

RESEARCH ARTICLE

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CFD-BASED NUMERICAL ASSESSMENT OF PIN-FIN HEIGHT OPTIMIZATION FOR ENHANCED THERMAL PERFORMANCE IN MWCNT NANOFLUID MICROCHANNEL HEAT SINKS

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ABSTRACT

Microchannel heat sinks are often redesigned with taller pin fins under the assumption that larger surface area inherently yields better cooling. However, this study reveals a counterintuitive yet significant finding that the shortest pin fin (6.2 μm) consistently outperforms all taller configurations across the full range of operating conditions. A three-dimensional CFD model developed in ANSYS Fluent evaluates a plain channel and four pin-fin heights (6.2, 18.6, 31.0, and 43.4 μm) over Reynolds numbers 55 - 225, using water and MWCNT nanofluids at 1, 2 & 3% concentrations. The thermo-hydraulic behavior is assessed through Nusselt number, friction factor, pressure drop, and thermal performance factor (TPF). Although taller fins enhance heat transfer, they impose substantial pressure and pumping penalties, diminishing overall system efficiency. In contrast, the 6.2 μm geometry demonstrates a TPF consistently above unity with a strong increase at higher Reynolds numbers, indicating an efficient flow-mixing mechanism achieved with minimal hydraulic cost. In microscale cooling, shorter and compact pin fins rather than taller ones unlock superior thermal performance. This unexpected dominance of the smallest configuration highlights an important design insight. A supporting data-driven heat-map visualization is also employed to interpret multi-parameter trends, reinforcing the identification of the 6.2 μm pin height with 3% MWCNT as the most efficient case. Overall, the study provides clear guidance for next-generation microchannel heat-sink optimization and challenges conventional assumptions about fin geometry in miniaturized cooling systems.



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I. INTRODUCTION

The need for compact and high-performance thermal management systems has increased due to the rapid miniaturisation of electronic devices, the emergence of high-density microprocessors, and developments in power electronics. In today's increasingly compact and powerful devices, effective thermal solutions are essential for maintaining optimal performance, preventing overheating, and ensuring device reliability. Because of their high surface area-to-volume ratios and superior convective heat transfer capabilities, microchannel heat sinks (MCHSs) have become a viable option for dissipating high heat fluxes. These devices are perfect for cooling next-generation electronics because they use laminar flow in narrow channels to achieve superior thermal performance while minimising coolant volume. The groundbreaking work by Tuckerman and Pease (1981) [1] introduced the first microchannel heat sink for high performance cooling planar integrated VLSI circuits. It showed that the convective heat transfer coefficient h between the substrate and coolant is the main limitation for thermal resistance, and laminar flow in narrow channels increases h .

Experiments at a power density of 790 W/cm² showed a maximum substrate temperature rise of 71 °C, validating theoretical predictions and demonstrating feasibility for ultrahigh-speed VLSI circuits. Weisberg et al. (1992) [2] further analyzed ultra compact microchannels proposing design algorithms to optimize dimensions for integrated cooling applications by highlighting the potential of precision machined channels for effective thermal management. Subsequent studies explored innovative channel geometries, such as wavy, fractal, and helical microchannels, to induce secondary flow, enhance turbulence, and increase convective heat transfer without significantly penalizing pressure drop. Hussein, H. A. (2024) [3] numerically analyzed six circular mini-channel heat sink designs with L-shaped flow passages to enhance coolant distribution and heat transfer. The L-channel (JMLCCHS) showed higher performance index and lower thermal resistance than rectangular channels. Haridas, D., (2014) [4] analyzed the behaviour of the Temperature fields and forced convection in compact thermal systems have been studied using digital interferometric techniques, such as the Michelson and Mach-Zehnder methods. Zhu, Q., (2024) [5] numerically studied a Symmetric wavy microchannel with wave trough-to-crest secondary channels (SW-TC) with slanted secondary channels for improved flow uniformity and heat transfer. The SW-TC design demonstrated better thermal efficiency than the Symmetric wavy microchannel with wave crest-to-crest secondary channels (SW-CC) and Symmetric wavy microchannel with wave trough-to-trough secondary channels (SW-TT) designs.

Subsequent studies have focused on improving heat transfer by incorporating microstructures and pin-fin arrangements. A and Chakraborty et al. (2020) [6] enhanced MCHS performance by integrating microstructures like pillars and blind holes. Pillars boosted heat transfer but increased pressure drop, while blind holes moderately improved both. A hybrid design with more pillars near the inlet achieved higher heat transfer with controlled pumping power. Xie et al. (2021) [7] numerically analyzed laminar flow ($Re = 10-300$) in microchannel heat sinks with pin fins at 0°, 30°, 45°, and 60° inclinations using the finite volume method. Strong secondary flow was created by the in-lined 30° configuration which improved heat transfer but increased friction. While pattern effects were negligible, the inclined angle had a significant influence on thermo-hydraulic performance. Optimal designs were recommended under equal pumping power conditions. Similarly, Ali et al. (2024) [8] numerically studied semi-elliptical pin fins ($\Psi = 0.36-1.0$) in water-cooled MCHSs ($Re = 100-900$, heat flux = 1×10^6 W/m²). The best performance (THPP = 1.41) occurred at $\Psi = 1.0$ and $Re = 300$, while the lowest was at $\Psi = 0.36$ and $Re = 900$. Recent work of Qi, X., et al.(2025) [9] is on thermally induced errors in feed-drive mechanisms has shown that advanced cooling technologies, such as sintered core heat pipes, play a crucial role in minimizing temperature gradients. This reinforces the broader engineering demand for compact and high-performance cooling systems like microchannel heat sinks.

Zhu, X. et al., (2025) [10] found that Geometric refinement and better heat transfer interfaces can significantly increase system efficiency as demonstrated by recent optimisation studies on stepped-configuration thermoelectric generators. These findings parallel the motivation of the present work where pin-fin geometry and nanofluid properties are tailored to elevate thermal performance in microchannel heat sinks. Zhu, X. et al., (2025) [11] Geometrical and material optimization of advanced TE modules, like hybrid configurations of N-SKD/P-HH, demonstrated that thermal efficiency increases while reiterating geometry optimization; hence, the pin-fin height optimization of microchannel heat sinks in this study actually follows suit. Ali et al. (2024) [12] numerically analyzed V-shaped rib angles (35°-90°) in MCHSs for $Re = 100-900$. Smaller angles (35°) enhanced heat transfer via stronger flow mixing and yielding higher Nusselt numbers but greater frictional losses. The study highlights the need to optimize rib angles for balanced thermal hydraulic performance. More recent work by Singh et al. (2025) [13] examined different micro pin-fin geometries, revealing that circular fins provide the highest thermal enhancement while triangular and square fins offer moderate improvement which influence of fin geometry and arrangement on thermo-hydraulic performance. Additionally, Polat et al. (2022) [14] highlighted that micro pin-fin shapes (diamond, square, circular) and their arrangements strongly affect both Nusselt number and pressure drop, requiring optimized design to balance heat transfer and pumping power.

Parallel to geometric optimizations, nanofluids have been extensively explored for enhancing thermal performance in MCHSs. Maheshwary et al. (2017) [15] experimentally showed that TiO₂-water nanofluid thermal conductivity increases with higher concentration, smaller particle size, and cubic shape, with concentration having the greatest effect (69.43%). However, higher concentrations also increase viscosity and pumping power. Experimental studies by Jeong et al. (2013) [16] showed that both viscosity and thermal conductivity of ZnO-water nanofluids increase with higher nanoparticle concentration. Particle shape is important at 5 vol.%, nearly rectangular particles produce higher enhancements than spherical ones (about 7.7% higher viscosity and 6% higher thermal conductivity). This suggests that the geometry of nanoparticles has a significant impact on the performance of nanofluids. Hung et al. (2012) [17] demonstrated that Al₂O₃-water and diamond-water nanofluids enhance microchannel heat transfer by over 20%, with optimal performance achieved using smaller particles and controlled volume fractions. Hybrid nanofluids combining multiple nanoparticles have shown even greater potential, Bhattad et al. (2020, 2022) [18] reported improved thermal conductivity and performance indices with minimal pumping penalty, while Murali Krishna et al. (2020) [19] confirmed that MWCNT-CuO/water hybrids outperform mono nanofluids in circular microchannels.

The integration of numerical simulations and artificial intelligence has become crucial for optimizing MCHS designs. Gao et al. (2022) [20] reviewed the application of CFD, LBM, and molecular dynamics in microchannel heat transfer emphasizing that CFD remains versatile for laminar flows while machine learning techniques as used by Said et al. (2022) [21] demonstrated that MWCNT/water nanofluids notably enhance shell-and-tube heat exchanger (STHX) performance, achieving a 3.23% rise in thermal conductivity, 31.08% higher heat transfer coefficient, and 5.49% improved efficiency at 0.3% volume fraction. CFD analysis with semicircular baffles further increased efficacy by 15.4%, confirming the nanofluid's superior heat transfer capability and sustainability for STHX applications. Yadav et al. (2021) [22] studied MWCNT/water nanofluids showing non-Newtonian behavior dependent on concentration, temperature, and shear stress, with SDS surfactant influencing rheology. The power-law model described flow behavior, while an ANN model achieved high prediction accuracy, demonstrating the strength of combining experimental, theoretical, and AI approaches for nanofluid rheology. Additionally, innovative channel geometries, such as fractal or plus-shaped designs, have been explored numerically Lyu et al. (2020) [23] demonstrated that fractal channels with CNT nanofluids achieve uniform temperature distribution and enhanced heat transfer, further highlighted the stability and efficiency of MWCNT-based nanofluids, achieving significant heat transfer enhancement with negligible pumping penalties.

While Manjur et al. (2025) [24] showed that novel geometries coupled with advanced nanofluids can significantly improve thermal performance and reduce pressure drop. Cai, B., (2025) [25] studied double-layered MCHSs, introducing TMC-ORTC with rib-cavity coupling, which enhanced mixing and heat transfer while reducing pressure loss, achieving 120% higher Nusselt number and 81.7% better temperature uniformity than conventional designs. Beyond MCHSs, the broader application of nanofluid-enhanced systems has been confirmed in photovoltaic thermal (PVT) systems, where Rajamony et al. (2024) [26] reported that MWCNT-enhanced phase change materials improve both thermal and electrical outputs. Furthermore, studies on laminar flows with nanofluids in non-conventional geometries, such as Analytical and statistical of Yahiaoui et al. (2025) [27] analysed Jeffery-Hamel flow for Cu–water and SiO₂/water nanofluids in rotating channels showed that rotation induces asymmetric shear flow and affects heat transfer. Though geometrically different, the study relates to pin-fin microchannels by revealing how nanofluid properties and Reynolds number influence flow, wall shear, and heat transfer. Babar Hussain Shah et al. (2024) [28] provide insights into how boundary conditions, Reynolds number, and fluid properties influence flow and heat transfer, which is relevant for optimizing MCHS performance. Emadi, A., (2025) [29] numerically assessed that multi-layer porous fins significantly enhance heat transfer by improving convection and temperature uniformity, with two-layer fins offering optimal performance under high porosity and permeability. Despite significant progress, research gaps remain in balancing heat transfer enhancement with pumping power penalties, exploring hybrid nanofluids with varying proportions, and applying machine learning for predictive optimization under diverse flow conditions. The present study aims to address these gaps by numerically investigating the thermohydraulic performance of MCHSs with MWCNT-based nanofluids across various pin-fin geometries and configurations, thereby contributing to the development of efficient microchannel cooling strategies for advanced electronic and thermal management applications.

II. COMPUTATIONAL MODELLING

II.1 COMPUTATIONAL DESIGNING

II.1.1 Design and Computational Model

The computational domain for the microchannel heat sink (MCHS) was created in ANSYS Workbench (Design Modeler) to analyze fluid flow and heat transfer across different pin fin geometries and nanofluid concentrations. The model simulates a three-dimensional, steady-state, laminar flow to capture the interaction between the solid microchannel and the working fluid [27]. Copper was chosen as the channel material for its high thermal conductivity and mechanical stability, making it suitable for microscale thermal management. The Microchannel design is shown in the Figure 1b as Computational Domain and the Schematic view of the channel is given in the Figure 1a. The dimensions of the microchannel as given in Table 1 has a length of 600 μm , a width of 265 μm , and a height of 62 μm , with cylindrical pin fins of 15 μm diameter distributed at a uniform pitch of 50 μm (Fig 1a, b). Each configuration includes 55 pin fins of varying heights 6.2 μm , 18.6 μm , 31 μm , and 43.4 μm to evaluate the influence of fin aspect ratio on the overall flow and heat transfer behavior. Water and MWCNT-water nanofluids with concentrations of 1%, 2%, and 3% by volume were used as the working fluids. These nanofluids were chosen for their enhanced thermal conductivity and capacity to intensify near-wall thermal interactions. The simulations were performed for inlet velocities of 0.5, 1.0, 1.5, and 2.0 m/s to capture the combined effect of flow rate and nanofluid loading on heat transfer and pressure characteristics. The fluid enters the channel through the inlet boundary with a uniform velocity profile and exits through an outflow boundary with zero gauge pressure. Constant heat flux is applied at the bottom surface, while all other external walls are adiabatic. A structured mesh was generated to accurately capture the boundary layer and the complex flow around pin fins, with grid independence verified through successive mesh refinements. The governing equations for mass, momentum, and energy were solved using the finite volume approach in ANSYS Fluent, ensuring numerical stability and convergence for all cases studied. This computational framework enables a comprehensive understanding of the microscale flow development, temperature distribution, and fluid-solid interaction under various geometric and operating conditions, providing a reliable basis for optimizing MCHS performance using MWCNT nanofluids.

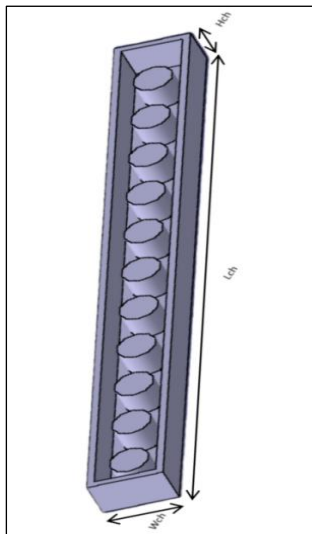


Figure 1 a: Schematic Diagram of the Channel.
Source: Authors, (2026).

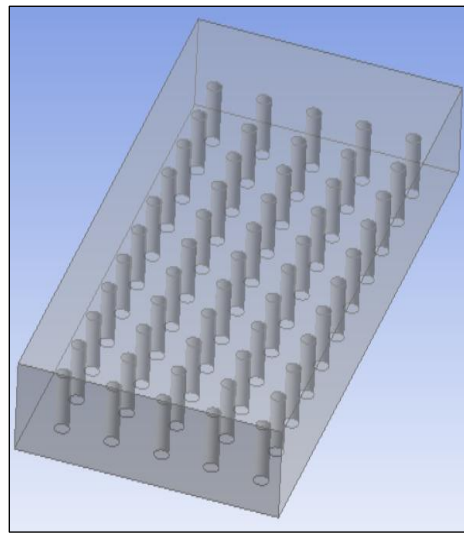


Figure 1 b: Computational Domain.
Source: Authors, (2026).

Table 1: Channel Descriptions.

Parameter	Values
Length of Channel, Lch	600 μm
Width of Channel, Wch	265 μm
Height of Channel, Hch	62 μm
Hydraulic Diameter	100.5 μm
Pin Diameter	15 μm
Pin Spacing	50 μm
Number of Pins	55
Pin fin Height	6.2, 18.6, 31.0, 43.4 μm
Nanofluid	MWCNT
Nanofluid Concentration	1%, 2%, 3%

Source: Authors, (2026).

II.2.2 Properties of Heat Transfer Fluid and Microchannel

The performance and thermal efficiency of the microchannel heat sink (MCHS) are highly dependent on the thermophysical properties of both the working fluid and the substrate material. These properties regulate the mechanisms of heat transfer, flow resistance, and overall cooling effectiveness under varying operating conditions. In the present study, deionized water and multi-walled carbon nanotube (MWCNT)/water nanofluids with volume concentrations of 1%, 2%, and 3% are employed as the heat transfer media while aluminium is used as the microchannel substrate. Water acts as the base fluid due to its favorable properties, including high specific heat capacity, moderate thermal conductivity, and low viscosity, which collectively promote efficient convective heat removal. The suspension of MWCNT nanoparticles within the base fluid further enhances its effective thermal conductivity and heat transport capability through mechanisms such as Brownian motion and micro-scale convection. The incorporation of MWCNTs also improves energy diffusion and temperature uniformity within the microchannel, leading to superior thermal performance compared to pure water. The thermophysical properties of the working fluids at a reference temperature are summarized in Table 2. Aluminium is selected as the microchannel substrate material because of its high thermal conductivity, low density, and excellent mechanical stability. These properties make it ideal for rapid heat spreading and lightweight thermal management applications. The high thermal conductivity of aluminium (386 W/m·K) ensures efficient heat distribution along the channel walls, while its moderate specific heat capacity allows it to accommodate transient temperature fluctuations. Furthermore, its structural robustness and resistance to thermal fatigue make it suitable for repeated heating and cooling cycles under high heat flux conditions. The thermophysical properties of aluminium are also listed in Table 2. The combined use of MWCNT–water nanofluids and Copper substrate provides a highly effective heat transfer configuration. The enhanced thermal conductivity of the nanofluid ensures rapid heat removal from the heated surface, while the superior heat spreading capability of Copper maintains a uniform temperature gradient across the microchannel. Together, they contribute to reduced thermal resistance, improved convective performance, and sustained operational efficiency of the MCHS under diverse flow and thermal loading conditions.

Table 2: Properties of the Materials.

Thermo-Physical Properties	Units	Water	MWCNT	Copper
Density	(kg/m ³)	997	1600	2700
Dynamic Viscosity	(kg/m·s)	0.000891	-	-
Thermal Conductivity	(W/m·K)	0.6	3000	386
Specific Capacitance	(J/kg·K)	4182	796	903

Source: Authors, (2026).

II. 2.3 Mesh Independence Study:

The grid independence test is conducted to ensure that the solution is unaffected by further mesh refinement, confirming numerical accuracy. A detailed mesh independence study was performed for the rectangular microchannel under a constant heat flux of 10,000 W/m² and an inlet velocity of 0.5 m/s. Three mesh densities were examined: coarse (247,886 elements), medium (707,788 elements), and fine (1,584,540 elements). The fine mesh produced a Nusselt number (Nu) of 4.347 and a friction factor (f) of 0.83, while the medium mesh yielded Nu = 4.507 and f = 0.8115. The coarse mesh slightly overpredicted Nu (4.437) and underpredicted f (0.81), consistent with the typical behavior of under-resolved meshes that inadequately capture near-wall gradients. The relative percentage variations between the fine and medium meshes were 3.68% for Nusselt number and 2.28% for friction factor—both within the generally accepted convergence tolerance of $\pm 5\%$. This confirms that further refinement produces negligible improvement in the predicted flow or thermal characteristics. Notably, although the fine mesh results in a marginally lower Nusselt number, this outcome reflects physical accuracy rather than numerical limitation. Coarser grids often overpredict heat-transfer coefficients due to exaggerated wall-temperature gradients caused by insufficient near-wall resolution. The fine mesh, with its higher element density and smaller y^+ values, resolves the viscous and thermal boundary layers more precisely, capturing the realistic conduction-convection interplay and producing a more reliable thermal prediction. The modest increase in pressure drop in the fine mesh ($\approx 2\text{--}3\%$) arises from improved resolution of recirculation and viscous losses around pin fins—an expected and physically justified trend. The fine mesh was therefore selected for all subsequent simulations owing to its superior resolution of wall-adjacent gradients, accurate representation of viscous and thermal dissipation mechanisms, and 4.2% enhancement in thermal-hydraulic efficiency, demonstrating that the pin-fin design offers meaningful heat-transfer improvement despite a modest rise in pressure loss.

Table 5: Mesh independence study.

Quantity	Coarse Mesh	Medium Mesh	Fine Mesh	Percentage Deviation Medium to Fine (%)	Percentage Deviation Coarse to Fine (%)
No. of elements	247,886	707,788	1,584,540	-	-
Pressure Drop	603.4312	604.3910	624.4862	+3.32%	+3.49%
Nusselt Number	4.437	4.507	4.347	-3.55%	-2.03%
Friction factor	0.81	0.81148	0.83	+2.28%	+2.47%

Source: Authors, (2026).

III. MATERIALS AND METHODS

III.3.1 Boundary Conditions and Numerical Procedure

The numerical investigation of the microchannel heat sink (MCHS) is performed under carefully defined boundary conditions and a systematic numerical procedure to accurately capture the coupled fluid flow and heat transfer characteristics. As in Table 3, At the inlet, a uniform velocity is prescribed corresponding to four distinct velocities (0.5, 1.0, 1.5, and 2.0 m/s), which correspond to the Reynolds numbers between 56 to 225 representing laminar flow regimes typical of microchannel applications. The inlet temperature of the working fluid is water and MWCNT–water nanofluids with 1%, 2%, and 3% concentrations maintained at 300 K. This ensures consistent thermal conditions for comparative analysis across all test cases. At the outlet, a zero-gauge pressure condition is imposed to simulate atmospheric discharge, ensuring a physically realistic pressure drop along the flow direction. A uniform heat flux of 10KW/m² is applied at the bottom wall of the microchannel, emulating high heat generation from electronic components or chip surfaces. The other three walls are considered adiabatic, restricting heat loss to the surroundings. All solid–fluid interfaces are assumed to have a no-slip condition to accurately represent viscous effects within the confined microchannel domain. The numerical procedure is executed using the finite volume method (FVM) in ANSYS Fluent in the governing equations. The pressure–velocity coupling is handled using the SIMPLE algorithm, which ensures convergence stability and accurate momentum–pressure interaction. The convective terms in the governing equations are discretized using the second-order upwind scheme. Convergence criteria are set to strict tolerances to ensure computational reliability. The residuals are maintained below 10⁻⁹ for the energy equation and 10⁻⁶ for both continuity and momentum equations.

Table 3: Boundary conditions.

Boundary Conditions	Values
Inlet Velocity	0.5 - 2 m/s
Inlet Temperature	300 K
Outlet Pressure	0 Pas
Heat Flux	10 kW/m ²
Reynolds Numbers	56.23, 112.5, 168.7, 225
Flow Type	Laminar

Source: Authors, (2026).

III.3.2 Mathematical model:

The steady-state, laminar, incompressible flow and conjugate heat transfer through the microchannel are modeled using the three-dimensional forms of the continuity, momentum, and energy equations. These equations govern both the fluid and solid domains to capture the coupled heat transfer behavior.

Continuity

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

Momentum.

$$\rho_1 (\vec{u} \cdot \nabla \vec{u}) = -\nabla p + \mu_1 \nabla^2 \vec{u} \quad (2)$$

Energy.

$$\rho_1 C_{p1} (\vec{u} \cdot \nabla T_f) = k_1 \cdot \nabla^2 T \quad (3)$$

Energy (solid).

$$\nabla(k_s \cdot \nabla T_s) = 0 \quad (4)$$

III.3.3 Numerical Calculations:

The Hydraulic Diameter (D_h) equation,

$$D_h = \frac{4(W_{ch} \times H_{ch})}{2(W_{ch} + H_{ch})} \quad (5)$$

Where H_{ch} - Height and W_{ch} -width

The Reynolds equation,

$$Re = \frac{\rho_l u_l D_h}{\mu_l} \quad (6)$$

The heat transfer coefficient,

$$h = \frac{Q}{T_w - T_f} \quad (7)$$

The Nusselt calculation,

$$Nu = \frac{h D_h}{k} \quad (8)$$

The friction factor can be determined from the following equations,

$$f = \frac{2 D_h \Delta p}{L \rho u_{in}^2} \quad (9)$$

The thermal Performance Factor (TPF) is calculated by,

$$TPF = \frac{Nu/Nu_o}{(f/f_o)^{1/3}} \quad (10)$$

III.3.4 Nanofluid Properties

Density:[28]

$$\rho_{nf} = (1 - \phi) \rho_{bf} + \phi \rho_p$$

Viscosity: (Brinkman - model) [29]

$$\mu = \frac{\mu_{bf}}{(1 - \phi)^{2.5}}$$

Thermal Conductivity: (Hamilton - Crosser) [30]

$$k_{nf} = k_{bf} \frac{k_s + (n-1) k_{bf} - (n-1) \phi (k_s - k_{bf})}{k_s + (n-1) k_{bf} + \phi (k_s - k_{bf})}$$

Specific Heat Capacity: [31].

$$C_{p,nf} = \frac{(1-\phi) \rho_{bf} C_{p,bf} + \phi \rho_p C_{p,p}}{\rho_{nf}}$$

Table 4: Thermo-physical properties of Nanofluid.

Fluid Type	Concentration (wt%)	Density (kg/m ³)	Specific Heat (J/kg·K)	Thermal Conductivity (W/m·K)	Viscosity (kg/m·s)
Water	0	998.2	4182	0.6	0.000891
MWCNT/Water Nanofluid	1	1003.03	4128	0.6254	0.0009128
MWCNT/Water Nanofluid	2	1009.06	4074.62	0.6441	0.0009367
MWCNT/Water Nanofluid	3	1015.09	4021.89	0.6633	0.0009617

Source: [27].

IV. RESULTS AND DISCUSSIONS

IV.4.1 Flow field Visualization:

The velocity contour, streamlines and the vector plots across all configurations exhibit the combined influence of velocity magnitude, pin height, and nanofluid concentration on the internal flow behavior of the microchannel. At lower velocities (0.5–1.0 m/s) the flow remains predominantly laminar with well-defined, symmetric streamlines and low momentum exchange near the pin surfaces shown in the Velocity Vector Fig 2. In these cases recirculation zones are small and confined to the wake regions behind each pin fin. The reduced inertia limits the fluid ability to reattach quickly, leading to a thicker boundary layer and consequently weaker convective heat transfer. As velocity increases (1.5–2.0 m/s) the streamlines become progressively elongated and aligned with the channel direction indicating higher kinetic energy and enhanced flow mixing shown in the Fig 3 with Velocity Contour enabled with the Velocity Streamlines. The wake regions shrink and the reattachment points move closer to the trailing edge of the pins which reflect the thinning of the velocity boundary layer and improved convective transport.

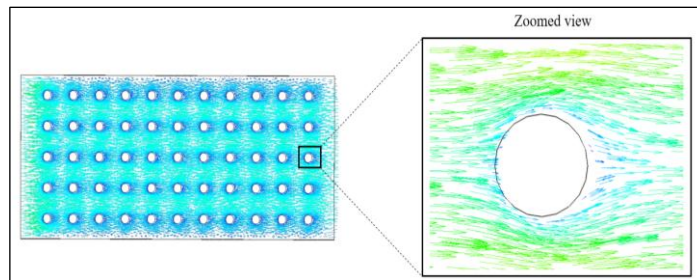


Figure 2: Velocity Vectors along the channel and a zoomed view around a single pin fin 6.2 μm with 3% concentration of MWCNT at 2 m/s.

Source: Authors, (2026).

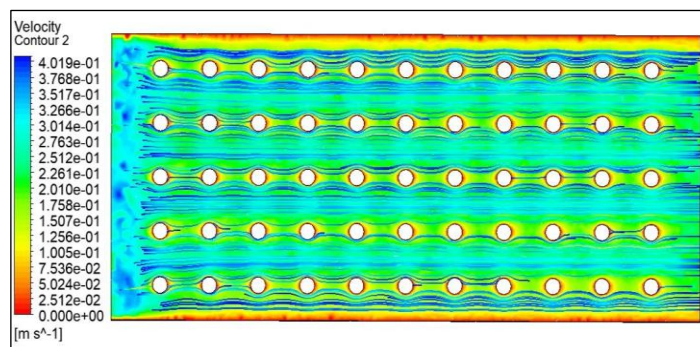


Figure 3: Velocity contour with Streamlines of 6.2 μm pin fin with 3% concentration of MWCNT at 2 m/s.

Source: Authors, (2026).

The influence of pin height of taller pins (31 μm and 43.4 μm) produce stronger flow obstruction and intensified local acceleration around the pin tips, resulting in elevated velocity gradients and increased flow intensity in their vicinity. This accelerates fluid mixing and disrupts the thermal boundary layer, boosting overall heat transfer performance. In contrast, shorter pin fins (6.2 μm and 18.6 μm) yield smoother streamlines with limited mixing at lower velocities. Figure 2 represents the velocity vector field in the microchannel, showing that the bulk region is dominated by smooth forward flow while each of the pin fins has distinct flow deviation. Enlarging the view locally elucidates the local acceleration and deflection of streamlines around the surface of the pin as velocity gradients become large in the wake region. The behavior confirms what was expected to be characteristic shear layers and localized recirculation zones that enhance the mixing and indeed improve convective heat transfer.

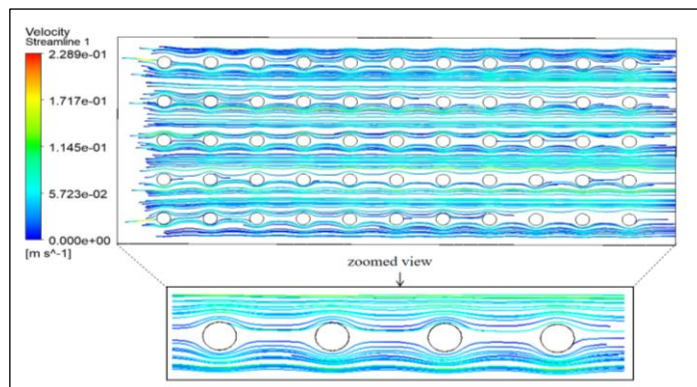


Figure 4: Velocity Streamlines along the plane above the pin fin (~6.3 μm pin fin) with 3% concentration of MWCNT at 2 m/s.

Source: Authors, (2026).

Figure 4, shows the transition of the Streamlines in the same channel with $6.2\ \mu\text{m}$ pin fin channel with 3% concentration of MWCNT at 2 m/s, but at a certain plane where the flow pattern changes slightly above the pin height approximately at $\sim 6.3\ \mu\text{m}$. The velocity streamlines show that when pin heights are lower and velocities are reduced, the flow develops more smoothly and evenly, allowing the fluid to interact longer with the channel walls. This creates clear boundary layers and stable flow areas between the pin fins. On the other hand, higher pin heights and faster velocities increase mixing and speed up the flow around the fins, which causes more turbulence and disturbances. Adding MWCNT nanofluids, especially at higher concentrations, improves momentum diffusion near the walls and changes how the streamlines behave, leading to better flow interaction and more uniform mixing. Even with these differences, the main flow pattern stays the same in all cases, with clear circulation zones and faster regions around the pin fins. When comparing the working fluids, the addition of MWCNT nanoparticles alters the velocity distribution by slightly increasing flow resistance due to higher viscosity. At 1% and 2% concentrations, this resistance modestly dampens near to the wall velocities. Whereas at 3% MWCNT, the effect is more pronounced with marginally thicker velocity gradients near pin surfaces. However the enhanced thermal conductivity and momentum diffusion of MWCNT nanofluids promote stronger velocity recovery downstream and greater homogenization of flow fields. Among all the cases, 3% MWCNT exhibits the most balanced velocity distribution demonstrating the best synergy between viscous dissipation and convective enhancement.

IV.4.2 Temperature Field Visualization:

The temperature contours reveal a clear dependence of thermal field uniformity on both pin-fin height and inlet velocity. At low pin height ($6.2\ \mu\text{m}$) and low velocity, all working fluids exhibit a higher wall temperature and a larger temperature gradient between the inlet and outlet. This occurs because short fins generate weaker flow disturbances and inadequate boundary-layer disruption, resulting in poor mixing of the near-wall fluid. Simultaneously, the low inlet velocity leads to a thicker thermal boundary layer and a longer fluid residence time, allowing heat to accumulate near the heated wall. The combined effect of weak geometric mixing and slow advection leads to the largest temperature rise in these cases. As the pin height increases, a substantial improvement in heat transfer uniformity is observed. Taller fins ($18.6\ \mu\text{m}$, $31\ \mu\text{m}$, and $43.4\ \mu\text{m}$) extend deeper into the flow field, promoting enhanced fluid impingement and wake re-circulation zones between successive pins. This intensifies boundary-layer disturbance and induces secondary mixing, which renews the fluid near the heated surfaces more effectively. Consequently, the local wall temperature decreases and the overall temperature distribution becomes more uniform along the flow path. The benefit of increased pin height is most evident at moderate-to-high velocities, where the combined influence of fin-induced turbulence and convective momentum transport yields superior cooling performance. Similarly, increasing the inlet velocity strengthens convective transport and reduces the temperature difference between the wall and bulk fluid. At higher velocities, the fluid residence time decreases yet the energy removal rate rises due to higher heat transfer coefficients and thinner thermal boundary layers. Hence, the contours at 2 m/s exhibit more uniform color gradients and lower maximum temperatures than those at 0.5 m/s (Fig 5). However, excessively high velocities also lead to increased frictional losses and pressure drop which can impact system efficiency a trade-off captured through the Thermal Performance Factor (TPF) analysis. Across all geometric and flow conditions the MWCNT nanofluids consistently outperform water in dissipating heat, with improvements increasing with nano-particle concentration. The 3% MWCNT nanofluid exhibits the lowest wall temperatures and the most uniform thermal contours due to its higher effective thermal conductivity and microconvective energy transport enabled by nanoparticle Brownian motion. The 1% and 2% nanofluids also enhance thermal performance compared to pure water but to a lesser extent yet following the same overall temperature trend. The influence of concentration becomes more prominent at higher pin heights and velocities where convective mixing complements the intrinsic conductivity enhancement of the nanofluid.

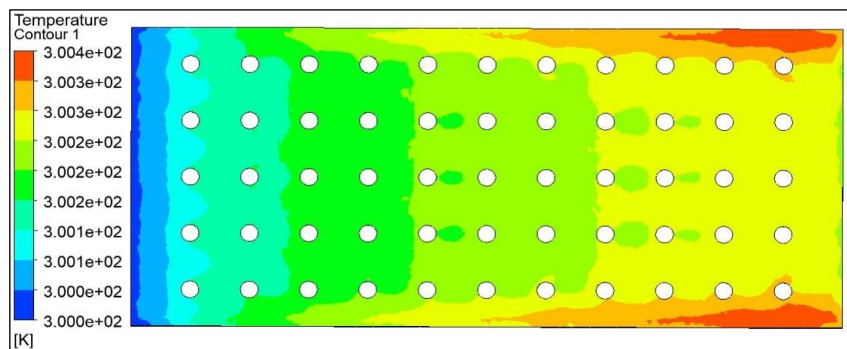


Figure 5: Temperature Contour along the channel of $6.2\ \mu\text{m}$ pin fin with 3% concentration of MWCNT at 2 m/s.
Source: Authors, (2026).

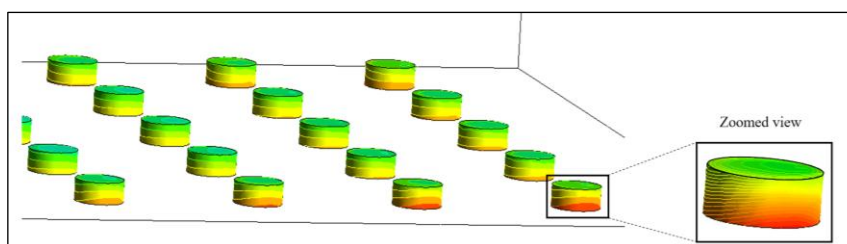


Figure 6: Temperature Distribution along the height of the $6.2\ \mu\text{m}$ pin fin with 3% concentration of MWCNT at 2 m/s.
Source: Authors, (2026).

The temperature contour over each pin fin reveals a noticeable axial gradient with higher temperatures near the base (red/orange) and gradually lower ones toward the top surface (yellow/ green) (Fig 6). This pattern demonstrates that the vicinity near the pin substrate interface is under stronger thermal loading while the upper portions of the pin fins are relatively cooler because of the enhanced removal of heat by convection of the fluid. In the zoomed view, there is an indication of smooth vertical temperature stratification which supports good heat spreading from the base to the fin tip.

IV.4.3 Pressure Drop:

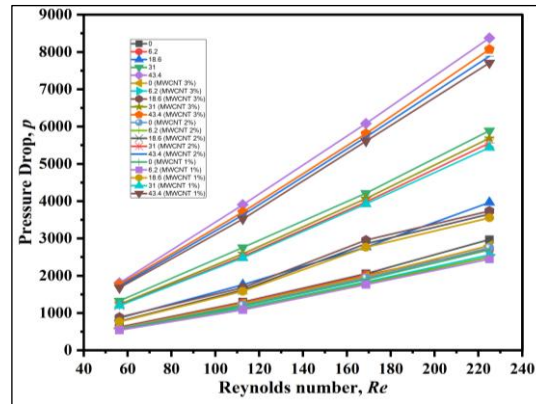


Figure 7: Pressure Drop evaluation with Reynolds number.
Source: Authors, (2026)

Figure 7 elucidates the pressure drop (ΔP) with Reynolds number for water and MWCNT nanofluids across various pin-fin configurations shows a sharp rise with velocity (shown in Fig 1) because the dynamic pressure term ($\rho v^2/2$) dominates at higher velocities causing ΔP to scale approximately with the square of the flow velocity. Although the friction factor slightly decreases with increasing Reynolds number, the rapid growth of inertial forces and intensified shear near pin structures leads to a net increase in pressure loss. At lower Reynolds number, ΔP is minimal, dominated by viscous effects and smooth flow. As velocity approaches 2 m/s, inertial effects intensify, causing greater local acceleration and pressure loss due to increased flow recirculation and wake formation around pins. The Channels with taller fins (43.4 μm) exhibit the highest pressure drop owing to increased obstruction, wetted area, and form drag, while the plain channel records the lowest. Intermediate fin heights (18.6 μm and 31.0 μm) offer moderate losses, balancing flow resistance and heat transfer enhancement. Thus, fin geometry directly governs the trade-off between pressure loss and surface augmentation. MWCNT nanofluids consistently produce higher ΔP than water because of increased effective viscosity and nanoparticle-induced flow resistance. At higher velocities (≥ 1.5 m/s), the gap between water and nanofluids narrows as inertial forces dominate. While MWCNT nanofluids require greater pumping power due to increased flow resistance, their superior heat transfer performance compensates for this drawback. Hence, the observed pressure drop trends reinforce the importance of optimizing fin geometry and nanoparticle concentration to achieve a balanced thermal hydraulic performance in microchannel heat sink applications.

IV.4.4 Friction Factor:

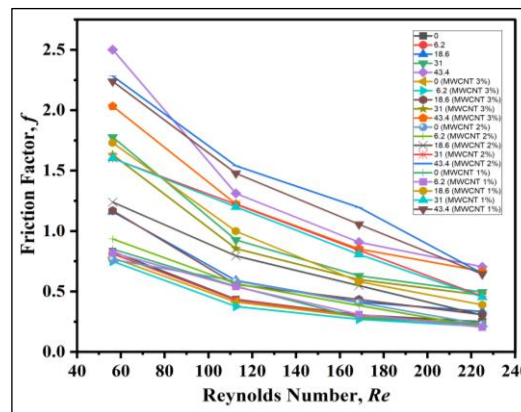


Figure 8: Friction Factor variation with Reynolds number.
Source: Authors, (2026).

The trend of friction factor (f) with Reynolds number for water and MWCNT nanofluid concentrations across different pin fin heights is presented for exact similar parameters in Figure 8. A clear inverse relationship between f and Reynolds number is observed for all configurations, which is typically consistent with laminar scale Reynolds number behavior ($Re = 56-225$) and the inverse proportionality to Re of viscous losses. Due to the dominance of viscous shear stress and form drag caused by the pin fin obstructions, the friction factor reaches its maximum value of $f \approx 0.8-2.5$ at lower Reynolds numbers. Significant pressure losses occur in this regime because the flow lacks the kinetic energy to overcome the surface friction and flow separation created behind each pin fin. The boundary layer thins and the wall shear stress contribution per unit mass flow decreases as the velocity increases (0.5 $\rightarrow$ 2.0 m/s) due to the inertial effects.

As a result, the friction factor gradually drops and tends to stabilise at higher velocities, signifying the relative suppression of viscous dominance. The plain channel has the lowest friction factor at all velocities, while the channels with taller pin fins (43.4 μm) have the highest. This is due to the fact that raising the pin height increases the wetted area and flow obstruction, which results in a larger pressure drop. Additionally, stronger flow separation and wake zones are produced by higher pin structures, which increase the local form drag. Higher friction factors result from the viscous drag dominating the total pressure loss at lower velocities. Because of their increased surface area and increased flow obstruction, taller pin fins intensify this effect, leading to significant form drag and pressure recovery losses. For all geometries, the flow moves towards inertia-controlled behaviour as velocity rises, resulting in a decrease in the friction factor. The MWCNT nanofluid typically shows higher flow resistance because of its increased effective viscosity, improved wall fluid and inter molecular interactions. However, due to the increase in thermal conductivity which indirectly affects viscosity, the nanofluid exhibits slightly lower dimensionless friction factors than water at 3% concentration. This effect is more pronounced at low velocities due to the dominance of viscous forces. However, as velocity increases, the relative difference between water and MWCNT decreases due to the increasing importance of inertial effects. Because of this, MWCNT nanofluid requires more pumping power because it has a higher hydraulic resistance than water. Additionally, as evidenced by the higher Nusselt number values, it has a greater capacity for heat transfer which leads to better thermal performance overall.

IV.4.5 Nusselt Number:

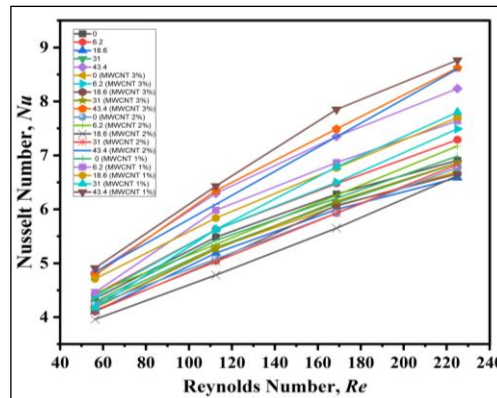


Figure 9: Nusselt number relation with Reynolds number.

Source: Authors, (2026).

Figure 9 elucidates the variation of the Nusselt number (Nu) with inlet velocity for both deionized water and MWCNT nanofluid concentrations. However, simulations has been conducted for 1%, 2% and 3% volume concentrations. A consistent increase in Nu with velocity is observed for all cases, highlighting the strong influence of Reynolds number on convective heat transfer in microchannel configurations (Fig 3). At the lowest Reynolds number (lowest velocity (0.5 m/s)), the Nu values remain relatively low for both water and MWCNT, ranging between 4 - 5. This behavior can be attributed to the dominance of conduction mechanism and the development of a thick thermal boundary layer in the absence of strong convective forces. Consequently, the difference between water and MWCNT is minimal under these conditions, as the nanofluid's enhanced thermal conductivity is not fully exploited. Even at lower nanoparticle concentrations (1% and 2%), the nanofluid exhibits noticeable enhancement compared to water which confirm that significant heat transfer improvement can be achieved without requiring the highest concentration. This can be primary attributed to the enhanced thermal conductivity of MWCNT nanofluids which is approximately 4.23%, 7.35%, and 10.55% higher than the base fluid (water) at 1 wt%, 2 wt%, and 3 wt% concentrations, respectively as compared to the based fluid. As the Reynolds number increases (velocity increases to 1.0-1.5 m/s), the Nusselt number rises steadily due to stronger convective transport and progressive thinning of the thermal boundary layer. In this regime, the benefit of MWCNT becomes more evident. The nanofluid consistently exhibits higher Nu values compared to water across all pinfin configurations which confirms the role of nanoparticles in enhancing heat transport through increased effective thermal conductivity and micro-scale convective effects.

For the 1% and 2% MWCNT concentrations, Nu values also exceed those of water demonstrating a clear improvement in heat transfer even though they remain lower than the 3% concentration. The relative enhancement is more pronounced at higher velocities and taller pin-fin geometries. In these scenarios the combined effect of thermal conductivity enhancement (due to nanofluid) and local disturbance of thermal boundary layer (due to pin geometries) plays a major role. The second effect is more visible at higher velocities and hence you can see a better heat transfer enhancement at higher velocities and taller pin heights. At the highest velocity of 2.0 m/s, the enhancement becomes most prominent. For water, Nu reaches values between ~7.0 and 8.0, while for MWCNT, the corresponding Nu extends more than 8.0, particularly in the channel with the tallest pin fins (43.4 μm). This is explained by the synergistic action of high flow momentum, which intensifies mixing around the fins, and the superior thermal conductivity of MWCNT nanofluid, which accelerates thermal diffusion within the fluid medium. Nu values are consistently higher than water but lower than those of the 3% nanofluid at the 1% and 2% MWCNT concentrations. Configurations with taller fins show the incremental improvement over water, emphasising the contribution of even lower nanoparticle loadings. Shorter fins (6.2 μm) provide only moderate boundary layer disruption, leading to smaller increments in Nu, whereas taller fins (31.0 μm and 43.4 μm) induce significant flow separation and accelerated motion, thereby initiates fluid mixing and promoting superior convective heat transfer. The tallest fins consistently deliver the maximum Nu for both working fluids, with the relative gain being more significant for the nanofluid. Overall, the results demonstrate that both increasing flow velocity and incorporating MWCNT nanoparticles substantially improve heat transfer performance in microchannel heat sinks. The improvement is further amplified by the presence of taller pin fins, which serve as passive promoters of thermal hydraulic enhancement.

IV.4.6 TPF:

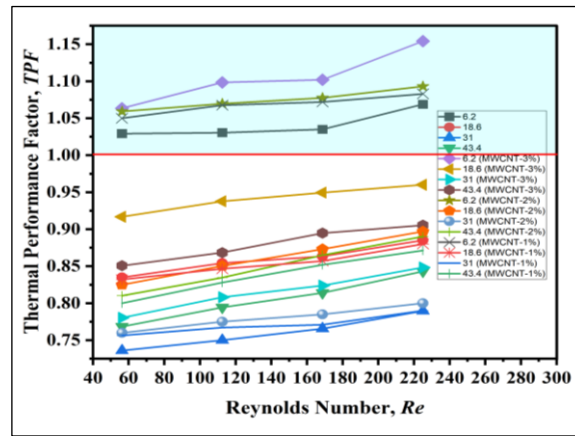


Figure 10: Thermal Performance Factor vs Reynolds number.
Source: Authors, (2026).

The variation of the Thermal Performance Factor (TPF) with Reynolds number for water and MWCNT nanofluids of different concentrations across various pin fin heights is illustrated in Fig. 10. The TPF measures heat transfer improvement in relation to increased frictional losses, indicating overall thermohydraulic efficiency. A general trend of increasing TPF with flow velocity is observed for all configurations, reflecting the dominance of convective heat transfer improvements over the associated hydraulic penalties at higher Reynolds numbers. At low velocity (0.5 m/s), the TPF values remain below unity for most configurations, particularly for the plain and shorter pin-fin channels. This indicates that, under laminar and viscous-dominated conditions, the pressure-drop penalty outweighs the modest heat-transfer enhancement achieved at such low flow rates. As the Reynolds number increases, TPF values progressively improve, reaching peak values around 1.1–1.4 depending on the configuration and nanofluid concentration. This rise can be attributed to enhanced flow mixing, boundary-layer disruption, and reduced thermal boundary layer thickness, which collectively augment the convective heat transfer rate more strongly than the corresponding rise in pressure losses. Among all the geometries, the channels with intermediate pin fin heights (18.6 μm and 31.0 μm and tallest 43.4 μm fins) have some fluctuations in the pattern, due to the combined reaction of convective transport and friction losses.

The shortest fin (6.2 μm) consistently demonstrate superior thermal performance compared to the other three fin heights (18.6 μm , 31 μm and 43.4 μm). The moderate pin structures provide an optimal balance between surface area augmentation and flow obstruction which is sufficient to intensify thermal mixing without inducing excessive form drag. All MWCNT nanofluids exhibit higher TPF values than water, confirming their superior thermohydraulic performance. The improvement becomes more evident with increasing nanoparticle concentration at 1% MWCNT, the enhancement is marginal, whereas at 3%, the TPF surpasses unity for most pin fin geometries, indicating a net performance gain. This improvement is due to higher thermal conductivity and enhanced micro-convection from Brownian motion in the MWCNT suspensions, which support heat diffusion and wall heat removal. However, as concentration rises further (more than 3%), increased viscosity limits these benefits, resulting in reduced enhancement at higher concentrations. Overall, the integration of pinfin structures with MWCNT nanofluids achieves a mutual enhancement in system efficiency. The combination of intensified heat transfer and manageable pressure drop results in a thermally favorable regime where $\text{TPF} > 1$ for optimized pin heights and flow rates. These findings clearly indicate that the use of MWCNT nanofluids, particularly at 2–3% concentration, provides a substantial improvement in the overall thermal-hydraulic performance of microchannel heat sinks, validating their potential as advanced cooling solutions for high heat-flux microscale applications.

IV.4.7 Data-Driven Heatmap Analysis of Thermal Performance (TPF) Using MATLAB

The combined thermal-hydraulic performance (TPF) for each of the five pin heights (0, 6.2, 18.6, 31, and 43.4 μm), four flow velocities (0.5–2 m/s), and four working fluids (water and 1%, 2%, and 3% MWCNT nanofluids) was visualized using a data-driven heatmap created in MATLAB. Multidimensional CFD output is processed, normalized and mapped using an ML-based exploratory analysis workflow in this visualization approach to uncover performance patterns that are not immediately apparent from isolated plots. This method supports high-dimensional engineering interpretation by using the same datacentric principles that are frequently used in machine-learning research, even though it is not a predictive machine learning model. The heat map clearly and consistently demonstrates that TPF rises with both Reynolds number and nanoparticle concentration throughout the entire dataset (Fig 11). Because of its much higher effective thermal conductivity, the enhancement is most noticeable for the 3% MWCNT nanofluid. The pin height of 6.2 μm with 3% MWCNT yields the highest TPF among all evaluated conditions when comparing the geometric configurations, especially at the highest velocity. This happens because the high concentration of nanoparticles significantly increases heat transfer, resulting in the most advantageous trade-off in the TPF metric, while the shorter pin height maintains a comparatively lower frictional penalty. Higher pin heights such as 31 μm and 43.4 μm , are thermally superior in terms of heat transfer. However, their overall TPF is lower than that of the ideal 6.2 μm case due to increased pressure losses. The anticipated gradation is followed by the intermediate concentrations (1% and 2% MWCNT). The proportionate impact of nanoparticle loading is demonstrated by the fact that 2% consistently performs better than 1% and water but both fall short of the 3% cases because of the relatively smaller heat-transfer enhancement. Predominantly, the heatmap visualisation offers a quick, thorough, and objective evaluation of the behaviour of the system. The most efficient configuration which is indisputably the 3% MWCNT nanofluid with 6.2 μm pin height. This highlights how crucial it is to strike a balance between fluid-property improvement and geometric optimisation when aiming for concurrent improvements in thermal and hydraulic performance.

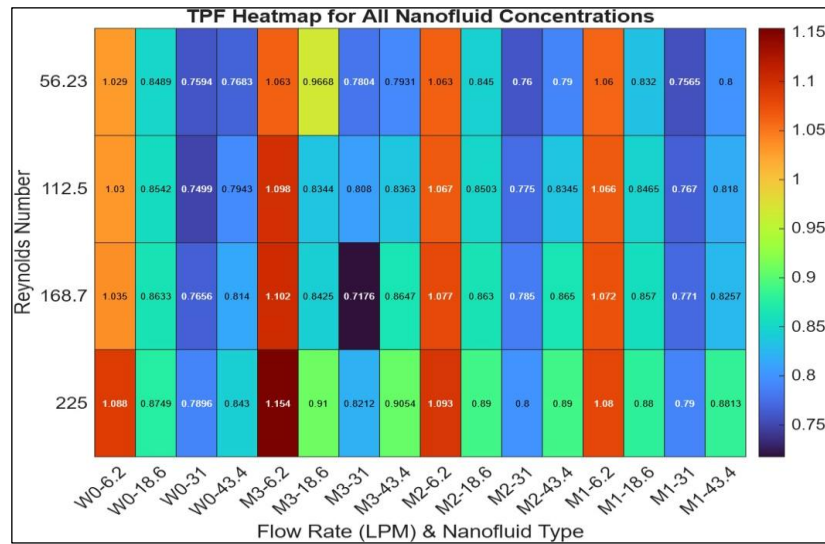


Figure 11: Heat Map Visualization of TPF for MWCNT across the pin heights for various Reynolds numbers.
Source: Authors, (2026)

The Heatmap's Advantages

The heatmap is a crucial analytical tool because it allows immediate comparison between four Reynolds numbers, five pin heights, and four fluids;
 exposing non-linear relationships between nanofluid concentration and geometric parameters;
 facilitating data-driven decision making akin to exploratory analysis in machine learning;
 creating a single, comprehensible visual framework from multidimensional CFD outputs.

V. CONCLUSIONS

The thermo-hydraulic performance of a microchannel heat sink with pin fins of different heights was examined in this study using water and MWCNT-water nanofluids (1-3 wt %). The findings verify that thermal transport in the microchannel is significantly improved by both nanoparticle concentration and Reynolds number. The 3% MWCNT nanofluid continuously produced the highest Nusselt number under all operating conditions, which increases notably at higher flow rates. The effect is especially noticeable for the lowest pin height configuration where superior heat transfer enhancement is produced by the combination of enhanced nanofluid conductivity and increased turbulence intensity. All pin heights exhibit the same increasing trend with mass flow rate and nanoparticle loading according to pressure-drop analyses. Compared to taller pin arrays, lower pin heights show noticeably smaller pressure penalties, particularly at lower velocities. Higher pin heights, on the other hand, result in significantly greater pressure drops because of increased flow obstruction; at higher Reynolds numbers, the difference between nanofluids and water is more noticeable. The 3% nanofluid produced the largest pressure drop across all fin heights, which is in line with its higher viscosity and momentum diffusion. For all working fluids, friction factor values drastically drop as the Reynolds number rises. Because viscous forces are less dominant at higher flow rates, the lowest friction factors are found at $Re = 225$. Because of their greater wetted perimeter and flow resistance, taller pin fins have higher overall friction factors. When compared to water, the use of nanofluids causes a slight increase in friction; however, this difference decreases at higher Reynolds numbers. According to Thermal Performance Factor (TPF) analysis, the 3 % nanofluid with the lowest pin height offers the best balance between hydraulic penalty and heat transfer enhancement. The lower Reynolds number range (especially around $Re \approx 56$) exhibits the highest TPF values, where thermal enhancements significantly outweigh the comparatively slight increase in pressure drop. For all pin heights and concentrations, TPF gradually decreases with increasing Reynolds number; however, because of their lower flow resistance and higher enhancement ratio, lower pin heights consistently perform better than taller pins. Overall, lower pin-height microchannel designs combined with MWCNT nanofluids particularly at 3 weight percent concentration produce a significant increase in heat transfer capability while preserving acceptable pressure drops. These results show that the thermal management efficiency of microchannel heat sinks used in next-generation electronic cooling applications can be greatly increased by optimizing fin geometry in conjunction with customized nanofluid formulations.

VI. AUTHOR'S CONTRIBUTION

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Investigation: S Sorna Latha, Ankush D Tharkar, Divya Haridas.

Discussion of results: S Sorna Latha, Ankush D Tharkar, Divya Haridas.

Writing – Original Draft: S Sorna Latha, Ankush D Tharkar, Divya Haridas.

Writing – Review and Editing: S Sorna Latha, Ankush D Tharkar, Divya Haridas.

Resources: S Sorna Latha, Ankush D Tharkar, Divya Haridas.

Supervision: S Sorna Latha, Ankush D Tharkar, Divya Haridas.

Approval of the final text: S Sorna Latha, Ankush D Tharkar, Divya Haridas.

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