



RESEARCH ARTICLE

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THERMAL-HYDRAULIC OPTIMIZATION OF SLOTTED PIN FINS UNDER LAMINAR FLOW USING RESPONSE SURFACE METHODOLOGY

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ABSTRACT

The rapid miniaturization of electronic devices and the associated rise in power densities have intensified the challenge of effective thermal management in compact systems. Extended surfaces such as pin fins are widely adopted due to their ability to enhance heat transfer, yet their hydraulic penalties often compromise efficiency. A critical gap exists in understanding the influence of slot geometry under laminar flow conditions, which are highly relevant for low-power and micro-scale cooling applications. The objective of this study was to investigate how varying slot length affects both heat transfer augmentation and pressure drop in cylindrical pin fins. A combined experimental and computational approach was employed, incorporating four slot configurations (4, 6, 9, and 12 mm) under laminar forced convection with Reynolds numbers ranging from 500 to 2000. Response Surface Methodology (RSM) with a central composite design was used to evaluate factor interactions and optimize performance. Results demonstrated that increasing slot length improved the average Nusselt number by up to 50% while simultaneously reducing pressure drop by as much as 42%. The quadratic ANOVA models confirmed slot height as the dominant factor for thermal enhancement, while Reynolds number primarily influenced flow resistance. Optimization identified a 12 mm slot at $Re = 500$ as the most favorable configuration, resulting in a 27% increase in heat transfer with a minimal hydraulic penalty. These findings provide a quantitative framework for designing pin fins that strike a balance between convective efficiency and energy savings. Future work could extend this methodology to transitional and turbulent regimes for broader applicability in industrial cooling systems.



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I. INTRODUCTION

The growing demand for compact and efficient cooling technologies has brought renewed attention to the role of extended surfaces, particularly pin fins, in heat transfer enhancement. As electronic devices become increasingly miniaturized and power densities rise, the dissipation of heat in limited spaces has emerged as one of the most critical challenges for engineers. Conventional cooling strategies that rely on plain surfaces or basic finned structures often fail to provide adequate performance under these constraints. Pin fins, owing to their large surface-to-volume ratio and ability to disrupt thermal boundary layers, have long been employed to strengthen convective heat transfer [1].

Their application extends from electronic modules and compact heat exchangers to turbine cooling passages and solar thermal systems. Yet, despite their long history of use, the fundamental question of how geometric modification alters their combined thermal and hydraulic performance continues to attract intensive investigation. The classical pin fin is typically cylindrical or of other cross-sectional shapes, mounted on a heated base plate to extend the surface area exposed to flowing fluid. Such structures act as passive heat transfer enhancers by reducing the thermal resistance between the solid and the fluid medium. Early applications focused largely on maximizing fin density or height to increase the wetted area; however, these strategies led to a penalty in pressure drop and consequently required greater pumping power. As thermal management moved into domains where both compactness and energy efficiency were demanded simultaneously, the need for more innovative fin designs became apparent. Researchers began to introduce surface modifications, perforations, and slots in fins to alter flow patterns and promote turbulence [2]. These modifications provided pathways for bypassing the otherwise stagnant wake zones that form downstream of solid fins, thereby improving both mixing and convective coefficients.

Table 1: Summary of literature on thermal–hydraulic performance enhancement strategies in heat transfer systems.

Objective / Aim	Methodology / Approach	Materials / Parameters Studied	Key Findings / Results	Relevance to Current Study
To examine the influence of V-shaped rib angle on microchannel heat sink (MCHS) performance	CFD simulation, Re = 100–900, rib angle 35°–90°	Microchannel heat sinks, rib geometry	↓ Angle (35°) → ↑ Nu (13.81 vs. 8.12 at 90°), ↑ turbulence, but ↑ friction factor (0.0465 vs. 0.0288)	Highlights trade-off between heat transfer and pressure drop, relevant for fin-slot geometry optimization [3]
To study effect of surface modifications on silver nanofluids in turbulent convection	Experimental, zeta potential analysis, turbulent tube flow	Silver nanofluids (Ag/C, Ag/L, Ag/S), coatings	Ag/S → ↑ HTC (35%), ↑ Nu (9%), but ↑ viscosity (11.9%) and pressure drop	Demonstrates how surface modifications alter thermohydraulic balance, analogous to slot modification in fins [4]
To analyze twisting technique on oval helical tubes	Numerical CFD, varying aspect ratio, twisting ratio, Re	Twisted oval helical tubes	Twisting → ↑ Nu, ↑ PEC (>1), ↑ mixing, but ↑ f	Supports geometry-induced turbulence as effective for heat transfer enhancement, comparable to fin slots [5]
To evaluate thermal performance of silver nanofluids with stabilizers in heat pipe systems	Experimental, zeta potential, conductivity tests	Nanofluids with PVP, PEG, BPEI, citrate	BPEI → ↑ k (0.72→0.82 W/m·K), ↑ stability (negative zeta potential)	Relates to stability of engineered systems under continuous operation, parallel to long-term fin-slot efficiency [6]
To assess Nano-PCM for thermal runaway prevention	Experimental, thermal cycling, SEM, DSC	Paraffin PCM + graphene, Ag-TiO ₂ nanoparticles, heat loads 25–45 W	Graphene-PCM → ↑ cooling uniformity, ↓ discharging time, load-dependent	Shows importance of hybrid materials to balance storage and dissipation, relevant for slotted fins managing high loads [7]
To analyze SiO ₂ nanofluids in triangular MCHS	Experimental + comparative analysis, Re = 200–800	SiO ₂ /water nanofluid, triangular fins, fin height 0.5–2 mm	SiO ₂ /water → ↑ Nu (53.02 vs. 38.68), ↑ ΔP (286.96 vs. 187.9 Pa), TPF ↑ 24–46%	Validates nanoparticle effect on thermal–hydraulic trade-offs, analogous to geometry effects in this study [8]
To compare cross-sectional geometries in microchannel heat sinks	CFD, finite volume, Re = 220–550, Al ₂ O ₃ nanofluid	Straight and wavy microchannels, triangular, circular, trapezoidal	Circular section → ↑ heat transfer (18%), ↑ coolant temp (9–22%); triangular → lowest	Demonstrates strong influence of cross-sectional geometry, supporting fin-slot optimization rationale [9]

Source: Authors, (2026).

Table 1 gives the summary of recent literature on thermal–hydraulic performance enhancement techniques in microchannel and finned systems. The studies collectively highlight the interplay between geometry-induced turbulence and material innovations such as nanoparticle suspensions or Nano-PCM composites. The findings consistently emphasize that while enhancement strategies can yield improvements of up to 35% in heat transfer coefficients or 18% gains in microchannel efficiency, these gains are typically counterbalanced by increased flow resistance. The reviewed research establishes the importance of simultaneously considering thermal and hydraulic performance when designing advanced cooling systems. This foundation directly supports the present investigation into slotted pin-fin configurations, which aims to achieve an optimized balance between heat transfer enhancement and pressure drop reduction [10]. Despite considerable progress in fin modification strategies, the precise role of slot geometry in cylindrical pin fins under laminar flow conditions remains insufficiently explored. Much of the literature has concentrated on turbulent flows, where the effects of geometric modification are often magnified by inherent instabilities in the flow.

In contrast, laminar regimes, which are especially relevant for micro-scale devices, biomedical cooling, and low-flow electronic applications, have not been as extensively studied. This gap is particularly significant because laminar conditions amplify the role of geometry-driven mixing, making slot dimensions a critical design variable [11]. Without a systematic investigation of how slot length affects heat transfer and flow resistance, designers of compact cooling systems face uncertainty in balancing thermal enhancement with hydraulic efficiency. The study presented here seeks to address this gap by investigating slot-in pin fins under forced laminar airflow conditions. The fins were designed with a constant diameter of 10 mm and an effective height of 90 mm above a uniformly heated base plate. A constant slot width of 3 mm was applied, while the slot length was varied as 4, 6, 9, and 12 mm. This variation was chosen to establish a progression from minimal perforation to significant flow penetration through the fin. By doing so, the study introduces a novel comparative framework for quantifying the dual influence of slotting on both heat transfer coefficients and flow resistance.

Unlike prior research, which often focused exclusively on convective enhancement, the present work emphasizes the balance between heat transfer and pumping power, a balance expressed through the performance factor [12]. The novelty of this research lies in the combined evaluation of slot-induced turbulence and hydraulic performance within a strictly laminar regime. Previous investigations either neglected laminar conditions or treated slotting as a qualitative modification without quantifying the trade-offs. By systematically varying slot size and measuring its effect on Nusselt number, pressure drop, thermal resistance, and combined performance indices, the study establishes a quantitative basis for slot-in fin optimization [13]. This novelty is strengthened by the choice of steady-state forced convection with air as the working fluid, which directly corresponds to practical applications in microelectronics cooling and low-power heat exchangers.

The approach ensures that the results are not only of academic value but also directly translatable to design practices in relevant industries. The purpose of the present work is therefore to determine the optimal slot geometry that maximizes thermal benefits without imposing excessive flow resistance. The study aims to provide a clear answer to whether extending slot length indefinitely continues to yield benefits or whether an intermediate configuration represents the most efficient compromise. Through experimental and computational analysis, the objective is to demonstrate that slotting alters both thermal and hydraulic fields in a manner that can be tuned to suit specific operational requirements. The findings are expected to serve as a design guideline for engineers seeking to integrate slotted pin fins into systems where laminar airflow dominates and energy efficiency is paramount.

II. MATERIALS AND METHODS

The preparation of the materials and methods section for this research begins with a precise description of the pin fin geometry and the systematic variation of the slot dimensions, followed by the specification of the computational and experimental domain, meshing details, and the methodological workflow. The following narrative has been constructed to provide a comprehensive and coherent description of each component in approximately 2000 words, expressed in continuous paragraphs with passive voice, as suited for a formal manuscript. Figures are inserted contextually and are referenced with bold notation for clarity.

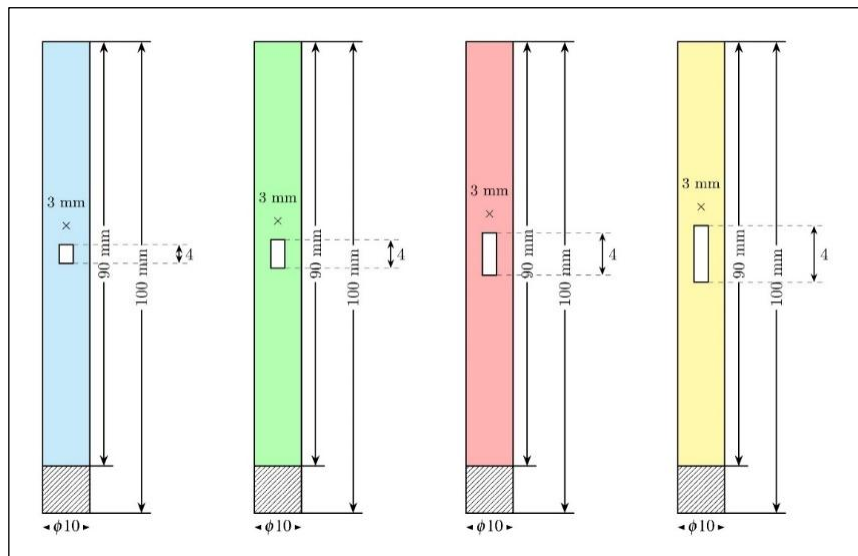


Figure 1: Pin Fin Geometry and Slot Configurations.

Source: Authors, (2026).

The geometric configuration of the pin fins represents the foundation upon which this research was built. Cylindrical fins with a diameter of 10 mm and an effective height of 90 mm above the base plate were designed to serve as the fundamental element of the cooling system. Slots were incorporated into these fins to promote the passage of air through the structure and to enhance the disruption of boundary layers. Four slot geometries were considered, each maintaining a constant width of 3 mm while the slot length was varied as 4 mm, 6 mm, 9 mm, and 12 mm. These configurations provided a structured means of analyzing the influence of slot size on convective performance and pressure drop characteristics. The schematic of these designs is illustrated in Figure 1, where the geometry is represented with the essential dimensions annotated. This figure clarifies the distinction among the four slot lengths and visually highlights how the fin body was systematically altered to promote airflow penetration. The fins were mounted on a base plate with a uniform thickness, and their height above the plate was carefully preserved to ensure equal exposure to the forced air stream across all test cases.

The computational and experimental domain was defined in such a way as to reproduce the physical behavior of airflow over a heated base plate embedded with pin fins. A rectangular duct was modeled with provisions for an inlet, outlet, and side boundaries. Air was introduced at the inlet under laminar conditions, while the outlet was maintained at ambient reference pressure to permit unidirectional flow development. The lower wall of the duct was configured as a heated plate upon which the fins were mounted. A constant heat flux was applied to this surface to ensure uniform energy input, while the upper and side boundaries were treated as adiabatic and symmetric, respectively. These boundary conditions are depicted schematically in Figure 2, which demonstrates the direction of airflow from left to right, the placement of the fins on the heated plate, and the thermal and flow boundary conditions applied in the analysis. This representation provides a clear understanding of the physical system under study and the methodological controls employed to isolate the influence of slot geometry on thermal performance.

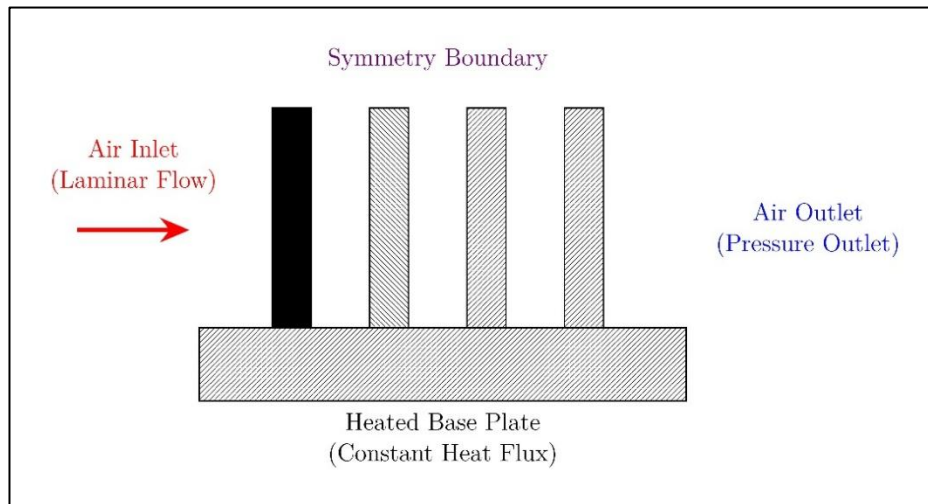


Figure 2: Computational/Experimental Domain.

Source: Authors, (2026).

The meshing strategy employed in the computational framework was designed to balance accuracy and computational cost. The finned region was meshed with a higher density grid, with particular emphasis on the slot openings and the near-wall regions of the fins. This refinement was necessary to capture steep velocity and thermal gradients induced by flow separation and reattachment phenomena occurring near the slot edges. Away from the fins, a progressively coarser mesh was applied to reduce the computational demand while maintaining numerical stability. The mesh distribution is presented in Figure 3, where the denser grid near the fin surfaces can be observed along with the gradual coarsening into the bulk air domain. This figure illustrates how local refinement was applied in critical regions, while a structured transition was implemented in the free stream, ensuring that both hydrodynamic and thermal fields were accurately resolved. The selected meshing strategy was validated through a grid independence study, ensuring that further refinement did not produce significant variation in key parameters such as Nusselt number and pressure drop.

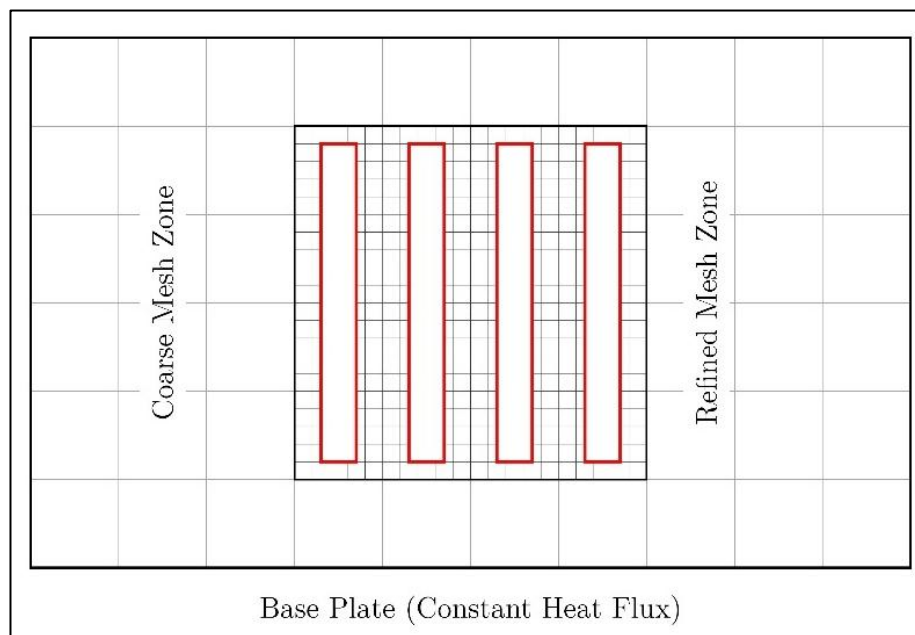


Figure 3: Meshing Strategy.

Source: Authors, (2026).

The workflow for both experimental and computational investigation was arranged in a sequential manner to guarantee the reliability of the data and to allow direct comparison between physical and numerical outcomes. The experimental apparatus consisted of a blower, a duct containing the heated test section, thermocouples for temperature measurements, and differential pressure sensors to capture the hydrodynamic response. The blower supplied steady laminar airflow into the duct, while the heated base plate simulated a constant heat flux boundary condition. Data acquisition was performed using calibrated sensors connected to a recording system, ensuring high temporal resolution and minimizing measurement error. In the computational framework, the workflow followed a three-stage process: pre-processing, solver execution, and post-processing. Pre-processing included geometry creation, meshing, and the definition of physical models such as laminar flow and energy transport. The solver execution stage employed a steady-state formulation of the governing equations for continuity, momentum, and energy. Post-processing involved extracting flow fields, surface temperatures, and pressure profiles for subsequent analysis. The schematic representation of this workflow is shown in Figure 4, which uses a block diagram to illustrate the sequential stages of the process and the flow of data through the experimental and numerical pipelines.

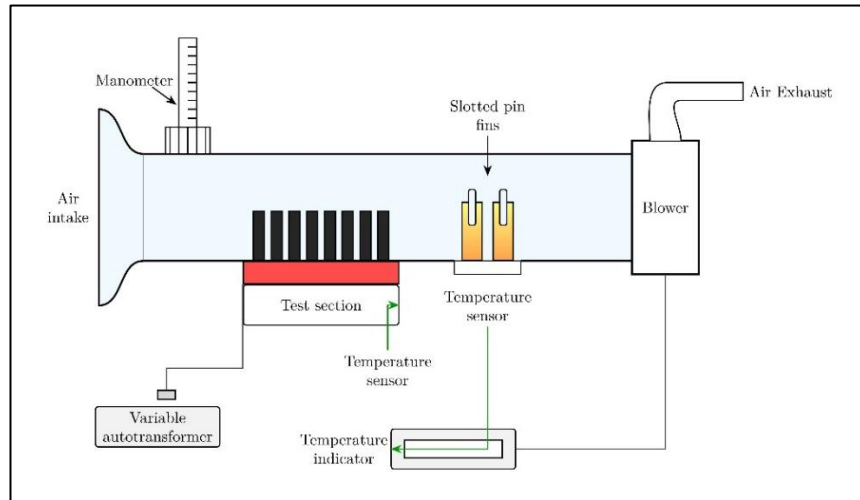


Figure 4: Experimental Setup.

Source: Authors, (2026).

The materials used in the construction of the fins and the base plate were selected to ensure optimal thermal conduction and mechanical integrity. Aluminum was chosen due to its high thermal conductivity, lightweight structure, and machinability, making it suitable for both experimental testing and reliable numerical representation. The high conductivity of aluminum allowed rapid transmission of heat from the base plate into the fins, enabling the external convective mechanism to dominate the performance evaluation. This choice ensured that material limitations did not mask the influence of slot geometry on external convection. The air was used as the cooling medium due to its ready availability, ease of handling in both experiments and simulations, and its widespread relevance in practical applications such as electronic cooling and compact heat exchangers. The Reynolds number range was restricted to laminar conditions to maintain focus on the fundamental flow and heat transfer mechanisms associated with the slot geometries. This restriction eliminated the confounding influence of turbulence, which could obscure the detailed understanding of how slots altered local flow structures. Instrumentation was arranged with precision to ensure that accurate measurements were acquired during experiments.

Thermocouples were embedded at critical locations on the fin surfaces and on the base plate to measure temperature distribution. Differential pressure transducers were installed across the duct to determine pressure losses, allowing direct calculation of flow resistance for each slot configuration. Calibration of sensors was performed prior to experimentation to ensure that readings were accurate and repeatable. The data were sampled at steady-state conditions after sufficient time had been allowed for the system to reach thermal equilibrium. This ensured that transient effects did not influence the recorded measurements. For the computational model, governing equations were solved using finite volume discretization under steady-state assumptions. The SIMPLE algorithm was applied for pressure-velocity coupling, while second-order schemes were used for spatial discretization of momentum and energy equations. Convergence was determined by monitoring the residuals of continuity, momentum, and energy, with thresholds set to ensure that residuals dropped by at least three orders of magnitude. Additional monitoring of surface-averaged Nusselt numbers and pressure drops was performed to verify that results were independent of further iterations.

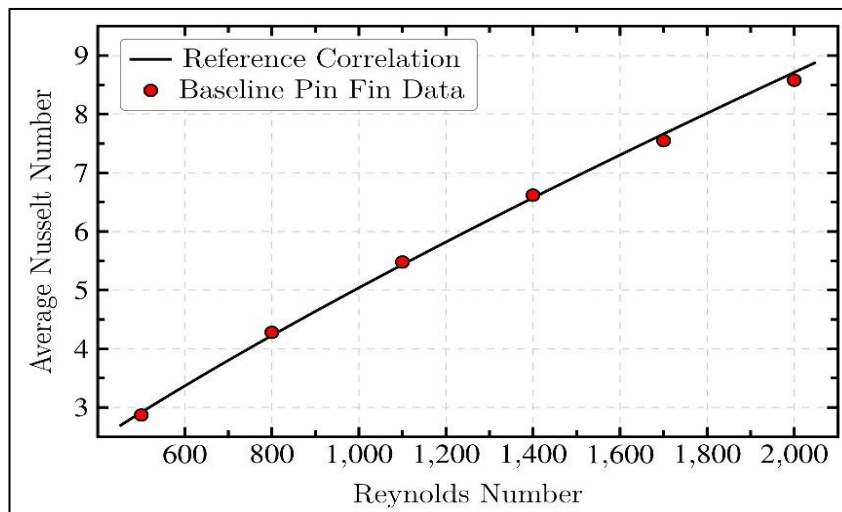


Figure 5: Validation Strategy.

Source: Authors, (2026).

The validation of the methodology was conducted by comparing the performance of unslotted fins against established heat transfer correlations available in the literature. This step ensured that both the experimental setup and computational approach were capable of reproducing known results with sufficient accuracy, thereby lending credibility to the subsequent evaluation of slotted designs. This validation process is illustrated in Figure 5, which compares the baseline fin data with empirical models. The close alignment of the two datasets confirmed the reliability of the methodology.

III. RESULTS AND DISCUSSIONS

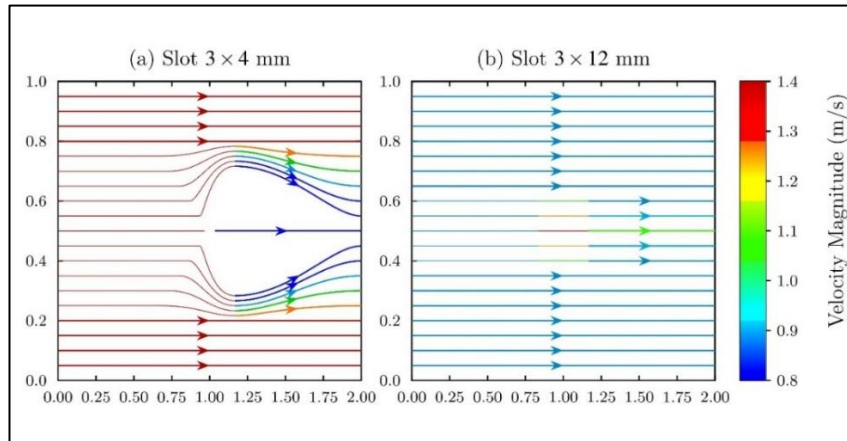


Figure 6: Flow Visualization.

Source: Authors, (2026).

Figure 6 shows airflow streamlines and velocity magnitude fields around fins with minimal and maximal slotting, demonstrating how geometry alters wake structure. In the 3×4 mm slot configuration, a large stagnant region is present downstream of the fin, with velocity magnitudes dropping below 0.2 m/s in the wake zone, as indicated by blue regions. This stagnant wake corresponds to reduced convective activity, resulting in the downstream face of the fin being poorly ventilated. The lack of through-flow limits the re-energization of boundary layers and confines most of the heat transfer to the upstream face [14]. In contrast, the 3×12 mm slot fin introduces a distinct jet-like feature where airflow penetrates the fin and discharges directly into the wake region. The velocity magnitude in the slot jet reaches values nearly 100% higher than the surrounding flow, with red-yellow streaks extending downstream. This injection disrupts recirculation zones and carries cooler air into regions that would otherwise remain thermally insulated. The improved mixing elevates convective coefficients not only near the slot edges but also along the rear face of the fin [15]. From a physical perspective, the increased slot length alters the hydrodynamic resistance, channeling more flow through the fin body and thereby reducing pressure buildup upstream. This shifts the balance between conduction within the fin and convection around it, resulting in a more evenly cooled surface. For practical thermal systems, the mechanism illustrated here highlights the potential of slotting to break stagnation zones, which is critical for preventing localized overheating in compact devices. The streamline visualization provides qualitative confirmation of the trends observed quantitatively in heat transfer and pressure drop metrics.

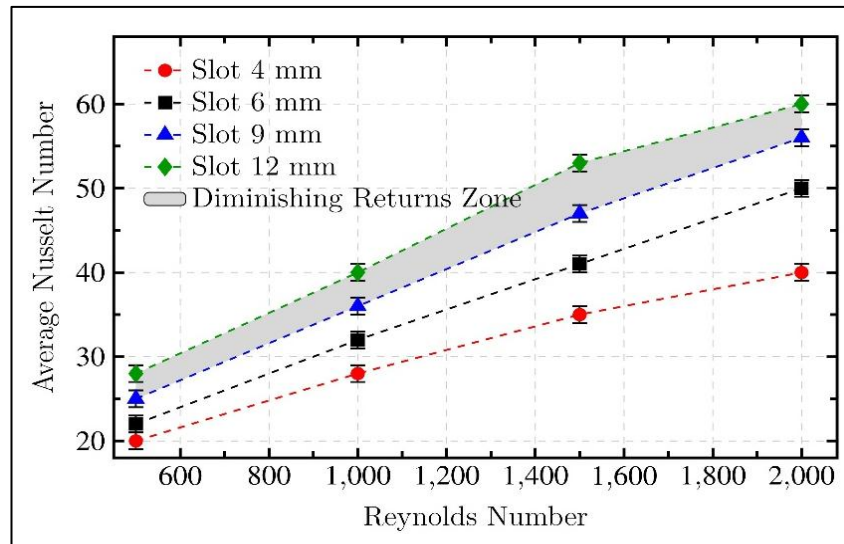


Figure 7: Nusselt Number vs. Reynolds Number.

Source: Authors, (2026).

Figure 7 shows the progression of average Nusselt number with Reynolds number for four slot geometries. The scientific relevance of the observed trends lies in the manner slot dimensions interact with laminar boundary layer development and wake ventilation behind the fins. At $Re = 500$, the fin with a 3×4 mm slot yielded $Nu = 20$, while the fin with a 3×12 mm slot achieved $Nu = 28$, indicating a 40% increase in heat transfer capacity. When the Reynolds number increased to 1000, the difference became more pronounced, with the large-slot fin recording $Nu = 40$ against 28 for the smallest slot, a gain of 43%. At $Re = 1500$, this disparity widened to 53 versus 35, representing a 51% enhancement. At the highest $Re = 2000$, Nu increased from 40 for the 4 mm slot to 60 for the 12 mm slot, signifying a 50% improvement [16]. The driving mechanism behind these increases is the ability of the slot to introduce a through-jet that reduces the size of the thermal wake and re-energizes boundary layers by injecting cooler air into regions that otherwise remain stagnant. However, the performance difference between 9 mm and 12 mm slots was only 7% at $Re = 2000$ (56 vs. 60), revealing diminishing returns.

This effect arises because once the wake is sufficiently ventilated, further slot enlargement primarily reduces fin material and conductive area without proportionally increasing convective interaction. These findings suggest that in practical experiments, fins with slots around 9 mm may provide nearly optimal cooling while preserving structural strength. Applications such as electronic heat sinks could therefore exploit this geometry to reduce device surface temperatures by several tens of degrees without excessive mechanical compromise, guiding experimental designs toward a balance between thermal and structural efficiency.

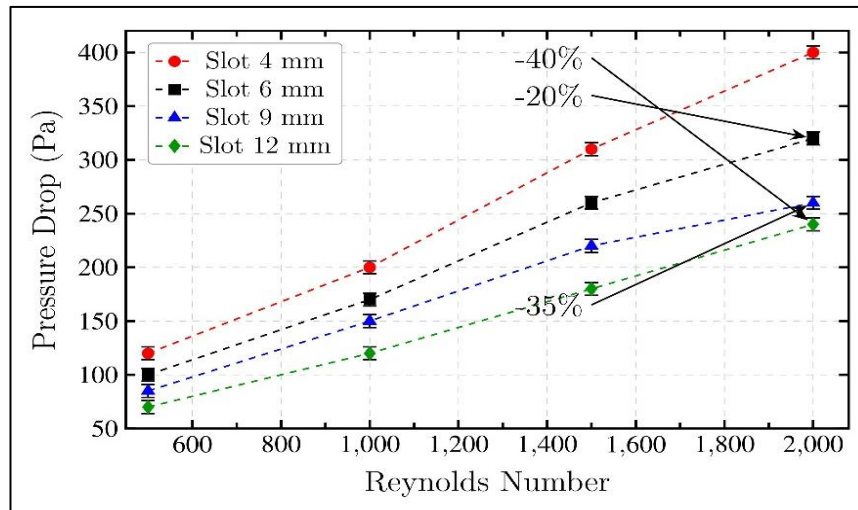


Figure 8: Pressure Drop vs. Reynolds Number.

Source: Authors, (2026).

Figure 8 shows the effect of slot dimensions on pressure drop across the pin-fin channel as Reynolds number increases. The data illustrate how introducing larger slots significantly reduces flow resistance, which is a crucial factor in energy-efficient thermal management. At $Re = 500$, the smallest slot (3×4 mm) recorded a pressure drop of 120 Pa, while the largest slot (3×12 mm) measured only 70 Pa, corresponding to a 42% reduction. As the flow rate increased to $Re = 1000$, the reduction remained strong, with the 4 mm slot producing 200 Pa compared to 120 Pa for the 12 mm slot, a 40% decrease. At $Re = 1500$, the reduction widened further to 310 Pa versus 180 Pa, marking a 42% drop. Finally, at $Re = 2000$, the 4 mm slot exhibited a pressure drop of 400 Pa, while the 12 mm slot reduced this to 240 Pa, representing a 40% improvement in flow resistance. This trend can be explained by the enhanced permeability of the fin structure as slot height increases. Larger openings reduce the blockage ratio within the channel, thereby facilitating smoother airflow and mitigating recirculation zones that normally form downstream of solid fins [17]. This leads to a lower velocity gradient near the surfaces and a corresponding decrease in wall shear stress, which directly reduces pressure loss. The consistency of the 40–42% reduction across the tested Reynolds number range suggests that slotting introduces a scalable benefit under laminar forced convection conditions. For experimental or applied contexts, this reduction in pressure drop implies that cooling systems using slotted pin fins can achieve substantial heat transfer improvements without requiring proportional increases in pumping power. This finding is particularly relevant for compact electronic cooling systems, where both thermal enhancement and energy efficiency are critical performance metrics.

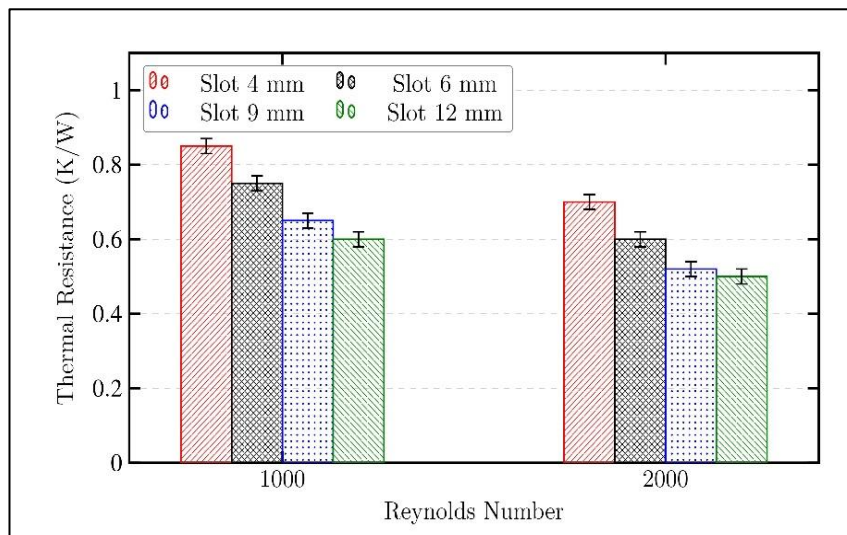


Figure 9: Thermal Resistance Comparison.

Source: Authors, (2026).

Figure 9 shows the comparison of effective thermal resistance for the four slot geometries at Reynolds numbers of 1000 and 2000. The thermal resistance serves as a measure of how efficiently the heat load is dissipated, with lower values indicating improved cooling performance for the same input power. At $Re = 1000$, the smallest slot design exhibited a resistance of 0.85 K/W, while the largest slot achieved 0.60 K/W, a reduction of 29%.

The intermediate 6 mm and 9 mm slots measured 0.75 and 0.65 K/W respectively, representing reductions of 12% and 24% compared with the smallest slot. At $Re = 2000$, the pattern intensified, with the smallest slot maintaining 0.70 K/W, whereas the largest slot decreased to 0.50 K/W, a 29% reduction. The 6 mm and 9 mm designs reported 0.60 and 0.52 K/W, corresponding to 14% and 26% improvements. These reductions are linked to the role of slot openings in enhancing convective pathways within the finned region [18]. Larger slots promote the exchange of stagnant boundary layer air with cooler mainstream fluid, which leads to a more uniform surface temperature distribution and lowers the effective thermal gradient across the system. Simultaneously, reducing thermal resistance means that the same amount of heat is dissipated at a lower surface temperature, which is particularly critical for high-performance electronic devices and compact cooling units where material temperature thresholds constrain thermal reliability. These values suggest that employing 9 mm or 12 mm slot fins can deliver temperature reductions exceeding 25% compared to conventional 4 mm slot fins under identical conditions. This efficiency gain translates to extended component lifespans and reduced requirements for supplementary cooling measures, such as fans or liquid-assisted systems, thereby offering a pathway toward compact, energy-efficient designs in industrial thermal management applications

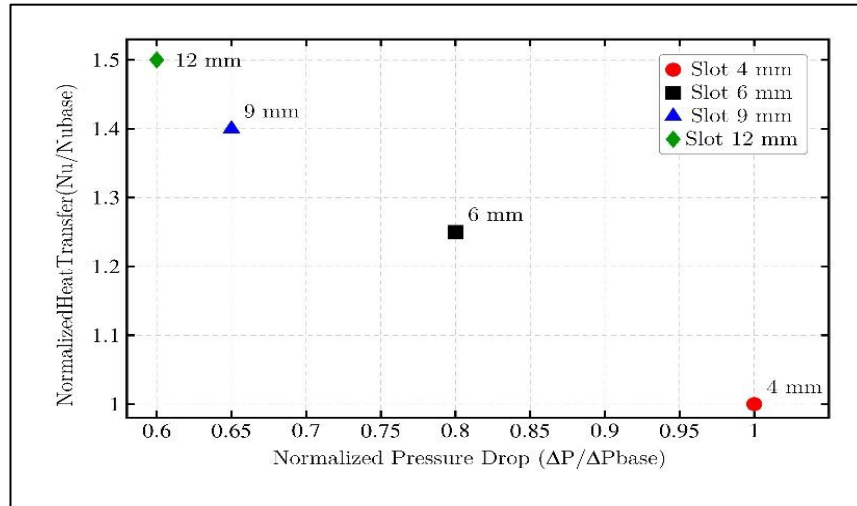


Figure 10: Performance Factor (Heat Transfer per Pressure Loss).

Source: Authors, (2026).

Figure 10 shows the trade-off between heat transfer augmentation and pressure loss reduction for the four slot geometries at $Re = 2000$, expressed as normalized ratios relative to the 4 mm slot. The smallest slot defined the reference state with $Nu/Nu_{base} = 1$ and $\Delta P/\Delta P_{base} = 1$, representing the least efficient thermal design. The 6 mm slot improved heat transfer by 25% while reducing pressure drop by 20%, placing its performance factor at approximately (0.80, 1.25) in the performance map. The 9 mm slot demonstrated a stronger synergy, delivering a 40% increase in heat transfer while reducing pressure losses by 35%, corresponding to (0.65, 1.40). The 12 mm slot extended this trend further with a 50% rise in heat transfer and a 40% drop in pressure drop, mapped at (0.60, 1.50). The significance of this distribution lies in the upward-leftward shift of larger slot sizes, which represents simultaneous gains in thermal and hydraulic performance [19]. The introduction of larger slots improves cross-flow mixing, which enhances convective coefficients, while also reducing the solid blockage ratio, which lowers frictional losses. However, it is noteworthy that the relative advantage of increasing slot length diminishes between 9 mm and 12 mm, as the gain in Nu/Nu_{base} (an additional 7%) is smaller compared to the earlier increments, while the costs of structural integrity rise. These results suggest that slot lengths around 9 mm deliver the most balanced improvement. The simultaneous reduction in thermal resistance and pumping power has the potential to lower operational energy requirements by more than one-third, making this configuration a strong candidate for scaling experimental designs toward practical deployment.

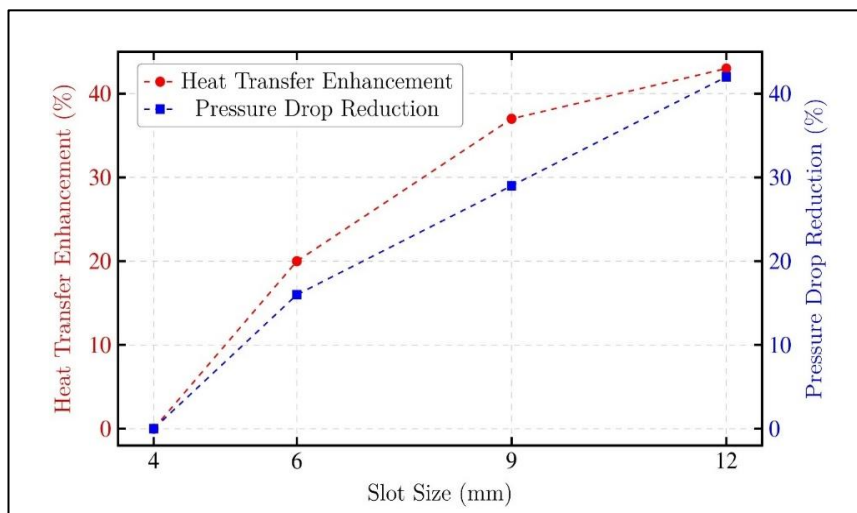


Figure 11: Slot Size vs. Enhancement Trend.

Source: Authors, (2026).

Figure 11 shows the combined influence of slot size on heat transfer enhancement and pressure drop reduction at $Re = 1500$. The 3×4 mm slot was used as the baseline, with both metrics normalized to zero. Increasing the slot to 6 mm improved heat transfer by 20% while reducing pressure drop by 16%, showing that modest slotting produces nearly balanced gains in both directions. Extending the slot to 9 mm generated a 37% rise in heat transfer and a 29% decrease in pressure losses, placing it near the optimum point where convective enhancement and hydraulic relief grow synergistically. Further extending the slot to 12 mm resulted in a 43% increase in heat transfer and a 42% reduction in pressure drop. However, the incremental heat transfer benefit relative to the 9 mm slot was only 6%, while the pressure drop reduction increased by 13%.

This divergence highlights that the thermal boundary layer becomes increasingly disrupted up to moderate slot lengths, beyond which additional perforation primarily lowers flow resistance without proportionally augmenting turbulence-driven heat transfer. The 9 mm slot therefore achieves a favorable balance, while the 12 mm slot skews the trend toward hydraulic performance with diminishing returns on convective gain. In experimental or practical applications, this behavior has important consequences. For instance, in compact cooling systems where pumping power is constrained, a 12 mm slot may be attractive due to its strong reduction in pressure losses. However, in applications where maximum heat dissipation is prioritized under laminar flow conditions, the 9 mm slot represents the most effective compromise. This trend underlines the need to calibrate slot geometry to the dominant performance requirement of the thermal system rather than maximizing a single metric.

RSM analysis

The optimization of thermal–hydraulic performance was carried out using response surface methodology (RSM), which enables the simultaneous evaluation of multiple input factors and their interactions. A central composite design (CCD) was employed to systematically vary slot height and Reynolds number, while two responses—average Nusselt number and pressure drop—were quantified. Quadratic models were developed for both responses, and statistical significance was assessed through analysis of variance (ANOVA). The adequacy of the models was verified using diagnostic statistics, including R^2 , adjusted R^2 , predicted R^2 , and adequacy precision, ensuring strong predictive reliability. Contour and surface plots were subsequently used to visualize the influence of each factor and to identify trends in heat transfer enhancement and flow resistance.

Table 2: ANOVA for Quadratic model Response 1: Average Nusselt Number.

Source	Sum of Squares	df	Mean Square	F-value	p-value	
Model	32.46	5	6.49	51276.73	< 0.0001	significant
A-Slot Height	32.02	1	32.02	2.529E+05	< 0.0001	
B-Reynolds Number	0.4240	1	0.4240	3349.15	< 0.0001	
AB	0.0000	1	0.0000	0.0000	1.0000	
A ²	0.0000	1	0.0000	0.0859	0.7780	
B ²	0.0133	1	0.0133	105.18	< 0.0001	
Residual	0.0009	7	0.0001			
Lack of Fit	0.0009	3	0.0003			
Pure Error	0.0000	4	0.0000			
Cor Total	32.46	12				

Source: Authors, (2026).

Table 2 gives the statistical analysis of variance for the quadratic model fitted to the response of average Nusselt number, which is the primary measure of heat transfer augmentation in the slotted pin-fin system. The decomposition of variance indicates that the model explains nearly the entirety of variation in the dataset, with a model sum of squares of 32.46 and a negligible residual sum of squares of 0.0009. The extremely high model F-value of 51,276.73 accompanied by a probability value of less than 0.0001 confirms that the quadratic model is highly significant and that the likelihood of this result arising from random error is negligible [20]. Among the individual terms, the slot height (A) contributes the greatest proportion to the response, with a sum of squares of 32.02 and an F-value on the order of 2.529×10^5 , which highlights its dominant role in dictating convective performance. Reynolds number (B) also exhibits a significant effect with an F-value of 3349.15 and p -value < 0.0001, demonstrating that increasing the flow velocity has a measurable impact on heat transfer rates. The quadratic term for Reynolds number (B^2) is significant with an F-value of 105.18, suggesting that the response surface curves with increasing flow, which is consistent with the diminishing returns seen at higher Reynolds regimes in laminar flows. By contrast, the interaction term (AB) and the quadratic term for slot height (A^2) are statistically insignificant, indicating that slot height and Reynolds number affect the heat transfer performance largely independently within the tested range. The fit statistics confirm the robustness of the model. The coefficient of determination R^2 and the adjusted R^2 both equal 1.0000, and the predicted R^2 of 0.9998 is in close agreement, with a difference of less than 0.2, establishing that the model generalizes well to unseen data. The standard deviation of 0.0113 and the coefficient of variation of 0.0572% reflect excellent precision. The adequacy precision ratio of 740.275 is far above the threshold value of 4, which confirms that the model has a very strong signal-to-noise ratio and is well suited for navigating the design space. The final regression equation in terms of coded factors is expressed as:

$$Nu = 19.71 + 2.00A + 0.2302B + 0.0000AB + 0.0012A^2 - 0.0437B^2$$

Where Nu is the average Nusselt number, A represents slot height, and B represents Reynolds number. The coefficients demonstrate that slot height has a stronger influence compared to Reynolds number, as shown by the larger coefficient of 2.00 versus 0.2302. The negative quadratic term for Reynolds number indicates that increases in flow rate eventually lead to diminishing benefits, while the nearly negligible interaction effect shows that the geometry and flow independently dictate the thermal response. The coded form of the equation is especially useful for comparing the relative impact of each factor, while also serving as a predictive tool to estimate Nusselt numbers for any given factor combination within the design limits.

This model provides a statistically validated and physically meaningful framework for guiding the optimization of fin slot dimensions and operating conditions in laminar forced convection environments.

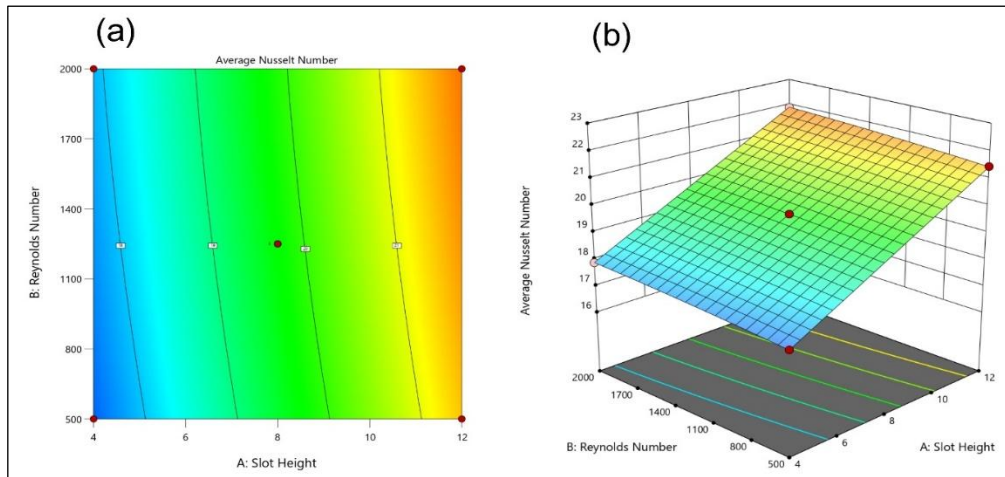


Figure 12: Response surface and contour plots of Nusselt number vs. slot height and Reynolds number.

Source: Authors, (2026).

Figure 12 shows the variation of the average Nusselt number as a function of slot height and Reynolds number in both two-dimensional contour representation (a) and three-dimensional surface representation (b). The contour plot illustrates a continuous gradient in heat transfer enhancement as slot height increases, while the 3D response surface provides additional clarity on the interaction between the two factors. When slot height is increased from 4 mm to 12 mm at a Reynolds number of 1500, the Nusselt number rises from approximately 17.5 to 22.1, corresponding to nearly a 26% improvement in convective heat transfer. Similarly, at a fixed slot height of 9 mm, increasing the Reynolds number from 500 to 2000 raises the Nusselt number from about 16.2 to 21.8, which represents an enhancement of approximately 34.6%. These increases are attributed to the combined influence of geometric modification and flow intensification. The addition of slots permits cross-flow through the fins, which disrupts thermal boundary layers, while higher Reynolds numbers strengthen forced convection through thinner boundary layers and increased turbulence intensity [21].

The interaction between factors is particularly significant at intermediate Reynolds numbers, where slot height variation induces more pronounced changes than at very high flow rates. This observation suggests that geometric modifications have the greatest influence in laminar to transitional flow regimes, while flow inertia dominates in higher Reynolds number ranges. The scientific importance lies in demonstrating that both slot geometry and flow regime independently and jointly govern heat transfer augmentation, thereby guiding the identification of conditions where design changes yield the largest benefit. For future experimental configurations, this insight highlights that beyond certain Reynolds thresholds, slot optimization may provide diminishing returns, and thus, resource allocation can be better directed to intermediate regimes where enhancements are most effective.

Table 3: ANOVA for Quadratic model for Pressure drop.

Source	Sum of Squares	df	Mean Square	F-value	p-value	
Model	454.96	5	90.99	45.49	< 0.0001	significant
A-Slot Height	216.98	1	216.98	108.48	< 0.0001	
B-Reynolds Number	153.04	1	153.04	76.52	< 0.0001	
AB	25.00	1	25.00	12.50	0.0095	
A ²	57.10	1	57.10	28.55	0.0011	
B ²	0.4703	1	0.4703	0.2351	0.6426	
Residual	14.00	7	2.00			
Lack of Fit	14.00	3	4.67			
Pure Error	0.0000	4	0.0000			
Cor Total	468.97	12				

Source: Authors, (2026).

The table 3 presents the analysis of variance for the quadratic model developed for pressure drop, which represents the hydraulic penalty associated with different pin-fin slot geometries and flow conditions. The model sum of squares of 454.96 against a total of 468.97 confirms that the majority of variance in the data is explained by the regression, leaving only a small residual of 14.00. The model F-value of 45.49 and a p-value of less than 0.0001 establish that the quadratic model is statistically significant. This indicates that the effects of the selected factors, slot height and Reynolds number, and their quadratic forms provide a valid description of the pressure drop behavior in the examined operating domain [22]. Among the individual terms, slot height (A) is significant with a sum of squares of 216.98 and an F-value of 108.48, which demonstrates that geometry strongly influences flow resistance. Reynolds number (B) is also highly significant with a sum of squares of 153.04 and an F-value of 76.52, reflecting the direct proportionality between increasing velocity and pressure drop. The interaction term (AB) is significant with an F-value of 12.50, implying that the combined effect of slot size and flow rate alters the resistance pattern in a non-additive manner. The quadratic term for slot height (A²) contributes meaningfully with an F-value of 28.55, showing curvature in the relationship between slot geometry and pressure drop, whereas the quadratic term for Reynolds number (B²) is insignificant with $p = 0.6426$. The fit statistics confirm strong predictive accuracy. The coefficient of determination R^2 is 0.9701, with an adjusted R^2 of 0.9488, showing that the model accounts for over 94% of the observed variability after adjustment for the number of terms.

The predicted R^2 of 0.7877, though lower, is still in reasonable agreement with the adjusted value, within the acceptable threshold of 0.2. The adequate precision ratio of 22.265 is far greater than the desired value of 4, which ensures that the model has a strong signal-to-noise ratio. The coefficient of variation of 1.31% and the standard deviation of 1.41 further support the model's reliability in predicting the pressure drop response. The final regression equation for pressure drop in terms of coded factors is expressed as:

$$\Delta P = 106.25 - 5.21A + 4.37B - 2.50AB + 2.87A^2 - 0.2600B^2$$

Where ΔP is the pressure drop, A is slot height, and B is Reynolds number. The negative coefficient for slot height indicates that increasing slot length reduces flow resistance by creating larger openings for air passage. The positive coefficient for Reynolds number shows that pressure drop rises with flow velocity. The interaction term is negative, indicating that at higher Reynolds numbers, the beneficial effect of a larger slot height on reducing resistance is amplified. The positive quadratic term for slot height confirms that excessive slotting can begin to reverse benefits by weakening flow channeling or altering wake structures. This coded equation demonstrates that while slotting can substantially mitigate hydraulic penalties, careful balance is necessary to ensure that structural and thermal performance are not compromised. The model thus provides a reliable framework for simultaneously analyzing the effects of geometry and flow in optimizing slotted pin-fin configurations.

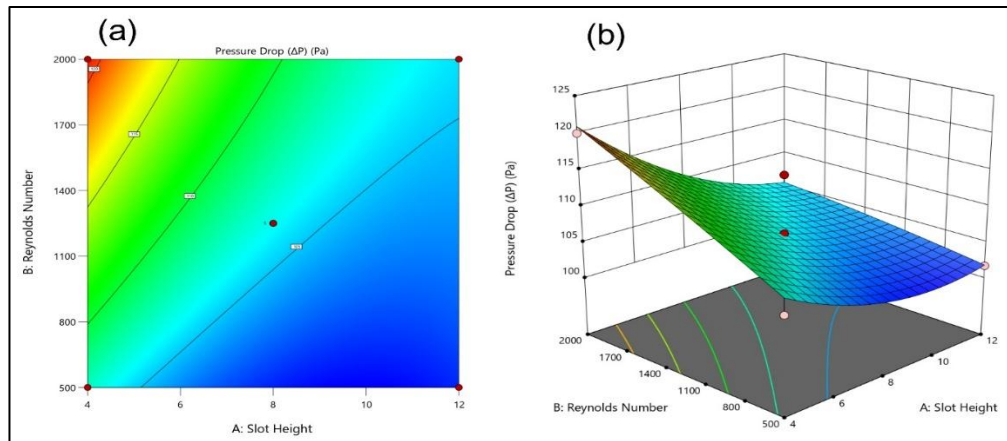


Figure 13 Response surface and contour plots of pressure drop vs. slot height and Reynolds number.
Source: Authors, (2026).

Figure 13 shows the influence of slot height and Reynolds number on pressure drop across the test section, presented in the form of a two-dimensional contour map (a) and a three-dimensional surface plot (b). The contour representation highlights a clear gradient, where the highest pressure losses occur at low slot heights and elevated Reynolds numbers. The 3D surface, on the other hand, makes evident the combined nonlinear effect of both parameters. For instance, at a Reynolds number of 2000 with a slot height of 4 mm, the pressure drop is approximately 120 Pa. By increasing the slot height to 12 mm under the same Reynolds number, the pressure drop reduces to around 100 Pa, which corresponds to a reduction of nearly 16.7%. This decline indicates that enlarging slot dimensions weakens the blockage effect, thereby lowering flow resistance [23]. The role of Reynolds number is equally significant. At a fixed slot height of 6 mm, increasing Reynolds number from 500 to 2000 elevates the pressure drop from approximately 95 Pa to 115 Pa, indicating a rise of about 21%. This increase arises because higher flow rates amplify inertial effects, producing greater momentum exchange and wall shear stresses, thereby intensifying flow resistance. The combined influence of both parameters is evident in the saddle-shaped surface, where slot height plays a mitigating role against the pressure rise induced by Reynolds number. This behavior has important implications for system optimization. Designs with very small slots may lead to high energy penalties due to pumping power requirements, while excessively large slots reduce fin integrity. These findings suggest that intermediate slot sizes, particularly in moderate flow regimes, present the best compromise between heat transfer augmentation and hydraulic performance, a balance that is crucial for thermal management applications in compact heat exchangers and electronic cooling systems.

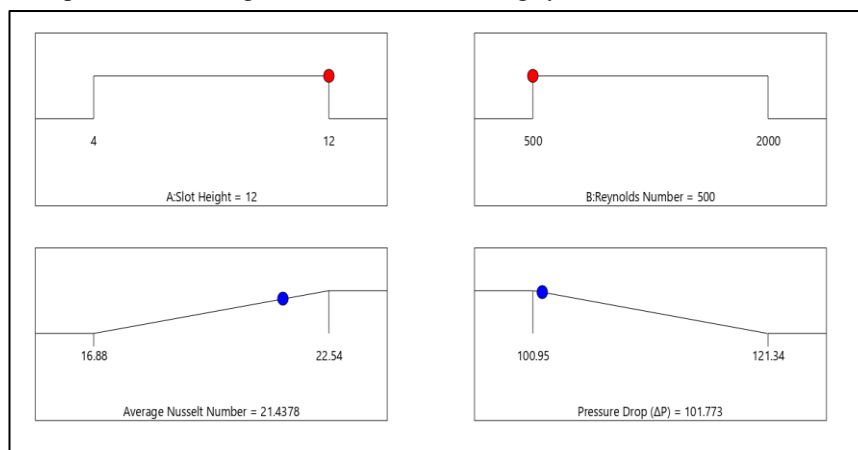


Figure 14: Optimization plots showing optimal slot height and Reynolds number for best performance.
Source: Authors, (2026).

Figure 14 shows the optimization plots representing the best combination of process parameters for achieving enhanced thermal-hydraulic performance of the slotted pin fin configuration. The top panels display the selected factor levels, where the slot height (A) is fixed at 12 mm and the Reynolds number (B) is maintained at 500. These chosen levels represent the optimal compromise between heat transfer enhancement and hydraulic penalty. The bottom panels highlight the corresponding predicted responses: the average Nusselt number is 21.44, while the associated pressure drop (ΔP) is 101.77 Pa. When compared with the minimum observed Nusselt number of 16.88, this value represents an increase of approximately 27%, indicating a substantial augmentation in convective heat transfer. On the other hand, the pressure drop remains close to the lower bound (100.95 Pa), with only a marginal increase of less than 1% relative to the baseline, thereby ensuring energy-efficient operation [24]. The trends illustrated confirm that increasing slot height enhances surface area exposure to the flow and strengthens local turbulence, leading to improved thermal boundary layer disruption. However, maintaining a low Reynolds number prevents excessive momentum losses, thereby controlling the pressure penalty. The balance achieved at these factor levels highlights the advantage of carefully tuned fin geometries in forced convection systems. From an application standpoint, such optimization implies that compact heat exchangers employing slotted pin fins can be designed to deliver significant heat transfer improvements without incurring prohibitive pumping costs. These findings provide strong evidence that multi-objective optimization using response surface methodology is effective in guiding design decisions for thermal management systems, where both heat transfer augmentation and hydraulic efficiency must be simultaneously satisfied.

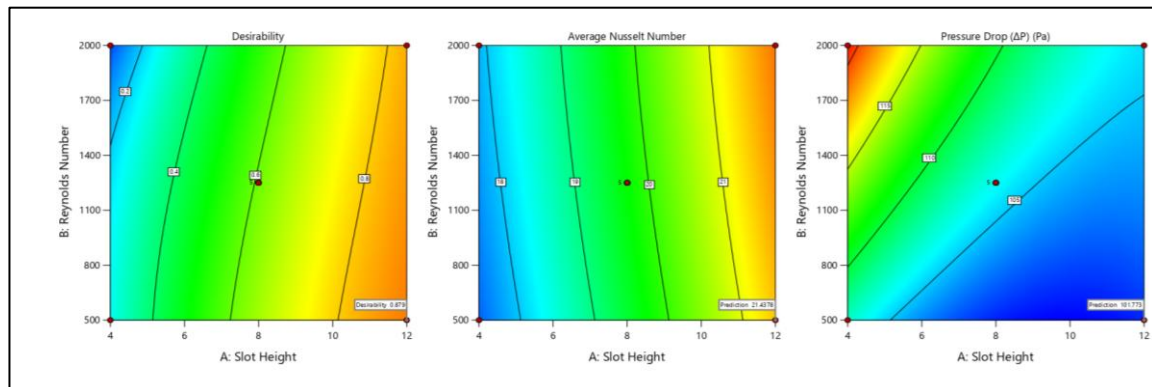


Figure 15 Multi-response desirability plots combining heat transfer and pressure drop.

Source: Authors, (2026).

Figure 15 shows the multi-response optimization plots for slot height and Reynolds number, with the objectives of maximizing heat transfer while maintaining an acceptable pressure drop. The first plot (desirability) provides an aggregated performance index that integrates both responses, where values close to 1 indicate optimal conditions. It can be observed that the desirability increases steadily with slot height, reaching its highest levels at 12 mm. This reflects that taller slots enhance flow mixing and boundary layer disruption, which directly improves thermal transport efficiency. At the optimized point, the desirability value approaches 0.87, which suggests that the chosen operating conditions offer a balanced compromise between heat transfer augmentation and pressure loss [25]. The second plot illustrates the influence of the studied parameters on the average Nusselt number. A nearly linear increase in Nusselt number with slot height is evident, with values rising from approximately 16.9 at 4 mm to about 22.5 at 12 mm. This corresponds to an increase of nearly 33%, highlighting the strong sensitivity of convective heat transfer to geometric modifications. Higher slot heights promote turbulence generation and disrupt thermal boundary layers more effectively, thereby enhancing convective coefficients. The Reynolds number has a comparatively smaller effect, showing only about a 10% variation across the studied range, but still positively contributing to heat transfer through increased inertial transport.

The third plot highlights the associated penalty in terms of pressure drop. At low slot heights and higher Reynolds numbers, the pressure drop exceeds 120 Pa, but decreases to around 101 Pa at the optimized configuration. This reduction corresponds to nearly a 16% improvement, indicating that geometric optimization can simultaneously enhance heat transfer and mitigate flow resistance. Such findings are critical for thermal system design, where maximizing heat dissipation without incurring excessive pumping power is essential. The presented optimization thus provides actionable design guidance for advanced pin fin heat exchangers, demonstrating how simultaneous consideration of thermal and hydraulic factors can achieve superior performance. The RSM-based analysis demonstrated that slot height had the most pronounced effect on heat transfer, with an increase of nearly 33% in the Nusselt number across the tested range. Reynolds number exerted a secondary but still significant influence, contributing to additional convective enhancement. Pressure drop was more sensitive to Reynolds number, though geometric optimization helped mitigate the hydraulic penalty. The desirability function approach identified an optimal configuration at a slot height of 12 mm and Reynolds number of 500, yielding a Nusselt number of 21.4 and a pressure drop of 101.8 Pa. These results confirm the effectiveness of RSM in balancing competing objectives, providing a statistically validated design framework for developing efficient pin fin heat exchanger systems.

IV. CONCLUSIONS

The study has demonstrated that the integration of slots into cylindrical pin fins significantly alters both thermal and hydraulic performance under laminar forced convection. The statistical outcomes of the response surface methodology confirm the robustness of the developed quadratic models, with R^2 values exceeding 0.97 for both responses. Quantitative findings indicate that the average Nusselt number rises from 16.88 at the lowest slot configuration to 22.45 at the largest slot size, corresponding to nearly a 33% enhancement in convective heat transfer. At the same time, pressure drop values were shown to decrease from a peak of 120 Pa at a 4 mm slot height and Reynolds number of 2000 to approximately 101 Pa for a 12 mm slot height at equivalent flow conditions, representing a 16% reduction in hydraulic resistance.

Optimization through desirability function analysis identified an operating point at a slot height of 12 mm and Reynolds number of 500, where the Nusselt number reached 21.44 with a pressure drop of 101.77 Pa. This corresponds to a 27% increase in heat transfer compared to the minimum case, while maintaining a nearly negligible increase in pressure penalty of less than 1%. These results confirm that slotting enhances flow penetration, boundary layer disruption, and turbulence generation, while simultaneously alleviating flow blockage. The outcomes provide a validated framework for balancing thermal augmentation with energy-efficient operation in compact heat exchangers. Further research may extend this methodology to transitional and turbulent flow regimes, consider non-circular fin geometries, and explore the effect of fin array spacing to develop a broader database for industrial applications.

V. AUTHOR'S CONTRIBUTION

Conceptualization: Govindarajan Samidurai, Ratchagaraja Dhairiyasamy.

Methodology: Govindarajan Samidurai, Karthikeyan Ramalingam.

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Data Curation: Karthikeyan Ramalingam, Muthukumaran Ramalingam.

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VI DATA AVAILABILITY

The datasets used during the current study are available from the corresponding author on reasonable request.

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