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Numerical Investigation of Wavy Microchannel Heat Sinks with Dual-Outlet Architectures Using Fe_3O_4 Nanofluids

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ABSTRACT: The thermo-hydraulic performance of three wavy microchannel heat-sink architectures, namely single-inlet single-outlet (SISO), single-inlet dual-outlet (SIDO), and branched single-inlet dual-outlet (BSIDO) configurations, are numerically investigated using deionized water and low-concentration Fe_3O_4 nanofluids for electronic thermal management. Steady-state laminar flow and heat transfer simulations are performed over Reynolds numbers of 700–1300 for Fe_3O_4 nanoparticle concentrations of 0.1%, 0.3%, and 0.5%, within a consistent numerical framework and under prescribed wall thermal boundary conditions. The Nusselt number, friction factor, and Performance Evaluation Criterion (*PEC*) are employed to assess the coupled thermal and hydraulic performance of the different architectures and to elucidate the effects of channel topology and coolant thermophysical properties. A quantitative sensitivity analysis is further conducted to identify the parameters governing cooling performance, while mesh-independence tests and comparisons with established empirical correlations are used to assess the reliability of the numerical model. The results show that the introduction of Fe_3O_4 nanofluids consistently improves the overall thermo-hydraulic performance, with all nanofluid cases yielding *PEC* values greater than unity. Among the configurations investigated, the SIDO architecture provides the most favorable overall thermo-hydraulic performance, benefiting from effective flow distribution and enhanced heat transfer with moderate pressure losses. Although the BSIDO configuration produces stronger local convective heat transfer, the repeated flow separation induced by the wavy channels results in substantially higher hydraulic losses, limiting its overall performance.

KEYWORDS: Nanofluids; heat sink; flow characteristic; heat transfer enhancement; bionic wavy channel; comprehensive thermo-hydraulic performance

1 Introduction

As high heat flux electronics and power batteries impose stricter heat dissipation demands, conventional air-cooled straight-channel heat sinks fail to meet industrial requirements. Thermal performance can be boosted via two approaches: heat sink structural optimization and high performance nanofluid coolants. Nanofluids deliver superior thermal conductivity and nanoparticle Brownian agitation, making them promising cooling alternatives.

Accordingly, extensive numerical and experimental work has focused on heat sink geometry. Applications include photovoltaic systems [1,2] and flow-boiling devices [3,4], while the proposed architectures encompass minichannel heat sinks [5], stacked double-layer mini- and microchannels [6], manifold

microchannels [7], channels with combined manifolds [8], semiconductor-package heat sinks [9], laser-powder-bed-fused designs [10], microchannel cold plates [11], and lattice-based heat sinks [12].

Many experimental studies have quantified thermal improvements from novel channel designs. Tesla-shaped heat sinks achieve 55.3–88.3% heat transfer enhancement [13]; staggered manifold fins perform best among manifold layouts [14]; sinusoidal passages gain 7.8–13.1% improvement [15]; cascaded PCM raises heat transfer by 25.48% [16]; topology-optimized structures via genetic algorithms also show outstanding cooling capacity [17].

Ribs offer another widely investigated means of disturbing the near-wall flow. In a diamond microchannel, the orthogonal ribs studied by Du et al. [18] produced a maximum heat transfer increase of 182.8%. Ali et al. [19] found that V-shaped ribs with a 35 degree angle yielded particularly strong thermal performance. Najafpour et al. [20] showed that, at $Re = 1200$, the Nu of case 2 exceeded that of the baseline by 46.75%, and Fu et al. [21] achieved a 37.5% reduction in thermal resistance with pin fins. Related concepts include plate fins [22], tapered pin fins [23], micrometer-scale composite pin fins [24], and slanted secondary channels [25].

Enhancement strategies are not limited to ribbed passages. Porous-media representations have been adopted for heat sink analysis [26], including systematic studies of porosity and pore density [27]. Other developments comprise a constructal finned heat sink incorporating PCM [28], nano-enhanced PCM cooling [29], a PCM-based manifold microchannel system [30], porous structures filled with PCM [31], impinging-jet arrays [32], and combined pin-fin/jet configurations [33].

Heat sink design has also benefited from topology and multi-objective optimization. Table 1 summarizes representative topology-optimization studies [34–37] and multi-objective investigations [38–41], together with the heat sink type, design variables, and principal outcome reported in each study. Beyond standard optimization algorithms, Response Surface Methodology (RSM) accurately quantifies parametric influences on local Nusselt numbers for channel cooling design [42].

Because their thermal conductivity can exceed that of air or water [43–45] and nanoparticles introduce Brownian-motion-induced microscale transport, nanofluids are frequently selected as alternative coolants. Examples include Al_2O_3 - TiO_2 - CuO /water in Li-ion battery thermal management [46] and Fe_3O_4 -water in cooling systems for electronic components [47,48]. Farade et al. systematically compared the trade offs of various nanofluids in transformer cooling systems, highlighting significant differences among different types of nanofluids in terms of viscosity, chemical stability and thermal enhancement capacity [49,50]. Notably, nanoparticle aggregation significantly distorts thermo-hydraulic performance predictions, while artificial neural networks trained via the Levenberg-Marquardt algorithm deliver high precision Nu prediction for battery and solar nanofluid cooling systems [51,52].

In summary, existing research separately investigates heat sink structures and nanofluid coolants. Channel layout, coolant thermophysical properties and flow conditions jointly determine overall cooling performance, yet several research directions remain insufficiently explored:

- (1) Most studies focus on single-inlet single-outlet channels; systematic comparisons of straight and bionic wavy double-outlet heat sinks are limited.
- (2) Channel geometry and nanofluid medium are mostly optimized separately; synergistic heat transfer coupling analysis is scarce.
- (3) Few published papers conduct quantitative parametric sensitivity analysis for nanofluid composite heat sinks.

This study implements steady state laminar fluid heat transfer simulations under prescribed wall thermal boundary conditions at $Re = 700\text{--}1300$ using nanofluids with different mass fractions, to provide design references for double-outlet heat sinks in electronic cooling. Its core novelties and contributions are as follows:

- (1) Under a unified numerical framework, laminar comparative simulations are performed for SISO, SIDO and BSIDO heat sinks with water and Fe_3O_4 nanofluids.
- (2) Nusselt number, friction factor and PEC are adopted to compare thermo-hydraulic performance of three layouts and reveal coupling effects between channel topology and coolant properties.
- (3) Quantitative sensitivity analysis of Reynolds number, heat sink geometry and nanoparticle concentration is carried out to identify key parameters dominating cooling performance.

Table 1: Optimization methods and conclusions of heat sink.

Ref.	Heat Sink Kind	Method	Key Factors	Main Conclusions
[34]	Heat sink in TEG waste heat recovery system	Topology optimization	Different fin structures	Thermoelectric conversion efficiency is enhanced by 10.7%
[35]	Microchannel heat sink for electronic devices	Topology optimization	Topology optimized structures	Convective heat transfer coefficient is increased by 55.5% compared to straight channel
[36]	Bionic leaf-vein like heat sinks	Topology optimization	Volume constraint	Heat dissipation performance is first enhanced and then stabilised
[37]	Micro-pin fin heat sinks	Topology optimization	Volume fraction, porosity and diameter	The max heat flux can reach 400 W/cm^2
[38]	Manifold microchannel heat sink with curved corners	Multi-objective optimization	Left curved corner and right curved corner	Thermal resistance is reduced by 4.02%
[39]	Microchannel heat sink for chip heat transfer systems	Multi-objective optimization	Thermal stress	Thermal stress can be reduced by 48.86%
[40]	Heat sink for thermal management system	Multi-objective optimization	Reynolds number	Nusselt number can be increased by 7.41%
[41]	Symmetrical hierarchical microchannel heat sinks	Multi-objective optimization	Secondary channel width and trapezoidal bottom angle	Thermal resistance is reduced by 23.79%

2 Method

2.1 Geometric Model

Three configurations were established to examine the effect of flow-path design: the single inlet and single outlet heat sink (SISO), single inlet and double outlets heat sink (SIDO), and bionic wave-shaped single inlet and double outlets heat sink (BSIDO). The SISO structure acts as the traditional baseline widely applied in low power chips. The SIDO design was adopted to eliminate downstream hotspots via symmetric flow splitting, which cannot be realized by standard SISO channels. The BSIDO geometry further introduces periodic flow disturbance to disrupt continuous thermal boundary layers, which is a typical passive heat transfer enhancement method. Their two-dimensional geometries are presented in Fig. 1. For each case, a nanofluid of prescribed mass fraction enters a rectangular domain measuring 40 mm in length, 40 mm

in width, and 9 mm in height. Both the SISO inlet and outlet are 10 mm wide, whereas each outlet in the double-outlet designs is 5 mm wide. The bionic waves have a vertical amplitude of 2 mm, and heat is transferred to the coolant as it traverses the channel.

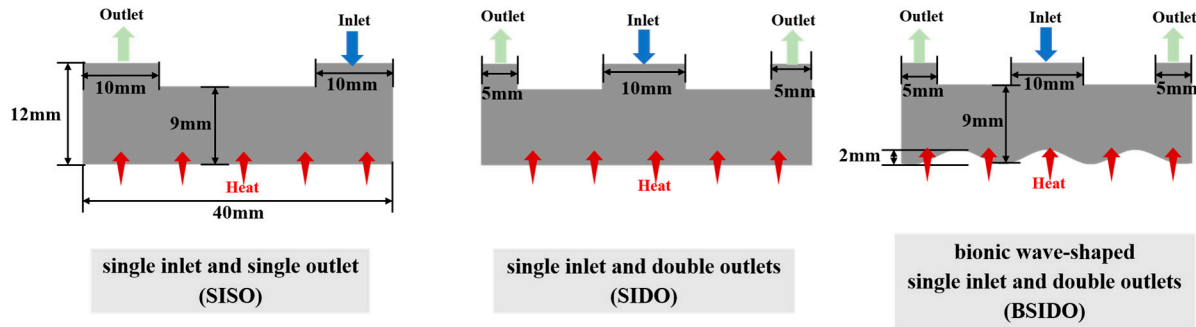


Figure 1: Heat sink model.

2.2 Governing Equations

To simplify the steady laminar fluid heat transfer simulation under prescribed wall thermal boundary conditions while maintaining prediction accuracy, the following fundamental assumptions were adopted in the present single-phase nanofluid model:

- (1) The flow inside the heat sink is steady, incompressible laminar flow within $Re = 700-1300$.
- (2) Fe_3O_4 nanoparticles are uniformly dispersed in deionized water without sedimentation, agglomeration or velocity slip between solid particles and base fluid.
- (3) All thermophysical properties of working fluids (density, specific heat, viscosity, thermal conductivity) remain constant independent of local fluid temperature.
- (4) Contact thermal-resistance effects are neglected at the channel wall boundaries.
- (5) Gravity effect and radiative heat transfer are negligible for compact electronic heat sinks under low temperature difference.

Based on the above assumptions, the governing equations are expressed as follows:

$$\nabla \cdot v = 0 \quad (1)$$

$$\rho v \cdot \nabla v = -\nabla p + \mu \nabla^2 v \quad (2)$$

$$\rho c_p v \cdot \nabla T = k \nabla^2 T \quad (3)$$

It should be noted that this two-dimensional model consists solely of the fluid computational domain. A constant heat flux boundary condition is imposed on the bottom wall of the fluid domain, while all other external boundaries are set as adiabatic. Since no solid domain heat conduction equation is solved, the wall temperature is determined directly from the fluid energy equation.

2.3 Boundary Conditions and Thermophysical Properties

The commercial CFD software ANSYS Fluent was adopted to solve the continuity, momentum and energy governing equations. Detailed numerical discretization and solution settings are listed in Table 2:

Table 2: Boundary conditions and numerical solution settings.

Item	Setting
Flow regime	Two dimensional, incompressible, and laminar
Inlet boundary	Velocity inlet
Inlet temperature	289.15 K
Outlet boundary	Pressure outlet, relative pressure of 0 Pa
Heating boundary	A constant heat flux of 15,000 W/m ² is applied at the bottom
Remaining external walls	Adiabatic
Solution type	Steady state
Pressure-velocity coupling	SIMPLE algorithm, Rhie-Chow: momentum based
Gradient discretization	Least Squares Cell Based
Pressure discretization	Second-order scheme
Momentum discretization	Second-order upwind
Energy discretization	Second-order upwind
Convergence criteria	Residuals of continuity, momentum and energy equations < 10 ⁻⁶

For the steady laminar simulations, the SIMPLE algorithm was used for pressure-velocity coupling with the Rhie-Chow momentum-based flux type to suppress pressure oscillations. The gradient reconstruction adopts the Least Squares Cell Based method. The pressure term is discretized by the second-order scheme, and the momentum and energy equations use the second-order upwind scheme, which yields second-order spatial accuracy.

The primary reason for selecting Fe₃O₄ nanofluid as the cooling medium in this study is that it exhibits magnetic properties, thereby creating opportunities to utilize a magnetic field to achieve active heat transfer optimization in subsequent research. Physical properties of the working fluids [53] are listed in Table 3.

Table 3: Physical property parameters of nanofluids.

Type of Fluid	ρ kg/m ³	c_p J/(kg·K)	m Pa·s	λ W/(m·K)	Pr
Base fluid (deionized water)	997.1	4179	0.001004	0.6130	6.845
Fe ₃ O ₄ nanoparticles	5180	670	-	80	-
0.1%Fe ₃ O ₄ nanofluids [53]	997.9	4178.32	0.001029	0.6136	7.007
0.3%Fe ₃ O ₄ nanofluids [53]	999.5	4176.97	0.001083	0.6140	7.368
0.5%Fe ₃ O ₄ nanofluids [53]	1001.1	4175.62	0.001141	0.6147	7.751

2.4 Data Reduction

Hydraulic diameter calculation formula:

$$D = \frac{4A_1}{L} \quad (4)$$

where A_1 is the effective flow cross-sectional area, L is the wetted perimeter of the cross-section.

Reynolds number calculation formula:

$$Re = \frac{\rho v d}{\mu} \quad (5)$$

where ρ denotes fluid density, v is flow velocity, d is characteristic length, and μ is dynamic viscosity.

Volume fraction of nanofluids calculation formula:

$$\varphi = \frac{1}{\frac{1}{\omega} \cdot \frac{\rho_{np}}{\rho_{nf}}} \quad (6)$$

where ω is nanofluid mass fraction, ρ_{np} is nanoparticle density, and ρ_{nf} is nanofluid density.

Density of nanofluids calculation formula:

$$\rho_{nf} = (1 - \varphi)\rho_{bf} + \varphi\rho_{np} \quad (7)$$

where ρ_{bf} is base fluid density.

Specific heat capacity of nanofluids calculation formula:

$$c_{nf} = (1 - \varphi)c_{bf} + \varphi c_{np} \quad (8)$$

where c_{bf} is base fluid specific heat capacity, c_{np} is nanoparticle specific heat capacity.

Convective heat transfer coefficient:

$$h = \frac{Q}{T_w - T_f} \quad (9)$$

where Q is the heat flux density, T_w is copper plate surface temperature, T_f is nanofluid bulk temperature.

Nusselt number:

$$Nu = \frac{hD}{\lambda} \quad (10)$$

where λ is fluid thermal conductivity.

Flow resistance coefficient:

$$f = \frac{2D}{\rho v^2} \cdot \frac{\Delta p}{\Delta l} \quad (11)$$

where Δp is inlet-outlet pressure drop, Δl is pressure measurement length.

Overall thermo-hydraulic merit was quantified with the performance evaluation criterion (PEC), which combines the Nusselt-number enhancement with the associated resistance penalty. A value above unity means that the thermal gain outweighs the increase in flow resistance, and a larger value represents a more favorable compromise. Conversely, $PEC < 1$ indicates that the resistance cost exceeds the heat transfer benefit. The criterion was evaluated as follows:

$$PEC = \left(\frac{Nu}{Nu_{water}} \right) / \left(\frac{f}{f_{water}} \right)^{\frac{1}{3}} \quad (12)$$

2.5 Grid Independence Study

A grid independence study was carried out for the three models at $Re = 700$ with a nanofluid mass fraction of 0.1%. Six sets of meshes with different resolutions were generated for each configuration, and the predicted Nusselt number (Nu) and friction factor (f) are compared in Fig. 2. The relative deviations of Nu and f obtained between the two finest mesh schemes are both less than 1%, which indicates that further mesh refinement has a negligible influence on these two parameters. To balance numerical accuracy and

computational cost, meshes containing 53,996, 57,498 and 61,912 elements were selected for the subsequent simulations of the SISO, SIDO and BSIDO heat sinks, respectively.

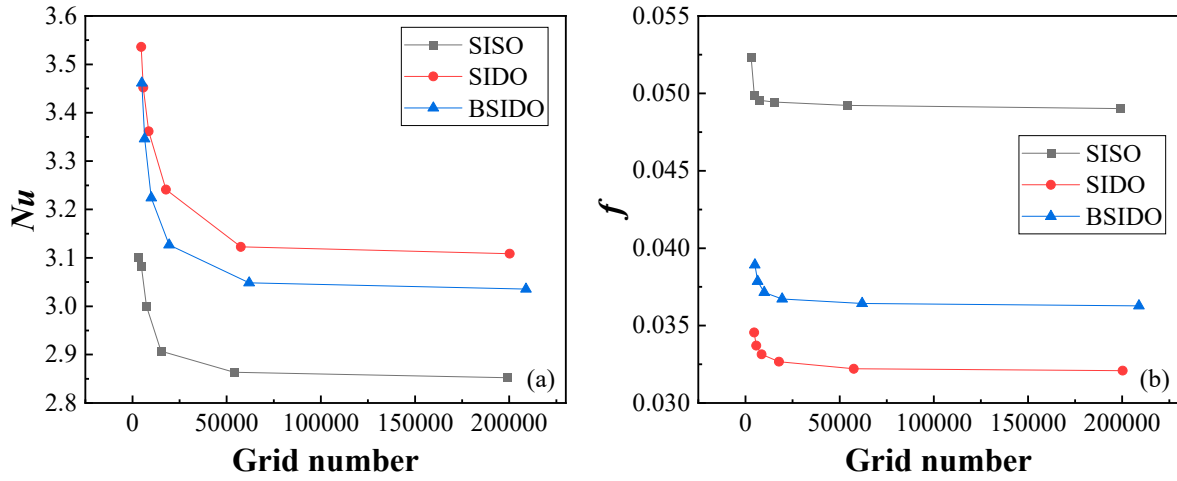


Figure 2: Grid independence verification. (a) Nu , (b) f .

2.6 Model Verification

Simulation results were benchmarked against empirical correlations from Ref. [54]:

$$Nu = 0.566(1 + \varphi)^{0.57} Re^{0.20} Pr^{0.18} \quad (13)$$

$$f = \frac{1.328}{\sqrt{Re}} \quad (14)$$

As shown in Fig. 3, simulated Nu and f values align closely with empirical predictions, following identical trends. The maximum deviation between simulated and empirical Nu values is 3.51%, while the maximum discrepancy for the friction factor is 5.21%. These results confirm the reliability of the numerical approach.

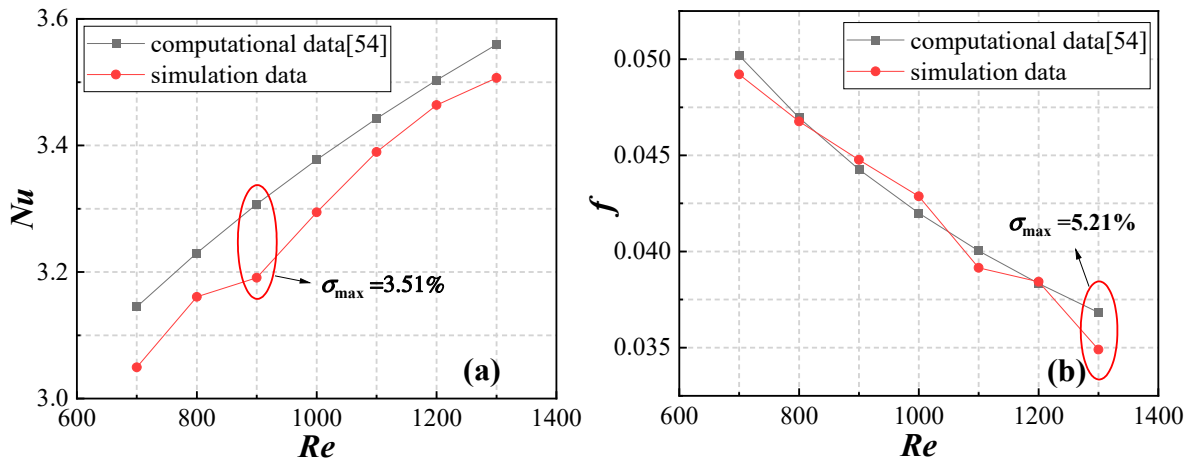


Figure 3: Comparison between experimental values and theoretical values [54]. (a) Nu , (b) f .

3 Results and Discussion

3.1 Single Inlet and Single Outlet Heat Sink (SISO)

The flow and thermal behavior of SISO is first examined through the velocity and temperature fields in Figs. 4 and 5, respectively.

Velocity contours together with streamlines for the SISO heat sink at $Re = 700$ – 1200 are plotted in Fig. 4. Fluid flows along the curved channel to form a mainstream zone, and noticeable recirculation vortices emerge at the left inner wall, upper middle part and right corner of the passage, matching the low velocity regions in the contour map. Increasing Reynolds number leads to higher peak velocity and wider distribution of high speed flow. The size and position of separation vortices keep relatively stable across the tested laminar conditions, demonstrating persistent flow separation inside the SISO channel.

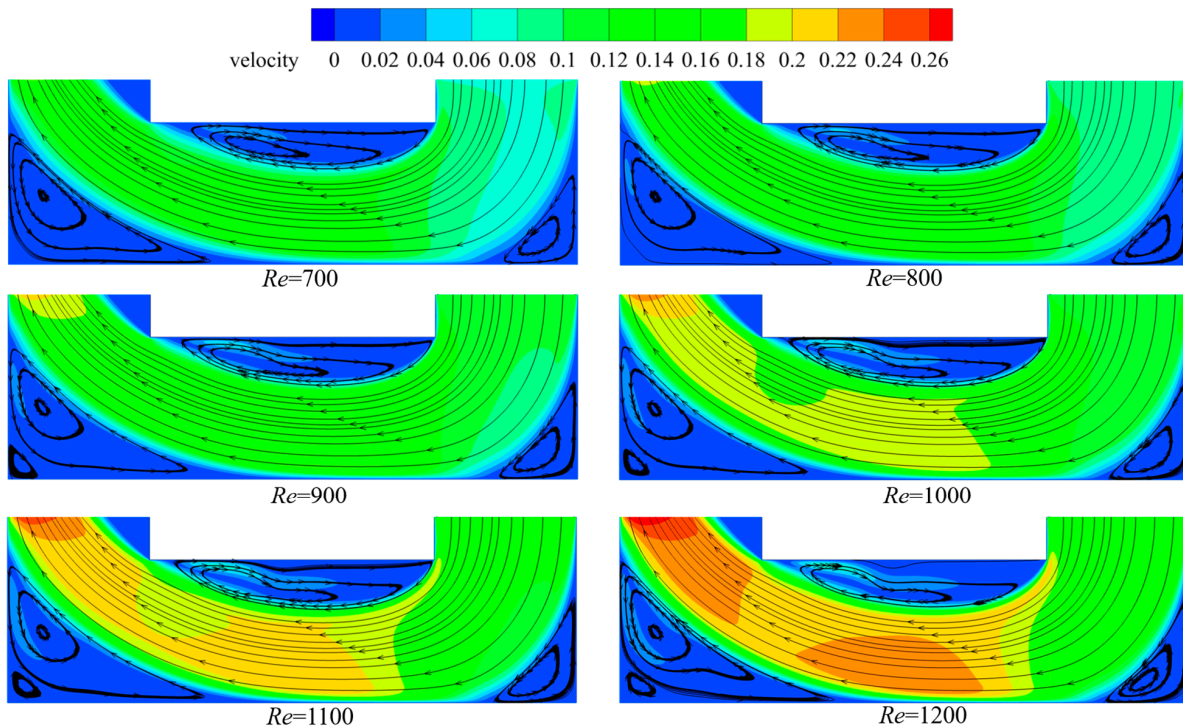


Figure 4: Velocity contours and streamlines under different Reynolds numbers for the SISO heat sink.

The corresponding temperature fields are given in Fig. 5. High temperature areas are confined to the bottom heated wall, and the main flow maintains low temperature near the inlet condition. A thermal boundary layer grows along the heating surface. Higher Re strengthens fluid heat transport, narrows the extent of high temperature zones and increases temperature gradients adjacent to the wall. Furthermore, the low speed vortex regions identified in Fig. 4 overlap with local hot spots. Limited flow mixing inside recirculation zones restricts convective cooling performance.

Fig. 6 provides a quantitative comparison in terms of Nu and f . Within $Re = 700$ – 1300 , Nu rises for every coolant as Re increases. The higher mean velocity compresses the near-wall thermal boundary layer, lowers the convective thermal resistance, and promotes energy exchange between the wall and the core flow. Consequently, the convective heat transfer rate increases throughout the tested range.

As Re increases from 700 to 1300, Nu rises by 15.04% for deionized water and by 15%, 15.91%, and 16.14% for the 0.1%, 0.3%, and 0.5% nanofluids, respectively. At any fixed Re , the ranking is 0.5% nanofluid >

0.3% nanofluid > 0.1% nanofluid > deionized water. Relative to water, the three nanofluids enhance Nu by 0.09%, 1.82%, and 3.61% at $Re = 700$; 0.10%, 1.86%, and 3.67% at $Re = 900$; 0.11%, 1.94%, and 3.85% at $Re = 1000$; and 0.06%, 2.59%, and 4.60% at $Re = 1300$. Hence, the benefit of higher nanoparticle concentrations becomes increasingly pronounced at larger Re , with the 0.5% nanofluid yielding the greatest enhancement.

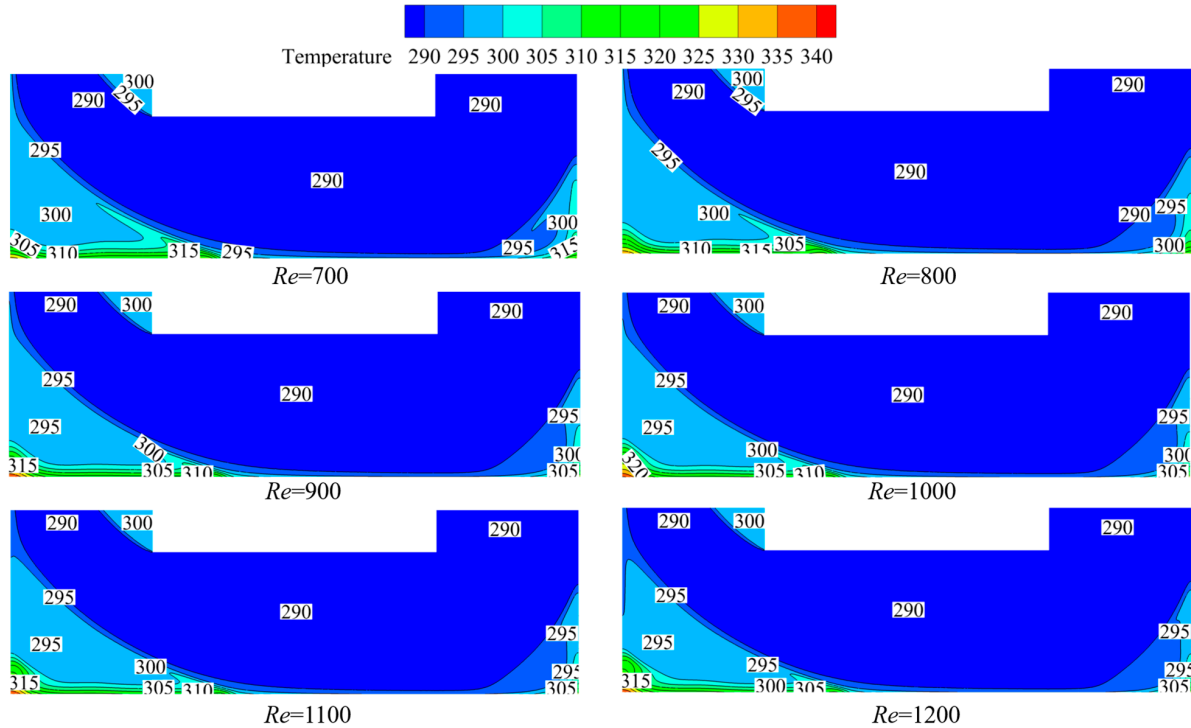


Figure 5: Temperature contours under different Reynolds numbers for the SISO heat sink.

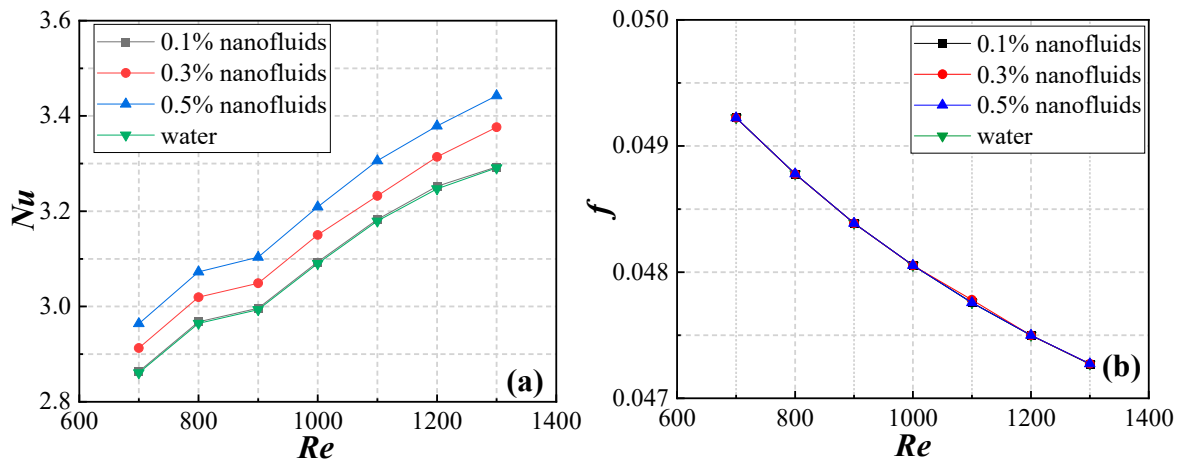


Figure 6: Changes of Nu and f with Re for different fluids in SISO. (a) Nu , (b) f .

As shown in Fig. 6b, f decreases approximately linearly as Re increases. From $Re = 700$ to 1300, the reduction is about 3.96% for all tested coolants. Over mass fractions of 0.1–0.5%, the nanofluid curves almost overlap that of deionized water. Hence, under laminar SISO operation, adding these low nanoparticle concentrations has virtually no effect on flow resistance. It is worth noting that all cases are compared at

equal Reynolds numbers. Nanofluids with higher mass fraction exhibit greater viscosity, hence a lower inlet velocity is required to maintain constant Re . As a result, pumping power cannot be simply compared directly, and the friction factor is adopted to quantify flow resistance penalty in this work.

3.2 Single Inlet and Double Outlets Heat Sink (SIDO)

Fig. 7 reveals a symmetric flow pattern in SIDO. After entering through the central inlet at the bottom, the coolant divides evenly, moves horizontally toward the two sides, follows the left and right curved passages, and turns upward to leave through the paired top outlets. The high and low speed regions and the velocity gradients are mirror images about the centerline. This balanced distribution prevents preferential flow and favors spatially uniform cooling.

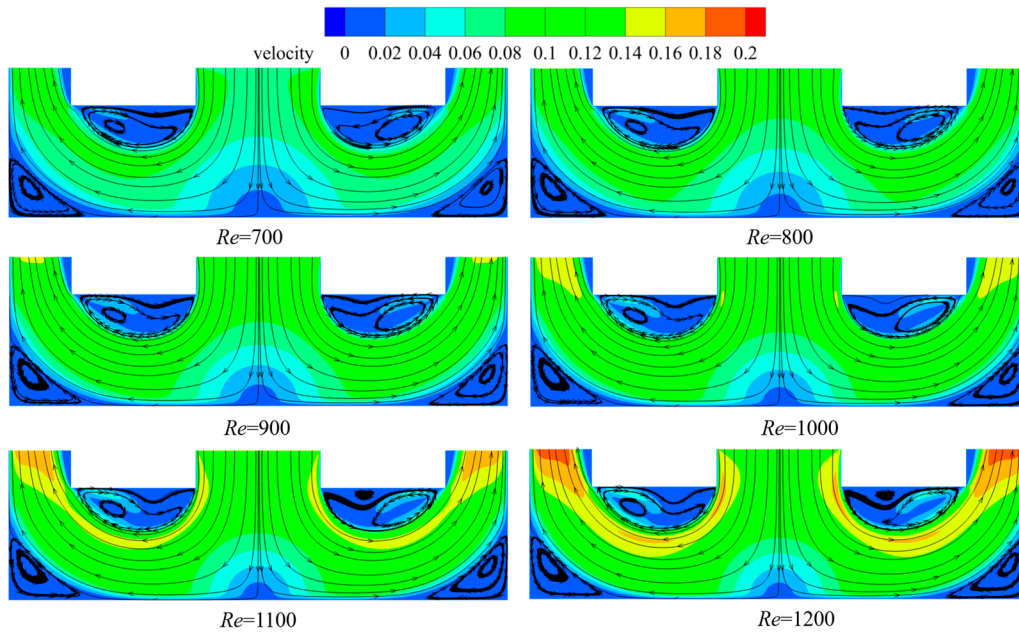


Figure 7: Velocity contours and streamlines under different Reynolds numbers for the SIDO heat sink.

A layered velocity distribution is also evident. Continuous low speed zones form along the upper and lower boundaries and the inner walls of the curved passages, consistent with the no-slip condition for laminar flow. By contrast, the channel center and the outer side of each bend carry the high speed core. Smooth transition regions connect these zones without abrupt velocity changes; viscous dissipation along the flow path therefore dominates over severe local losses. The associated temperature field exhibits a comparable spatial response (Fig. 8).

For every coolant in Fig. 9, Nu increases steadily and almost linearly over $Re = 700\text{--}1300$, with no abrupt variation. Raising Re across this interval produces Nu increases of 27.67% for deionized water and 27.69%, 27.82%, and 27.95% for the 0.1%, 0.3%, and 0.5% nanofluids, respectively.

This common trend originates from the larger channel velocity at higher Re . Stronger wall directed convection compresses the near wall velocity and thermal boundary layers, thereby reducing convective thermal resistance. Meanwhile, momentum exchange and mixing within the two outlet branches intensify, improving energy transport between the heated surface and the cooler bulk fluid.

The gap in Nu between each nanofluid and deionized water widens as Re increases. Stronger bulk convection at the higher flow rates allows the nanofluid's improved thermal transport, including the

contributions of particle Brownian motion and thermal diffusion, to yield a larger heat transfer advantage over the base fluid.

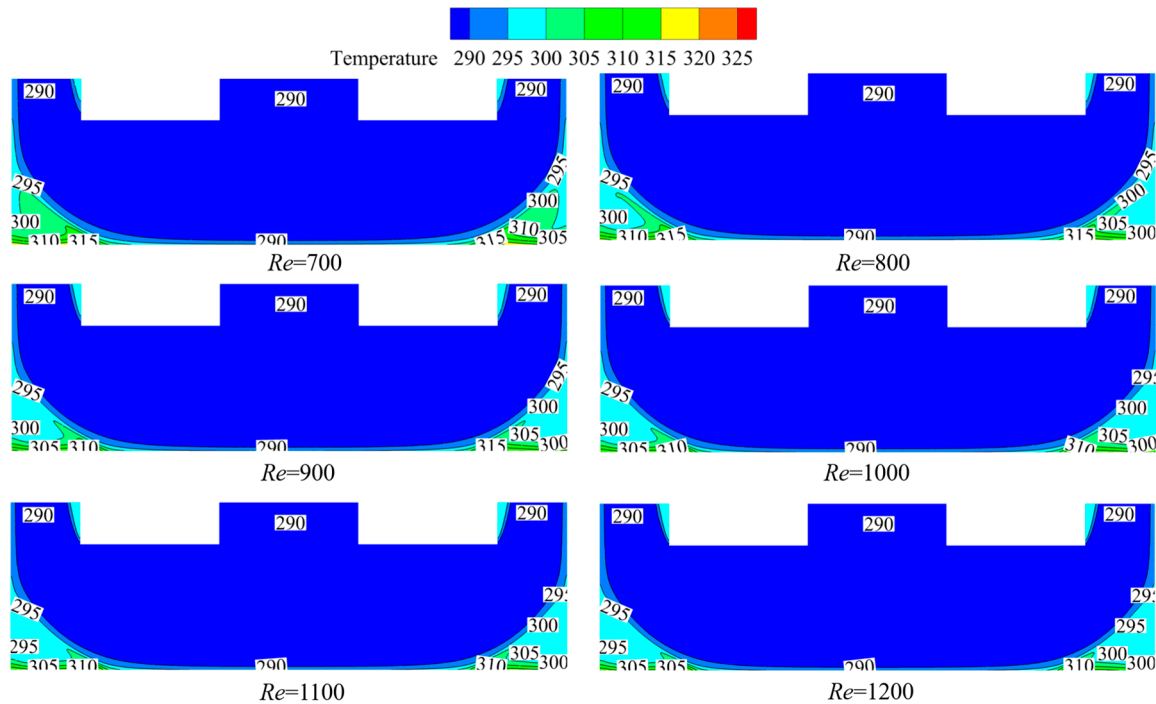


Figure 8: Temperature contours under different Reynolds numbers for the SIDO heat sink.

At a given Re , increasing the particle loading improves heat transfer monotonically: the 0.5% nanofluid performs best, followed by the 0.3% and 0.1% nanofluids and then deionized water. At $Re = 700$, the respective Nu gains over water are 0.08%, 1.78%, and 3.53%. At $Re = 1300$, the respective Nu gains over water are 0.09%, 1.89%, and 3.75%.

The f curves in Fig. 9 remain close for all coolants. As Re changes from 700 to 1300, f falls by approximately 10.12%, and the reduction between successive Reynolds numbers is nearly uniform.

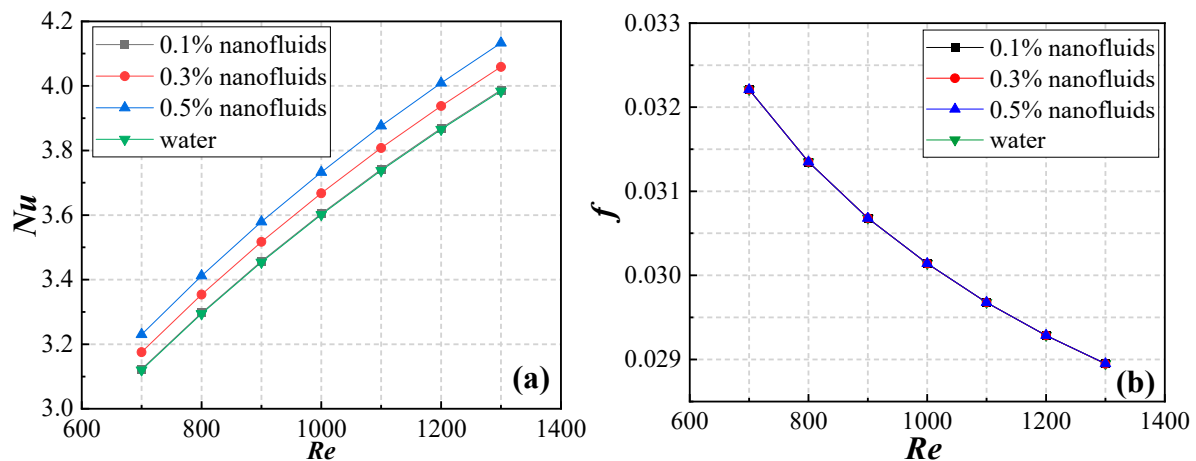


Figure 9: Changes of Nu and f with Re for different fluids in SIDO. (a) Nu , (b) f .

3.3 Bionic Wave-Shaped Single Inlet and Double Outlets Heat Sink (BSIDO)

The BSIDO velocity field in Fig. 10 forms a symmetric two branch pattern that reflects the single-inlet, double-outlet geometry. Coolant supplied at the central inlet divides along the bionic wavy walls and travels toward the outlets on both sides. A high speed core occupies the central portion of each passage and accelerates toward the outlets, whereas recirculating low velocity pockets develop inside the wave troughs. Normal to the wall, the velocity changes from zero at the surface through the boundary layer to its maximum in the core, as expected for laminar flow. The near identical flow delivered to both outlets limits maldistribution and supports symmetric heat removal. Consistent behavior appears in the temperature fields plotted in Fig. 11.

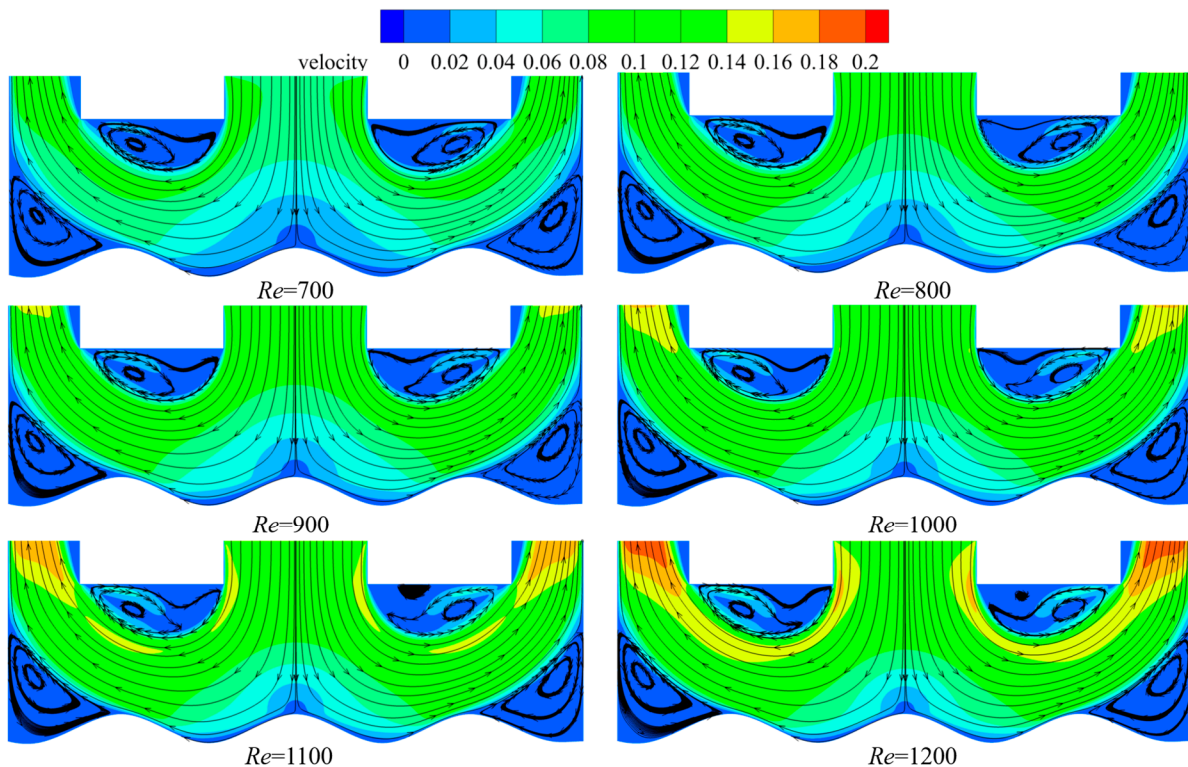


Figure 10: Velocity contours and streamlines under different Reynolds numbers for the BSIDO heat sink.

As presented in Fig. 12, Nu increases monotonically with Re for all BSIDO coolants over the laminar range $Re = 700$ – 1300 . Increasing the flow rate strengthens momentum and energy exchange, reduces the thickness of the thermal boundary layer, and steepens the fluid temperature gradient near the heated surface. These changes raise the convective heat transfer between the coolant and the heat sink wall.

For deionized water, increasing Re from 700 to 1300 raises Nu by 14.43%, directly demonstrating the favorable influence of flow rate on laminar convection.

At the same Re , each nanofluid yields a higher Nu than deionized water. Nu also rises continuously with nanoparticle mass fraction, confirming that greater particle loading improves heat transport in the heat sink over the investigated concentration range.

Under the same Reynolds number, the Nusselt number of the four test fluids shows a certain regularity. The heat transfer performance from high to low is: 0.5% nanofluids > 0.3% nanofluids > 0.1% nanofluids > deionized water. At $Re = 700$, the Nu enhancements relative to water are 0.09%, 1.84%, and 3.66% for the

0.1%, 0.3%, and 0.5% nanofluids, respectively. The corresponding values are 0.09%, 1.84%, and 3.64% at $Re = 1000$ and 0.09%, 1.76%, and 3.48% at $Re = 1300$. Although the incremental benefit per additional particle fraction diminishes slightly, the overall improvement remains clear, and the 0.5% nanofluid provides the highest convective performance under all tested Reynolds numbers.

Fig. 12 further shows that f decreases as Re rises. Between $Re = 700$ and 1300, the reduction is approximately 6.12% for water and for all three nanofluids, with an almost linear variation and no discontinuity. Throughout the 0.1–0.5% concentration range, the nanofluid friction factors remain very close to that of water.

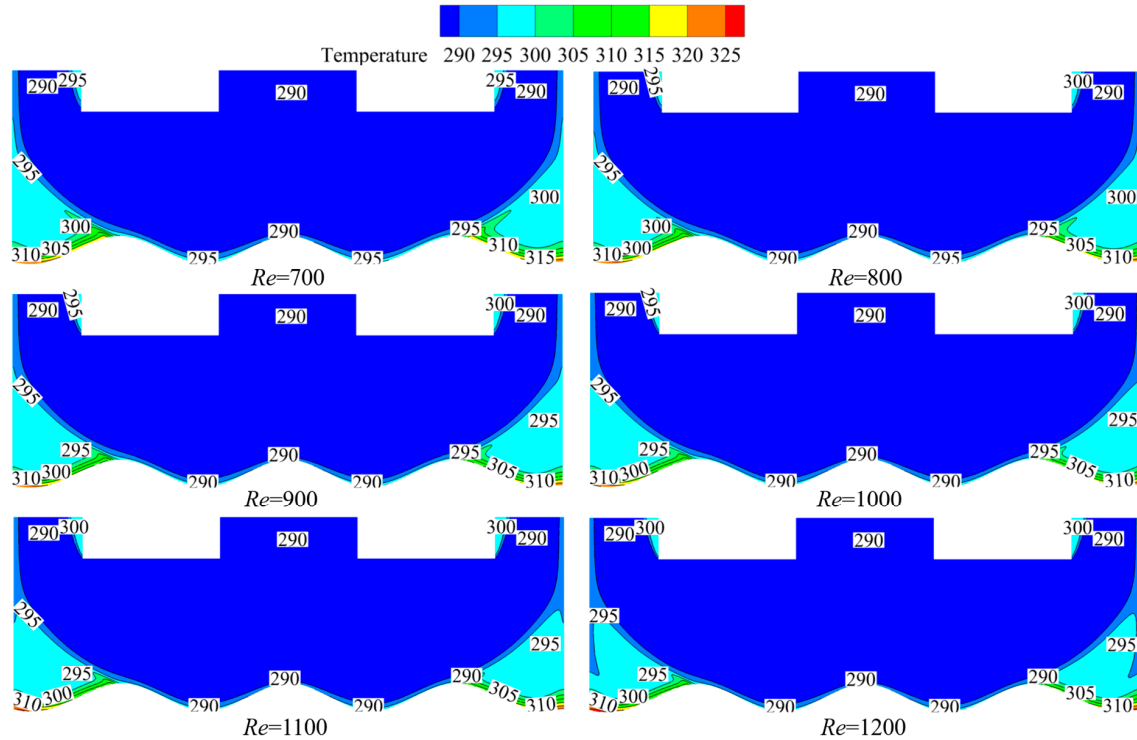


Figure 11: Temperature contours under different Reynolds numbers for the BSIDO heat sink.

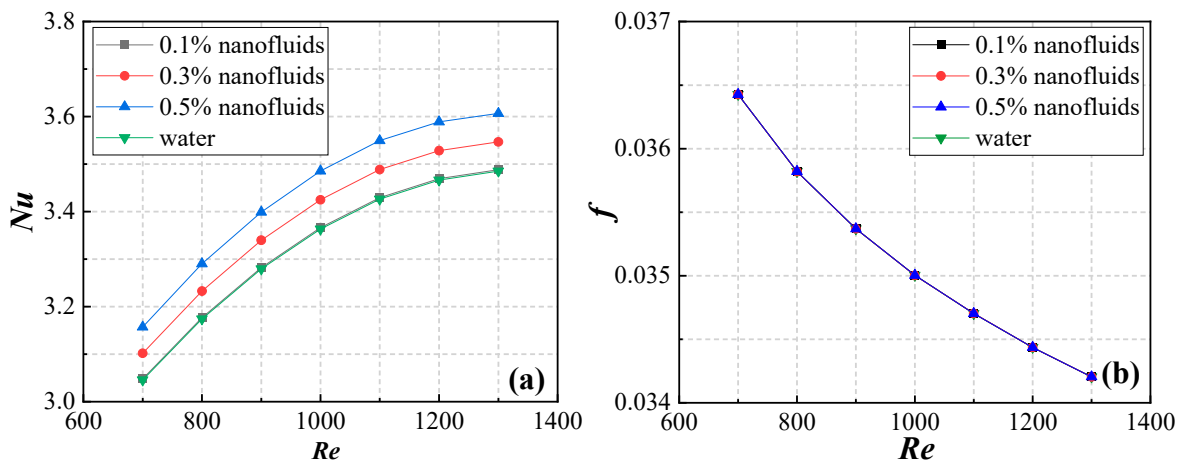


Figure 12: Changes of Nu and f with Re for different fluids in BSIDO. (a) Nu , (b) f .

It should be noted that the BSIDO heat sink has distinct manufacturability characteristics together with inevitable fabrication limitations. The BSIDO configuration, with overall dimensions of $40\text{ mm} \times 40\text{ mm} \times 9\text{ mm}$, fits standard electronic cooling modules and can be fabricated by micro-copper milling or SLM 3D printing; its symmetric single-inlet double-outlet layout adapts to universal liquid cooling pipelines for custom chip cooling. However, several manufacturing bottlenecks restrict large-scale industrial deployment: periodic wavy profiles demand ultra-precise machining with extra polishing to avoid burr-induced flow resistance growth. Accumulated dimensional errors of wave units cause uneven outlet flow and surface temperature maldistribution, requiring high-accuracy fixtures. SLM printing generates internal micropores and surface spatter that impair thermal conductivity and raise pressure drop, calling for post densification treatment. In addition, ultra precision machining and additive manufacturing increase production costs relative to straight SISO/SIDO channels, and extra double-outlet pipe joints raise long term nanofluid leakage risks. Combined solutions including milling polishing composite processing, precision clamping, hot isostatic pressing and integrated sealing structures can alleviate these fabrication defects.

3.4 Comprehensive Performance Evaluation

Different structural layouts and nanoparticle additives improve heat transfer at the cost of elevated flow resistance. To quantitatively assess the overall thermo-hydraulic characteristic, this section adopts the performance evaluation criterion (*PEC*) for comprehensive fluid and heat transfer performance comparison, as shown in Fig. 13. It can be found that *PEC* of nanofluids is all higher than 1, and *PEC* increases with nanoparticle mass fraction. 0.5% nanofluids show the highest *PEC*. For SISO, SIDO and BSIDO, *PEC* of nanofluids can reach 1.046, 1.479 and 1.261 at best. The main reasons for the above results are as follows:

- (1) Nanofluid microscale enhancement: Fe_3O_4 nanoparticles greatly raise the effective thermal conductivity of the coolant and generate random Brownian motion to break the laminar sublayer near heated walls, leading to obvious Nu growth. At mass fractions $\leq 0.5\%$, the viscosity-induced rise in friction factor remains mild and cannot counteract the heat transfer gain.
- (2) Double-outlet geometric regulation: The symmetric SIDO layout evenly distributes fluid flow to eliminate thermal stagnation zones, thinning thermal boundary layers across the heating surface without introducing extra vortex-induced pressure loss, hence achieving the optimal *PEC*. The BSIDO bionic structure further strengthens heat transfer via periodic wave vortices, yet accompanying flow separation increases flow resistance and reduces its comprehensive *PEC* relative to SIDO.
- (3) Synergistic coupling: Microscale particle disturbance and macroscale channel flow disturbance jointly amplify convective heat transfer capacity. Since heat transfer improvement always outweighs flow resistance penalty, the *PEC* value stays above unity for all geometries and nanofluid concentrations.

To benchmark the comprehensive performance of the proposed SIDO and BSIDO heat sinks, a comparative assessment is conducted against the microchannel heat sink data reported in the open literature [55]. According to previous studies, the *PEC* of the parallel-flow manifold microchannel heat sink (PFMMC) generally remains below 1.5 and can approach unity under certain operating conditions, whereas the counter-flow manifold microchannel heat sink (CFMMC) yields *PEC* values mostly exceeding 1.4 at elevated flow rates. As presented in Fig. 13, the maximum *PEC* of the SIDO configuration reaches 1.479, while that of the BSIDO design attains approximately 1.261. Although the peak *PEC* of the BSIDO is marginally lower than that of the CFMMC, it still substantially surpasses the conventional SISO layout. Notably, unlike the monotonically increasing *PEC* trends observed for the SISO and SIDO configurations, the *PEC* of the BSIDO heat sink exhibits non-monotonic behavior; it initially rises and then declines with increasing

Reynolds number, suggesting the existence of an optimal operating Re for this bionic design. Furthermore, all nanofluid cases for the BSIDO maintain $PEC > 1$ across the entire tested range, confirming that the bionic wavy topology effectively balances the trade off between heat transfer enhancement and flow resistance penalty.

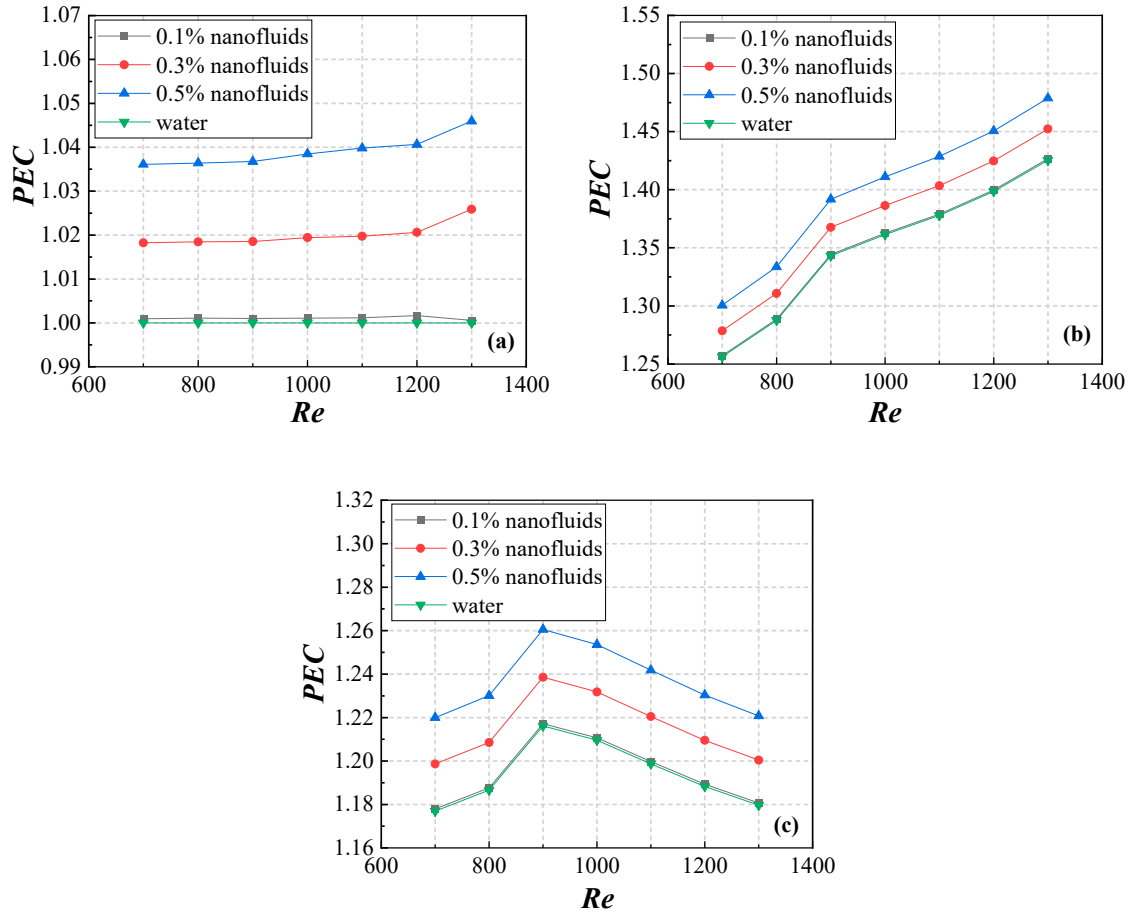


Figure 13: PEC of nanofluids in different heat sinks. (a) SISO, (b) SIDO, (c) BSIDO.

Table 4 presents a quantitative comparison of the thermo-hydraulic performance among the three microchannel configurations at $Re = 1000$, with the SISO heat sink using deionized water serving as the baseline. Both the SIDO and BSIDO designs exhibit appreciable Nusselt number enhancement alongside a marked reduction in friction factor. In the straight SISO channel, the thermal boundary layer develops continuously along the flow direction, gradually thickening and consequently degrading the wall heat transfer efficiency. In contrast, for the SIDO and BSIDO configurations, the flow-splitting layout and periodic wall undulations continuously interrupt the thermal boundary layer development and intensify local fluid mixing. Such flow modifications effectively suppress boundary layer growth across the entire channel length, thereby yielding a superior average heat transfer performance relative to the conventional SISO counterpart. Among the three layouts, the SIDO configuration delivers the most substantial overall performance improvement, with a maximum PEC enhancement of 48.74% achieved with the 0.5% nanofluid. For a fixed channel geometry, increasing the nanoparticle mass fraction further elevates both Nu and PEC , whereas the friction factor remains virtually unchanged over the investigated concentration range. Overall, the tabulated results clearly demonstrate that topological reconstruction serves as a more effective

strategy than merely adjusting nanofluid concentration for boosting the comprehensive thermo-hydraulic performance.

Table 4: Summary of performance improvements compared with the SISO heat sink (water, $Re = 1000$).

Configuration	Working Fluid	Nu Improvement (%)	Variation of Friction Factor (%)	PEC Improvement (%)
SISO	Water	-	-	-
SISO	0.10% nanofluid	0.10	0.00	0.76
SISO	0.30% nanofluid	1.94	0.00	1.94
SISO	0.50% nanofluid	4.19	0.00	3.85
SIDO	Water	16.88	-37.48	36.13
SIDO	0.10% nanofluid	16.96	-37.48	36.26
SIDO	0.30% nanofluid	18.68	-37.27	40.35
SIDO	0.50% nanofluid	20.30	-37.27	48.74
BSIDO	Water	8.80	-27.16	21.61
BSIDO	0.10% nanofluid	8.93	-27.16	21.71
BSIDO	0.30% nanofluid	11.48	-27.16	23.86
BSIDO	0.50% nanofluid	12.79	-27.16	26.06

3.5 Sensitivity Analysis

Dimensionless sensitivity coefficient S is adopted to quantify the parameter impact on Nu and PEC :

$$S = \frac{\Delta Y / Y_{\text{baseline}}}{\Delta X / X_{\text{baseline}}} \quad (15)$$

where Y stands for thermo-hydraulic performance indicators and X represents the control variable; a larger absolute value of S indicates a stronger regulating effect on cooling performance. The unified baseline condition is set as the SISO heat sink filled with pure water at $Re = 700$. Since the baseline nanoparticle mass fraction equals zero and geometric configurations are discrete categorical variables, normalized full-range perturbation and unit disturbance assumptions are adopted to eliminate mathematical singularity and enable consistent comparison across all parameters. Three core factors are examined: Reynolds number (Re), nanoparticle mass fraction (ω) and channel geometry (Geo). For the SISO pure-water case with Re varying from 700 to 1300, Nu increases from 2.86082 to 3.29111, yielding $S_{Re,Nu} = 0.1755$, while PEC remains constant at 1 for water, corresponding to $S_{Re,PEC} = 0$. At fixed $Re = 700$ on the SISO layout, the nanoparticle mass fraction rises from 0 to 0.5%, with $Nu = 2.96413$ and $PEC = 1.03611$ at the maximum concentration; the resulting coefficients are $S_{\omega,Nu} = S_{\omega,PEC} = 0.0361$, which demonstrates that low-concentration Fe_3O_4 nanofluids only produce limited heat transfer enhancement. Geometric comparison is carried out at $Re = 700$ and $\omega = 0.5\%$, taking SISO as the baseline and BSIDO as the alternative layout. The calculated coefficients are $S_{Geo,Nu} = 0.0651$ and $S_{Geo,PEC} = 0.1775$, revealing that channel remodeling greatly improves comprehensive performance and PEC is much more sensitive to geometric adjustment than Nu . All sensitivity results are summarized in Table 5.

Table 5: Sensitivity coefficients of studied parameters.

Parameter	S_{Nu}	S_{PEC}
Reynolds number (Re)	0.1755	0
Nanoparticle mass fraction (ω)	0.0361	0.0361
Geometry (SISO $\rightarrow$ BSIDO)	0.0651	0.1775

The parameter influence ranking is sorted as $Re > \text{geometry} > \omega$ for Nu , and $\text{geometry} > \omega > Re$ for PEC . From an engineering optimization perspective, raising flow velocity is the most effective way to promote heat transfer. Within the parameter range investigated in this work, modifying channel topology delivers better overall thermo-hydraulic balance than tuning nanoparticle concentration within the tested range.

4 Conclusions

This numerical study compares the flow and heat transfer behaviors of nanofluids in three different heat sink configurations. The main findings are summarized as follows:

- (1) For all three heat sinks, Nusselt number rises and friction factor drops with the increase of Reynolds number. At the same Re , higher nanoparticle mass fraction improves heat transfer. Low concentration Fe_3O_4 nanofluids barely increase flow resistance, and the 0.5% nanofluid achieves the best heat transfer performance.
- (2) SIDO heat sink shows the best overall thermo-hydraulic performance. Its symmetric flow splitting structure suppresses thermal stagnation and disrupts thermal boundary layers with minor extra pressure loss. At $Re = 1000$, SIDO with 0.5% nanofluid obtains 48.74% PEC improvement compared with baseline SISO. Though BSIDO bionic wavy structure enhances local heat transfer, wave caused flow separation brings notable flow losses, so its comprehensive performance is inferior to SIDO. PEC of BSIDO heat sink increases first and then decreases with Re , meaning an optimal Reynolds number exists. All nanofluid cases have $PEC > 1$.
- (3) Sensitivity analysis indicates that Nu is dominated in sequence by Reynolds number $>$ channel geometry $>$ nanoparticle mass fraction. For PEC , the influence order is channel geometry $>$ nanoparticle mass fraction $>$ Reynolds number. Within the parameter range investigated in this work, channel structural optimization plays a more important role in overall cooling performance than adjusting nanoparticle concentration, so structural design should be given priority in electronic-cooling practice.

Although the present comparative numerical analysis systematically revealed the thermo-hydraulic characteristics of SISO, SIDO and BSIDO heat sinks with Fe_3O_4 nanofluids, several inherent limitations remain in this work:

- (1) A 2D steady single-phase nanofluid model was used, ignoring 3D flow vortices and nanoparticle agglomeration slip phenomena.
- (2) Only a fixed wave amplitude of 2 mm was adopted, without discussing other geometric parameters of the bionic channel.
- (3) The simulation was limited to laminar $Re = 700\text{--}1300$ and constant steady heat load, excluding transient and turbulent operating conditions.
- (4) No experimental prototype tests were performed to verify the numerical predicted thermo-hydraulic performance.

In view of the above limitations, the author will conduct further research in the following aspects:

- (1) 3D two-phase flow models will be built to describe particle motion and three-dimensional mixing vortices in wavy channels.
- (2) Multi-parameter optimization of bionic wave geometry will be implemented to seek the optimal structural combination with balanced Nu and f .
- (3) Transient variable heat flux and high Re turbulent working conditions will be added to simulate practical electronic cooling environments.

- (4) 3D printed heat sink samples will be manufactured for bench scale experimental measurement to validate the numerical conclusions of this paper.
- (5) In future follow-up studies, external magnetic fields will be introduced to investigate the active heat-transfer regulation performance of magnetic Fe_3O_4 nanofluids, which lies outside the scope of the present work.

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