions are considered for the calculation of velocity, pressure and temperature.

- Inlet velocity condition: $u = U_{in}$
- The outlet pressure condition: $P = P_{atm}$
- Inlet fluid temperature: 300 K
- Top wall: Isothermal condition
- Bottom wall: Constant heat flux condition
- Side walls: Adiabatic condition.

Numerical Validation:

In the present work, the validation for verifying the performance prediction capability of the numerical model is carried out by comparison of measured Average Nusselt number as a function of Reynolds number with the reported study by Wang et al. [8] in the literature at similar boundary conditions. The validation is made on the basis of Nusselt number for in-phase channel and predictions of results are found to be in well agreement with that reported by Wang et al. as shown in Fig. 3.

Fig. 3 Comparison of average Nusselt number obtained from present computation with results of Wang et al. [8]



4 Results and Discussions

4.1 Velocity Variation

The heat transfer performance of wavy channels with different phase shifts has been studied numerically for Reynolds number range of 50–600. Figure 4 illustrates the velocity contours at $Re = 150$. It can be seen that velocity is maximum towards the centre of the channel in all cases. However, overall velocity field is completely different in both the wavy channels. In case of in-phase wavy channel, the gap between the top and bottom wall remains constant, and therefore, velocity streamlines of the fluid towards the centre are almost continuous along the axial direction. While in case of out-phase channel, there is non-uniformity in the gap between the two walls due to formation of converging and diverging areas and it leads to the discontinuation of the velocity streamlines along flow direction. Therefore, in out-phase channel, there is maximum velocity in converging areas and minimum velocity in diverging areas. Based on discontinuity in the formation of velocity boundary layer, wavy microchannels exhibit enhancement in heat transfer over a straight microchannel.

From the velocity vectors depicted in Fig. 5, it is clearly evident that in straight channel (Fig. 5a), there is no formation of Dean Vortices in the flow. The fluid flows smoothly with improper mixing near the heated wall and results in poor convection. While in wavy channels (Fig. 5b, c), there is formation of Dean Vortices which leads to proper mixing of fluid and enhances the heat transfer performance of wavy channels.

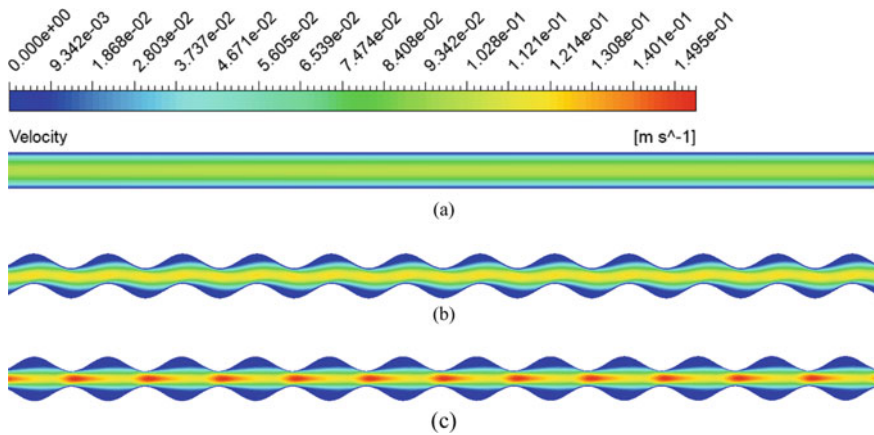


Fig. 4 Velocity contours of **a** straight channel, **b** in-phase and **c** out-phase wavy mini-channel at $Re = 150$

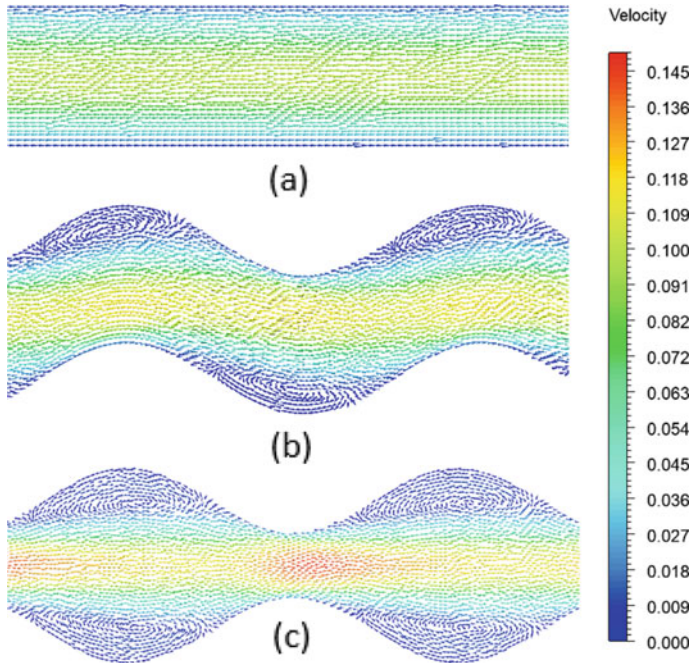


Fig. 5 Representation of velocity vectors along flow direction: **a** straight channel, **b** in-phase channel, **c** out-phase channel

4.2 Temperature Variation

In straight channel (Fig. 6a), the thickness of isotherms representing the thickness of thermal boundary layer is considerable and uniform along the flow direction. Therefore, the temperature gradients become weakened along the flow direction. However, in wavy channels (Fig. 6b, c), thermal boundary layer is not smooth along the flow direction. There arises disturbance in the formation of thermal boundary layer due to channel geometry and hence enhances the thermal performance of wavy channel.

The graphical variation of temperature along axial direction is shown in Fig. 7. Here also in wavy channels the temperature is not increasing gradually like in case of straight channel. The temperature starts increasing when fluid starts flowing over crest areas while it starts falling at trough areas. Which means destabilization of thermal boundary layer that results in enhancement in heat transfer in wavy channels than that of straight channel. However, in case of in-phase and out-phase channel, in-phase channel shows better heat performance than that of out-phase channel.

In Fig. 8, the variation of pressure along the flow direction is shown. Wavy microchannels have a greater pressure drop than straight microchannels. This is because wavy structure of the channel disturbs the entire flow field and there is rise of perturbation and vorticity inside the waves which causes the pressure to rise.

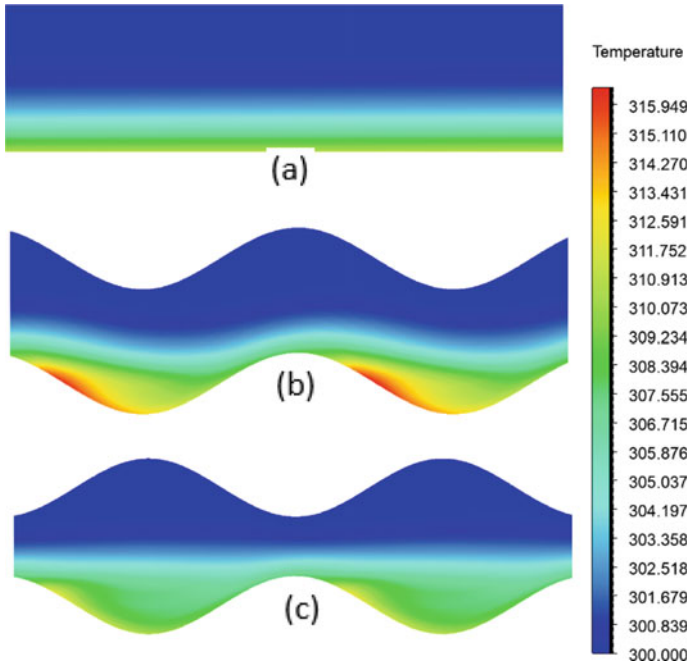
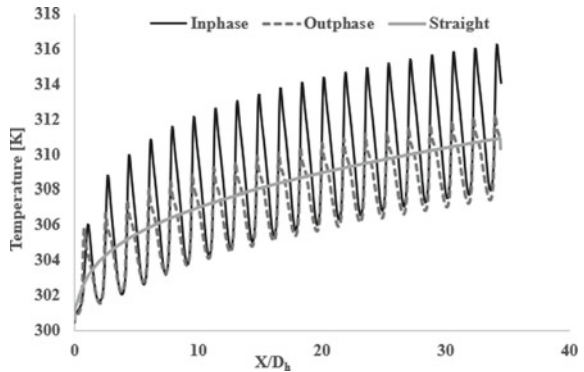


Fig. 6 The contours of temperature distribution at $Re = 150$: **a** straight channel, **b** in-phase channel, **c** out-phase channel

Fig. 7 Axial variation of temperature at $Re = 150$



Further, it can be clearly seen that out-phase channel in comparison to in-phase channel attains higher pressure drop and requires higher input power to maintain the flow.

Figure 9 shows the variation of Surface Heat Transfer Coefficient in all the channels. In wavy channels, the trend follows the sinusoidal variation along axial direction with maximum peaks occurs at crest of the channel (corresponding to bottom wall) followed by lower peaks at trough portion of channel.

Fig. 8 Variation of Pressure in Axial direction at Re = 150

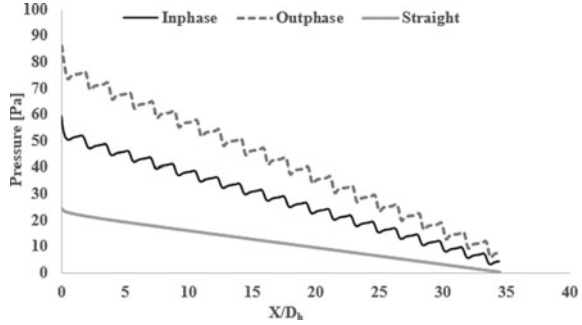


Fig. 9 Surface heat transfer coefficient at Re = 150

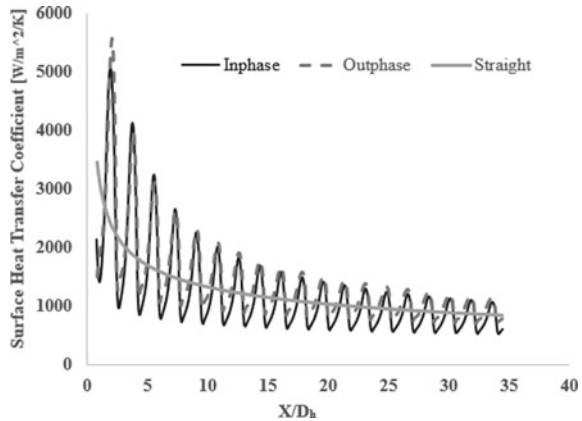
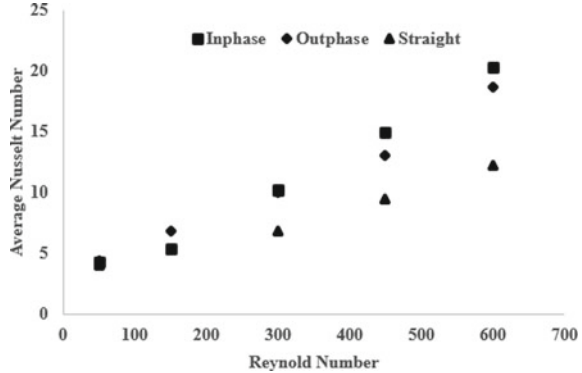


Figure 10 shows the average variation of Nusselt number with respect to Reynolds number. It is clearly seen that Nusselt number increases by increasing Reynolds number and wavy channels shows higher variation in average Nusselt number than that of straight channel. The reason for better thermal efficiency of wavy channels is because the thermal boundary layers are redistributed more due to the wave structure at the top and bottom walls. Also, the average Nusselt number comes out to be high for in-phase channel as compared to out-phase microchannel and for a given Reynolds number, the mixing of flow and the resulting instabilities occur further upstream in in-phase channel as compared to out-phase channel. Therefore, the mixing of the flow enhances the heat transfer in in-phase channel as compared to out-phase channel.

Moreover, for microchannel based heat sinks, in case of straight channels in laminar flow region, the flow becomes regular with considerable thermal boundary layer thickness. This led to the deterioration of heat transfer performance along the flow direction. However, for wavy channels Dean vortices quickly develop along the flow direction and disrupt the thermal boundary layer. It leads to an enhanced heat transfer performance of the wavy microchannel heat sinks.

Further experimental studies need to be carried out for better insight to investigate the heat transfer and fluid flow phenomena in great details.

Fig. 10 Average Nusselt number w.r.to Reynolds number



5 Conclusion

Numerical study has been performed to analyse the heat transfer characteristics of wavy microchannels with that of straight microchannel. The results interpret that wavy channel shows higher heat transfer characteristics than that of straight channel. Wavy channel exhibits a sinusoidal variation in flow which results perturbation and formation of vortices, results an enhancement in heat transfer. Also, Average Nusselt number increases with increase in Reynolds number in all the cases.

In-phase channel shows better thermal performance than out-phase channel. However, higher pressure drop is obtained in wavy channels than straight channels. The overall study indicates that microchannel heat sinks based on wavy channel have advantage over those with straight channels.

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Nomenclature

P	Pitch of wavy channel (mm)
α	Amplitude of wavy channel (mm)
Re	Reynolds Number
T_{in}	Inlet water temperature (K)
Q	Heat input (W)
p	Pressure
U	Velocity
H	Height of channel
W	Width of channel
MCHS	Micro channel heat sinks

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