

Numerical Investigation of Thermal Performance of Mini-Channel Heat Sinks with Different Geometries Using Al_2O_3 -Water Nanofluid

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I. INTRODUCTION

Abstract: Rising power density in electronic and power-electronic devices has made high-heat-flux thermal management a critical design constraint, motivating continued interest in mini-channel heat sinks as a compact, high-performance cooling solution. This study presents a numerical investigation, using ANSYS Fluent, of five mini-channel heat sink geometries circular (0.8 mm and 1.0 mm diameter), rectangular (0.8 mm and 1.5 mm) and converging-diverging (0.8 mm throat) cooled by Al_2O_3 -water nanofluid at Reynolds numbers from 300 to 1800 and nanoparticle volume fractions from 0% to 0.09%, under a constant base heat flux of 50,000 W/m². Outlet temperature, maximum and minimum wall temperature, and wall temperature non-uniformity were evaluated for each configuration. Results show that increasing Reynolds number is the single most effective lever for reducing both outlet temperature (by up to 151.6 K) and wall non-uniformity (more than 3-fold), that a low nanofluid concentration of $\phi \approx 0.01\%$ consistently gives the best thermal enhancement across all geometries, and that the converging-diverging channel and the 1.5 mm rectangular channel outperform the circular channels, combining the lowest outlet temperature with the most uniform wall temperature at $\text{Re} = 300$. These findings provide practical guidance for selecting channel geometry and nanofluid concentration in compact electronic cooling applications.

Key Words: Mini-channel heat sink, Al_2O_3 -water nanofluid, Converging-diverging channel, Reynolds number, CFD, Electronics cooling

Rapid advances in electronic packaging have pushed the power density of microprocessors, laser-diode arrays and power-electronic modules to levels that conventional air cooling can no longer dissipate safely. Thermal stress arising from excessive junction temperature remains the single largest cause of electronic device failure, so the rate at which heat is removed — not merely the total energy transferred is the parameter that ultimately governs component reliability. Among the three classical modes of heat transfer, convection dominates practical thermal management: forced convection, driven by an external pump or fan, is preferred over natural or mixed convection wherever a high, controllable heat-removal rate is required, since its convective coefficient can be tuned largely independently of the surrounding environment.

The search for compact, high-flux cooling technology led Tuckerman and Pease to propose the microchannel heat sink in 1981, demonstrating that shrinking the channel hydraulic diameter substantially raises the attainable heat transfer coefficient while keeping the device small and light. This single contribution effectively founded the field, and the growth of MEMS and VLSI fabrication since has made mini- and microchannel heat sinks central to high-heat-flux compact heat exchangers, supercomputer and data-centre cooling, and high-power laser and plasma-facing component cooling. Because the term “microchannel” has been defined inconsistently across the literature, this work follows the widely used hydraulic-diameter-based classification of Mehendale et al., later refined by Kandlikar and Grande and simplified by Obot, who classifies channels of hydraulic diameter below 1 mm as microchannels the convention adopted throughout this study for the mini-channel geometries investigated.

A microchannel heat sink typically comprises many parallel channels through which a coolant is forced to remove heat from a hot base; because flow is predominantly laminar, the convective coefficient scales directly with velocity, and the high surface-area-to-volume ratio gives a small, low-inventory heat sink well suited to high-performance electronics cooling. However, the same narrow passages that raise the heat transfer coefficient also impose a steep pressure-drop penalty, so channel geometry circular, rectangular or a

flow-accelerating converging-diverging profile must be chosen to balance heat removal against pumping power rather than to minimize hydraulic diameter alone.

The coolant itself is an equally important design variable. Conventional heat transfer fluids such as water, ethylene glycol and mineral oil possess comparatively low thermal conductivity, and classical attempts to improve it by suspending millimetre- or micrometre-sized solid particles were limited by erosion, clogging and excessive pressure drop. Modern nanotechnology overcomes this limitation: Choi introduced the term “nanofluid” for a stable suspension of particles below ≈ 100 nm typically metals, oxides or carbides such as Al_2O_3 , CuO or TiO_2 whose thermal conductivity is orders of magnitude higher than that of the base liquid and whose small size avoids the fouling problems of earlier slurries. Because nanofluid thermal conductivity is a strong function of nanoparticle volume fraction, the concentration itself becomes a design parameter that must be optimised alongside channel geometry and Reynolds number.

Despite extensive individual studies of channel shape and of nanofluid concentration, comparative work that varies both together, across several geometries under otherwise identical operating conditions, remains limited. The present study therefore undertakes a numerical investigation, using ANSYS Fluent, of five mini-channel heat sink geometries circular, rectangular and converging-diverging cooled by Al_2O_3 -water nanofluid over a range of Reynolds numbers and volume fractions, with the aim of identifying the geometry-nanofluid combination that gives the best thermal performance for the smallest flow penalty.

II. LITERATURE REVIEW

Thermal management of high-heat-flux electronic devices has driven extensive research into micro- and mini-channel heat sinks since Tuckerman and Pease [1] first showed that reducing channel hydraulic diameter substantially raises the convective heat transfer coefficient. Subsequent foundational studies by Mehendale et al. [3], Kandlikar and Grande [4], and Obot [5] established flow-regime classifications and heat/mass transfer correlations for micro- and minichannel passages, while Tamayol et al. [6] characterised creeping flow through ordered minichannel arrays, together forming the analytical basis for later geometry- and coolant-focused work.

Channel geometry has been shown to strongly influence thermo-hydraulic performance. Xie et al. [7] numerically demonstrated that narrow, deep minichannels give better heat transfer than wide, shallow ones, despite a higher pressure-drop penalty, and proposed an optimised configuration balancing the two. Ghaedamini et al. [9] studied converging-diverging microchannels and found that pronounced wall curvature induces chaotic advection, raising heat transfer by up to 20% relative to straight channels,

though at increased pumping cost. Sreehari and Sharma [17] compared rectangular, U- and V-shaped serpentine microchannels and identified the U-shape as thermally superior with the lowest pressure drop, while Srivastava et al. [19] showed that adding ribs and cavities to a convergent-divergent microchannel cuts thermal resistance by up to 40% and raises heat transfer by 15–46% through induced flow recirculation.

Nanofluids have emerged as an effective route to enhance coolant thermal conductivity since Choi [2] first proposed suspending nanoparticles in base fluids. Selvakumar and Suresh [8] reported a 29.6% rise in convective coefficient using CuO /water nanofluid in a copper heat sink, while Sohel et al. [10] achieved an 18% enhancement with Al_2O_3 -water in a rectangular minichannel. Ijam et al. [11] found up to 17.3% heat-flux improvement with Al_2O_3 -water and 16.5% with TiO_2 -water, with gains diminishing at higher inlet velocity and particle loading. Arshad and Ali [14] and Saeed and Kim [15] confirmed comparable trends 12.75% enhancement with TiO_2 nanofluid at lower heating power, and up to 31.1% improvement in convective coefficient with Al_2O_3 nanofluid as fin spacing was reduced. Sajid et al. [16] extended this to wavy channels, obtaining a 40.6% Nusselt-number enhancement with dilute TiO_2 nanofluid, and Wang et al. [20] reported up to 47.5% enhancement in parallel flat minichannels at a very low 0.01% Al_2O_3 concentration, underscoring that dilute nanofluids can be highly effective without a large viscosity penalty.

Broader reviews by Dixit and Ghosh [12] and Murshed and De Castro [13] consolidate this body of work, highlighting persistent challenges in fabrication accuracy, the limited applicability of classical correlations at micro-scale, and the lack of standardised performance-prediction models across studies. More recently, attention has extended to liquid-metal and hybrid coolants: Muhammad et al. [20] modelled Galinstan-cooled minichannels capable of dissipating higher heat flux than nanofluid coolants under equivalent conditions, An et al. [22] reviewed liquid-metal micro/minichannel heat sinks and their material-compatibility challenges, and Zongyi et al. [23] compared water, nanofluid, nanoemulsion and EGaInSn , showing that the optimal channel cross-section for minimising pressure drop depends strongly on the coolant's thermophysical properties.

Very recent (2025–2026) studies extend this work toward magnetic actuation, hybrid nanofluids and data-driven design. Bhattacharyya et al. [24] numerically showed that combining ribbed mini-channels with an external magnetic field on Fe_2O_4 -water nanofluid raises the thermal enhancement factor above 2 at 2000 G, while cutting friction losses by up to 86% when the magnet sits in the developed-flow region. Siricharoenpanich et al. [25] experimentally showed that an $\text{Ag-Fe}_2\text{O}_4$ hybrid nanofluid lowers mini-channel heat-sink base temperature by up to 14% relative to water, and trained six machine-learning

models AdaBoost giving the best accuracy ($R^2 > 0.98$) to predict base temperature and Nusselt number directly from flow rate, nanoparticle type and channel geometry. Reviewing the field, Geng et al. [26] identify nanofluid long-term stability and fluid–structure interaction with advanced microchannel geometries as the key open challenges for chip-level liquid cooling, reinforcing that magnetic, hybrid and ML-assisted approaches are the emerging directions beyond single-nanofluid, single-geometry studies.

Collectively, the literature confirms that both channel geometry and nanofluid concentration significantly affect minichannel heat sink performance, but most studies examine a single geometry or a single coolant in isolation, and enhancement in heat transfer is consistently accompanied by increased pressure drop. A systematic, direct comparison of multiple channel geometries circular, rectangular and converging-diverging under identical Reynolds number and nanofluid volume-fraction conditions remains limited. This gap motivates the present numerical investigation, which uses ANSYS Fluent to identify the optimum geometry–nanofluid combination for maximum heat transfer enhancement with minimum flow penalty.

III. MATHEMATICAL MODEL

A. Physical Model and Geometry

The physical domain is a mini-channel heat sink cooled by Al_2O_3 –water nanofluid, evaluated for three channel cross-sections circular, rectangular and converging-diverging machined into an aluminium block of base 100×50 mm and height 20 mm (Fig. 1). Four parallel channels run the length of the block at equal pitch; a constant heat flux of $50,000 \text{ W/m}^2$ is applied at the bottom wall, and the coolant enters at 300 K. Key geometric parameters for each configuration are summarised in Table I.

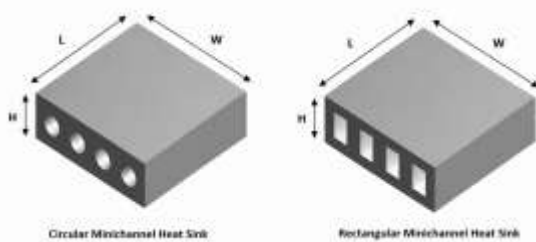


Fig. 1 Schematic of the circular and rectangular minichannel heat sink geometries (L, W, H denote block length, width and height).

TABLE I. GEOMETRIC PARAMETERS OF THE MINI-CHANNEL HEAT SINK CONFIGURATIONS

Parameter	Circular	Rectangular	Converging-Diverging
Block L × W × H (mm)	100 × 50 × 20	100 × 50 × 20	100 × 50 × 20
Block material	Aluminium	Aluminium	Aluminium
No. of channels, N	4	4	4
Channel arrangement / pitch	Parallel / 5.0 mm	Parallel / 5.0 mm	Parallel / 5.0 mm
Half-angle θ (°) /			
Throat width Wt (mm)	—	—	7 / 1.0
Converging / Diverging length (mm)	—	—	10 / 10

B. Fluid and Nanoparticle Properties

The nanoparticles are assumed spherical ($< 100 \text{ nm}$), uniformly dispersed and in thermal equilibrium with the base fluid (single-phase model). Thermophysical properties of pure water, Al_2O_3 nanoparticles and the aluminium block, evaluated at 300 K, are given in Table II.

TABLE II. THERMOPHYSICAL PROPERTIES OF WATER, Al_2O_3 NANOPARTICLES AND ALUMINIUM AT 300 K

Material	ρ (kg/m ³)	C_p (J/kg·K)	k (W/m·K)	β (1/K)
Pure water (base fluid)	998.2	4182	0.6	2.1×10^{-4}
Al_2O_3 nanoparticle	3900	850	46	1.67×10^{-5}
Aluminium (heat sink block)	2719	871	202.4	—

C. Assumptions

The flow is three-dimensional, steady and laminar; thermophysical properties are constant; the flow is incompressible; body forces are neglected; and a single-phase nanofluid model is used.

D. Governing Equations

1) Continuity and Momentum Equations

Continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \quad (1)$$

Momentum equations (x, y and z directions):

$$\left(u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z}\right) = -\frac{1}{\rho} \frac{\partial p}{\partial x} + \frac{\mu}{\rho} \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) \quad (2)$$

$$\left(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z}\right) = -\frac{1}{\rho} \frac{\partial p}{\partial y} + \frac{\mu}{\rho} \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right) \quad (3)$$

$$\left(u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z}\right) = -\frac{1}{\rho} \frac{\partial p}{\partial z} + \frac{\mu}{\rho} \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right) \quad (4)$$

2) Energy Equations

General energy equation for the coolant:

$$\rho c_{pf} \frac{DT}{Dt} = \nabla \cdot k \nabla T + \beta T \frac{Dp}{Dt} + \mu \phi \quad (5)$$

where ϕ is the viscous dissipation function:

$$\phi = 2 \left(\frac{\partial u}{\partial x} \right)^2 + 2 \left(\frac{\partial v}{\partial y} \right)^2 + 2 \left(\frac{\partial w}{\partial z} \right)^2 + \left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} \right)^2 + \left(\frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} \right)^2 + \left(\frac{\partial u}{\partial z} + \frac{\partial w}{\partial x} \right)^2 \quad (6)$$

For an incompressible fluid, the thermal-expansion and dissipation terms vanish and the energy equation reduces to:

$$\rho c_{pf} \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} \right) = k_f \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \quad (7)$$

Energy equation for the solid (heat sink) region, where k_s is thermal conductivity of the solid region:

$$k_s \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = 0 \quad (8)$$

3) Effective Properties of the Nanofluid

Effective density:

$$(\rho)_{nf} = (1 - \phi)\rho_f + \phi\rho_s \quad (9)$$

Effective heat capacitance:

$$(Cp)_{nf} = \frac{[(1-\phi)(\rho Cp)_f + \phi(\rho Cp)_s]}{\rho_{nf}} \quad (10)$$

Effective thermal-expansion coefficient:

$$(\beta)_{nf} = \frac{[(1-\phi)(\rho\beta)_f + \phi(\rho\beta)_s]}{\rho_{nf}} \quad (11)$$

Effective (stationary) thermal conductivity:

$$\frac{k_{stationary}}{k_f} = \frac{k_s + 2k_f - 2\phi(k_f - k_s)}{k_s + 2k_f + \phi(k_f - k_s)} \quad (12)$$

Effective dynamic viscosity (Brinkman correlation):

$$\mu_{nf} = \frac{\mu_f}{(1 - \phi)^{2.5}} \quad (13)$$

where ϕ is the nanoparticle volume fraction and the subscripts nf, f and s denote nanofluid, base fluid and solid (nanoparticle) properties respectively.

E. Numerical solution and Validation

The continuity, momentum, and energy equations are discretized and solved using the finite volume method. For the pressure-velocity coupling SIMPLE scheme has been used, and the energy and momentum are Solved by the consideration of second-order upwind and fast solutions, respectively. The convergence criterion for energy is considered to be on the order of 10-10, while continuity and momentum are considered to be on the order of 10-9 and 10-9, respectively.

The numerical model was validated by comparison with the numerical results of Bezaatpour [18] for a mini-channel heat sink of identical geometry, boundary

conditions and working fluid. Three coolant conditions were compared pure water, Al₂O₃-water nanofluid at $\phi = 0.03$, and the same nanofluid combined with a porous insert of porosity $\varepsilon = 0.75$ across the average convective heat transfer coefficient (h_{avg}), thermal resistance (R_{th}) and pressure drop, each plotted against coolant flow rate (Fig. 2). The present predictions track the reference data closely across the full flow-rate range (0.05–0.20 lpm) and for all three coolant conditions, correctly reproducing both the magnitude and the trend of each parameter including the substantially higher heat transfer coefficient and pressure drop, and lower thermal resistance, obtained with the porous-insert case relative to the plain-channel cases.

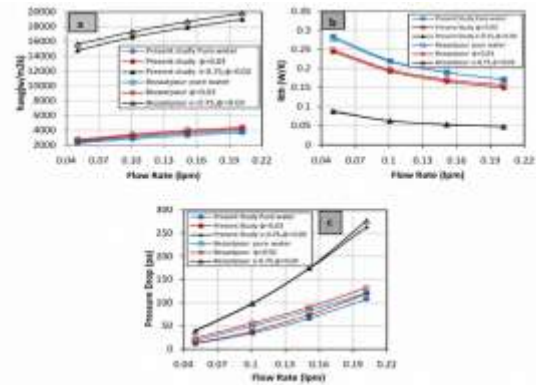


Fig. 2 Comparison between the present study and Bezaatpour [18] for (a) average heat transfer coefficient, (b) thermal resistance and (c) pressure drop.

IV. RESULTS AND DISCUSSION

CFD simulations were carried out in ANSYS Fluent for five mini-channel heat sink geometries: circular (0.8 mm, 1.0 mm), rectangular (0.8 mm, 1.5 mm) and converging-diverging (0.8 mm throat) under a constant base heat flux of 50,000 W/m² and a coolant inlet temperature of 300 K. Thermal performance was assessed in terms of outlet temperature (T_{out}), maximum and minimum wall temperature ($T_{w,max}$, $T_{w,min}$), and wall temperature non-uniformity ($\Delta T_w = T_{w,max} - T_{w,min}$), across Reynolds numbers $Re = 300$ –1800 and Al₂O₃ volume fractions $\phi = 0$ –0.09%.

A. Circular Channel ($D = 0.8$ mm): Effect of Reynolds Number and Nanofluid Concentration

At $Re = 300$ with pure water, the outlet and maximum wall temperature reached 506.41 K, with a minimum wall temperature of 416.52 K and a wall non-uniformity ΔT_w of 89.89 K. Raising Re to 1800 reduced the outlet temperature to 354.78 K a drop of 151.63 K as higher velocities thin the thermal boundary layer and raise the convective coefficient. ΔT_w fell in parallel, from 89.89 K to 28.45 K, confirming that higher Re both cools and equalises the wall temperature. Nanofluid addition shows an optimum: at $Re = 300$, $\phi = 0.01\%$ gives the lowest T_{out} (499.64 K) among the tested fractions, while $\phi \geq 0.05\%$ raises T_{out}

above the pure-water value (518.84 K and 538.85 K at $\phi = 0.05\%$ and 0.09%) as the higher effective viscosity throttles flow and offsets the conductivity gain (Fig. 3). This U-shaped trend is repeated in $T_{w,max}$ and persists, though less severely, at every Re studied.

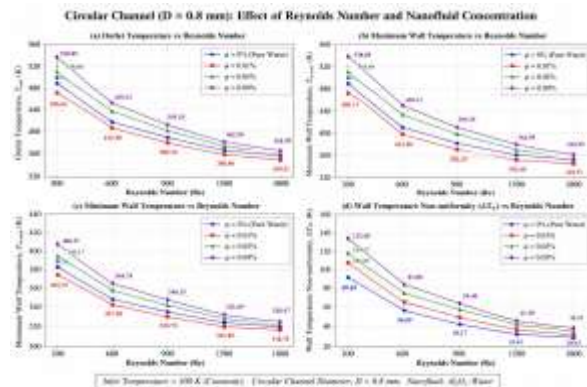


Fig. 3 Circular channel ($D = 0.8$ mm): outlet temperature, maximum/minimum wall temperature, and wall non-uniformity vs. Reynolds number at $\phi = 0\%$, 0.01% , 0.05% and 0.09% .

B. Circular Channel ($D = 1.0$ mm)

The 1.0 mm circular channel follows the same qualitative trends but at consistently lower temperatures for $Re \geq 600$. With pure water, T_{out} falls from 479.83 K at $Re = 300$ to 356.67 K at $Re = 1500$, before rising anomalously to 387.38 K at $Re = 1800$ a deviation attributed to possible flow instability or a geometric effect at the highest flow rate. At $Re \geq 1500$, all nanofluid concentrations (0.01 – 0.09%) outperform pure water, unlike at $Re = 300$ where only $\phi = 0.01\%$ (470.37 K) beats the base case; this indicates that at high Re the inertial forces dominate the viscosity penalty, widening the useful concentration range (Fig. 4).

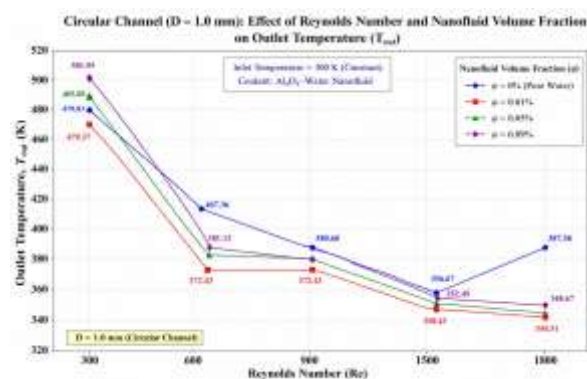


Fig. 4 Circular channel ($D = 1.0$ mm): outlet temperature vs. Reynolds number for $\phi = 0\%$, 0.01% , 0.05% and 0.09% .

C. Comparison of Circular Channel Diameters

Superimposing the two diameters (Fig. 5) shows the 1.0 mm channel outperforming the 0.8 mm channel at $Re = 300$ by 26.58 K with pure water (479.83 K vs. 506.41 K), and at $Re = 1800$ with $\phi = 0.01\%$ by a further 5.00 K (344.51 K vs. 349.51 K). The larger flow area of the 1.0 mm channel permits a higher volumetric flow rate at the same Re, so the gain in mass-flow-driven heat removal outweighs the loss in convective

coefficient from the larger hydraulic diameter except at the anomalous $Re = 1800$, pure-water point noted above.

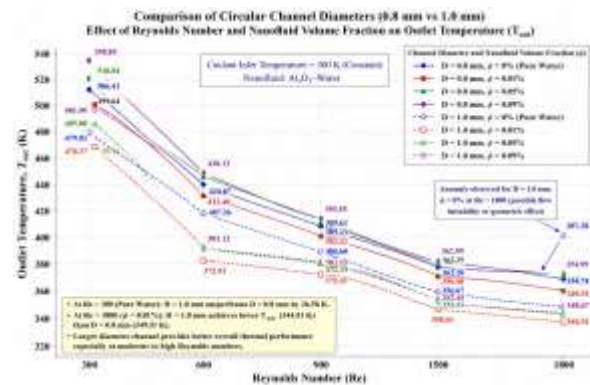


Fig. 5 Comparison of circular channel diameters (0.8 mm vs. 1.0 mm): outlet temperature vs. Reynolds number and nanofluid volume fraction.

D. Rectangular Channel: 0.8 mm vs. 1.5 mm ($Re = 300$)

The 1.5 mm rectangular channel is markedly cooler than the 0.8 mm channel at every nanofluid concentration, by 58–71 K, again reflecting its larger cross-sectional flow area and correspondingly higher mass flow rate at equal Re. Both channels exhibit the same non-monotonic response to ϕ , cooling to a minimum before rising again at higher loadings $\phi = 0.01\%$ for the 0.8 mm channel (463.32 K) and $\phi \approx 0.03\%$ for the 1.5 mm channel (403.50 K), after which viscosity-driven flow resistance dominates (Fig. 6).

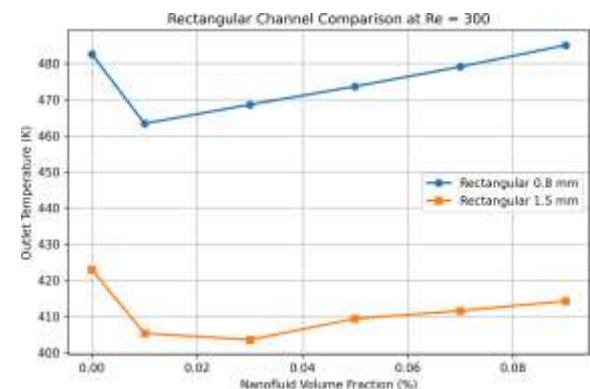


Fig. 6 Rectangular channel outlet temperature vs. nanofluid volume fraction at $Re = 300$: 0.8 mm vs. 1.5 mm.

E. Converging-Diverging Channel (0.8 mm Throat, $Re = 300$)

The converging channel shows outlet temperature bottoming out at $\phi = 0.01\%$ (405.07 K, a 9.17 K reduction from the 414.24 K pure-water case) before rising again at higher ϕ , mirroring the other geometries. Minimum wall temperature falls sharply from 365.99 K ($\phi = 0\%$) to about 347 K by $\phi = 0.01\%$ and then stays essentially flat, while wall non-uniformity ΔT_w instead rises steadily with ϕ , from 48.25 K to 72.88 K indicating that although higher nanofluid loading does not much change the coolest wall point, it increasingly

widens the spread between the hottest and coolest wall locations (Fig. 7).

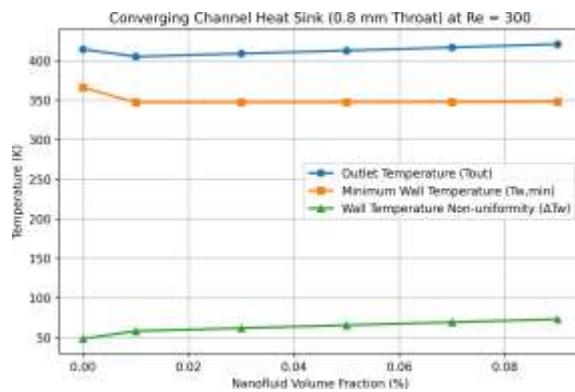


Fig. 7 Converting-diverging channel (0.8 mm throat): outlet temperature, minimum wall temperature and wall non-uniformity vs. nanofluid volume fraction ($Re = 300$).

F. Comparative Analysis of All Channel Geometries ($Re = 300$)

Across the full ϕ range at $Re = 300$ (Fig. 8), the ranking of outlet temperature is consistent at every concentration: circular 0.8 mm (highest) > circular 1.0 mm $\approx$ rectangular 0.8 mm > rectangular 1.5 mm $\approx$ converging 0.8 mm (lowest pair, within a few kelvin of each other). All five geometries share the same U-shaped nanofluid response, bottoming out near $\phi = 0.01$ – 0.03% before rising toward $\phi = 0.09\%$, confirming that the optimum nanofluid concentration is essentially geometry-independent even though its absolute benefit is not.

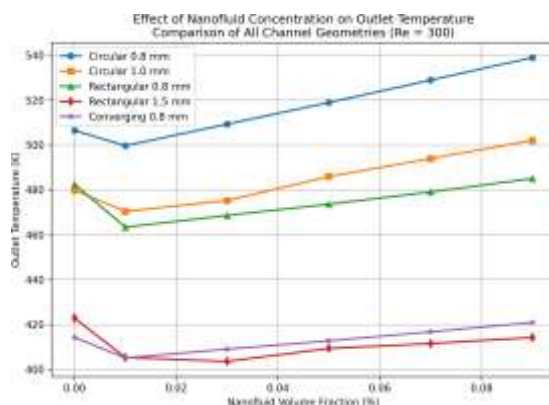


Fig. 8 Effect of nanofluid concentration on outlet temperature for all five channel geometries ($Re = 300$).

With pure water ($\phi = 0\%$) as the baseline (Fig. 9), the converging-diverging channel achieves the lowest outlet and maximum wall temperature (414.24 K) among all designs, followed by the rectangular 1.5 mm channel (422.85 K), circular 1.0 mm (479.83 K), rectangular 0.8 mm (482.50 K), and circular 0.8 mm (506.41 K, highest). Wall non-uniformity follows a related but distinct order: it is lowest for the rectangular 1.5 mm channel (43.68 K) and the converging channel (48.25 K), and highest for the circular 0.8 mm channel (89.89 K). This confirms that channels promoting flow acceleration or larger cross-sections the converging

throat and the 1.5 mm rectangular duct deliver both the coolest and the most thermally uniform heat sink base, which is the more favourable outcome for electronic-cooling reliability.

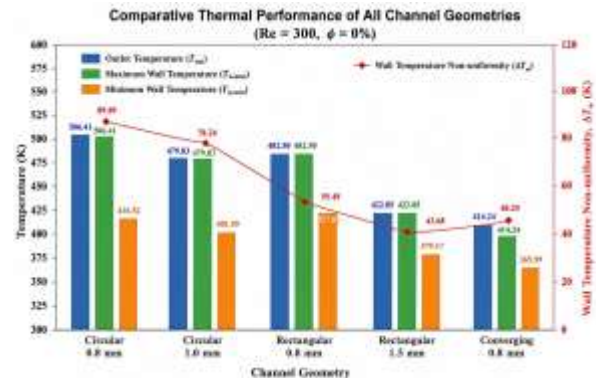


Fig. 9 Comparative thermal performance of all five channel geometries at $Re = 300$, $\phi = 0\%$: outlet, maximum and minimum wall temperature, and wall non-uniformity.

TABLE III. THERMAL PERFORMANCE OF ALL CHANNEL GEOMETRIES AT $Re = 300$ AND $\phi = 0\%$

Geometry	T_{out} (K)	$T_{w,max}$ (K)	$T_{w,min}$ (K)	ΔT_w (K)
Circular 0.8 mm	506.41	506.41	416.52	89.89
Circular 1.0 mm	479.83	479.83	401.59	78.24
Rectangular 0.8 mm	482.50	482.50	427.01	55.49
Rectangular 1.5 mm	422.85	422.85	379.17	43.68
Converging-Diverging 0.8 mm	414.24	414.24	365.99	48.25

V. CONCLUSION

A numerical investigation of five mini-channel heat sink geometries circular (0.8 mm, 1.0 mm), rectangular (0.8 mm, 1.5 mm) and converging-diverging (0.8 mm throat) cooled by Al_2O_3 -water nanofluid was carried out over $Re = 300$ – 1800 and $\phi = 0$ – 0.09% under a constant $50,000 \text{ W/m}^2$ base heat flux. The main conclusions are:

- Reynolds number is the dominant design variable: raising Re from 300 to 1800 cut outlet temperature by up to 151.6 K and shrank wall temperature non-uniformity more than 3-fold in the circular 0.8 mm channel, with similar trends in every other geometry.
- A dilute Al_2O_3 nanofluid concentration of $\phi \approx 0.01\%$ consistently gave the best thermal performance across all five geometries; beyond this, rising effective viscosity offset the conductivity gain and outlet temperature increased again, so nanofluid loading should be treated as an optimisation variable rather than maximised.
- Channel geometry has a first-order effect independent of nanofluid concentration: the converging-diverging channel and the 1.5 mm rectangular channel consistently

outperformed both circular channels, combining the lowest outlet temperature with the most uniform wall temperature at $Re = 300$.

- Increasing circular channel diameter from 0.8 mm to 1.0 mm improved performance at every Re tested (with one anomaly at $Re = 1800$, pure water), showing that a larger flow area and hence higher mass flow rate at fixed Re can outweigh the loss in convective coefficient from a larger hydraulic diameter.
- For compact electronic cooling applications, the converging-diverging channel with Al_2O_3 -water nanofluid at $\phi \approx 0.01\%$ is recommended as the most effective combination identified in this study, offering the best balance of low base temperature and thermal uniformity.

Future work should extend this comparison to a wider range of nanoparticle materials and hybrid nanofluids, include pressure-drop and pumping-power optimisation alongside thermal performance, and validate the numerical predictions against experimental data for the converging-diverging geometry.

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