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OPTIMIZING HEAT TRANSFER: COMPARATIVE ANALYSIS OF VARIOUS FIN CONFIGURATIONS

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Abstract
Using heat inputs of 20 V, 35 V, 50 V, 65 V, and 80 V, this study conducts trials to examine the heat transmission performance of different fin configurations. Seven configurations, including plain, semicircular notch, rectangular notch, triangular notch, inverted semicircular notch, inverted rectangular notch, and inverted triangular notch fins, were tested. Each configuration underwent fifteen experiments, repeated thrice across five different heat inputs, resulting in a total of 105 experiments. The results demonstrate that the triangular notch fin exhibits the maximum coefficient of heat transfer of 10.74 W/m^2K at 80 V, highlighting its superior heat enhancement capability. The findings emphasize the significant influence of fin geometry and orientation on heat transfer efficiency, with notched fins, particularly triangular ones, showing marked improvement at higher heat inputs and Grashof numbers. This research provides valuable insights into optimizing fin designs for enhanced thermal performance in heat transfer applications.

1. Introduction
Temperature differentials cause natural convection, which is vital in many engineering applications where effective heat transfer is necessary to improve system efficiency and preserve thermal stability. This phenomenon is important for academics and engineers to understand since it affects many different industries, such as electronics cooling, energy systems, and environmental control etc. Additionally, convective heat transfer mechanisms play a vital role in optimizing thermal management in diverse applications, including porous insulations, cooling revolving electric windings, irrigation systems, geothermal reservoirs, and oil and gas field exploration. Electronics utilizing natural convection heat transfer offer a sustainable cooling solution, reducing reliance on energy-intensive cooling systems powered by fossil fuels. By harnessing the inherent properties of airflow and temperature differentials, natural convection systems can greatly reduce the amount of energy used and emissions of greenhouse gases associated with traditional cooling methods. This shift towards eco-friendly cooling not only aligns with global sustainability goals but also contributes to cost savings and operational efficiency for businesses and industries.
One popular method for maximizing natural Heat transmission by natural convection is the utilization of fin arrangements extended surfaces, or fins, are used to increase heat dissipation between a solid surface and the fluid around it.[7] These fins are available in a variety of shapes such as rectangular, triangular, semicircular etc. and each one has special qualities related to thermal performance [1],[11],[15].The investigation of novel fin shapes and arrangements has emerged as a key area of focus in the search for improved heat transfer effectiveness.
Shape-change functionality is a critical innovation for natural convection heat transfer in fins because it allows fins to adapt to temperature changes and spatial constraints. In situations involving natural convection, these fins can increase efficiency by up to 50 %, primarily by maximizing airflow and surface area for heat transmission [7][9].Fins are strategically designed and arranged to maximize convective heat transfer processes and provide this efficiency boost. Heat dissipation from the system

is greatly increased, resulting in a more effective cooling process, by expanding the surface area accessible for heat exchange and encouraging airflow around the fins. Additionally, fins can be engineered to minimize resistance to airflow, further improving the overall efficiency of the system.
The study demonstrates that triangular fins made of aluminum alloy 6061 outperform other materials, validating theoretical assessments and offering valuable insights for enhancing heat transfer efficiency in academic research publications [1].The results of above triangular fins made of aluminum strongly provide a theoretical basis for the choice of material in our model. Since all our research system fins are made of aluminum, this insight underscores the effectiveness of aluminum in optimizing heat transfer, aligning with our experimental focus on various fin configurations. Three notched layouts—none, 20%, and 40% notches—were analyzed, revealing that heat transfer coefficients for 20% and 40% notched fins are reduced by 7% and 10%, respectively, hence it contextualizes the impact of different notch designs on heat transfer efficiency, aligning with our goal of optimizing fin performance in natural convection scenarios[2]. Under natural convection, the study compares Heat sinks with plate- and pin-fins operating under convection in nature with base plates arranged upwards. It finds that while plate-fin sinks dissipate heat more efficiently overall, pin-fin sinks do so more efficiently per unit mass because of a specific correlation. Hence, above mentioned findings is relevant to our model, which induce use of plate-fin sinks, as it highlights the overall heat dissipation efficiency of plate-fin designs and provides a comparative context for evaluating our experimental results [3]. The study found that fin height influences a rectangular plate fin array's natural convection heat transfer performance both computationally and practically, which encourage us to utilize this finding in our research as it highlights a crucial factor affecting heat transfer efficiency, most relevant to our study of various fin configurations [4]. For heat sinks with plate having pins that are half-round, a new thermal design showed notable improvements over conventional designs. Along with decreases in temperature at the base & resistance to heat of roughly 25.1 and 29 percent, respectively, the Nusselt value improved by 34.48 percent. [6], so authors conclusion is pertinent to our research as it showcases advancements in heat transfer efficiency, which informs our exploration of optimized cooling solutions in similar configurations. The investigation of heat transfer by natural convection in exchangers with finned tubes having square-shaped fins under various input powers (8.4 to 56.7 Watts) and fin spacing (5, 9, and 14 mm) reveals that natural convection contributes about 80% of the total input power, while radiation heat transfer accounts for about 10%. With increasing fin spacing, the average natural heat transfer coefficient rises and then falls, and an empirical connection is established to calculate the Nusselt figure, which ranges from 6.5 to 1,335 Rayleigh numbers, depending on fin spacing. This finding is essential for optimizing our model's fin spacing configurations [11]. Inverted trapezoidal fins demonstrate significantly higher heat transfer coefficients under natural convection—10% and 25% more than both rectangular and trapezoidal fin structures. These findings, supported by numerical simulations, illustrate temperature variations, reduced fin profiles, and lower airflow resistance, collectively enhancing overall heat transfer efficiency [13]. Roody Charles findings will be our exploration of inverted combinations, prompting our investigation into inverted rectangular, inverted semi-circular, and inverted triangular notch fins to achieve superior heat transfer rates in our study.
Experimental results comparing vertical rectangular fins with and without notches revealed that unnotched fins exhibited over 18% lower heat transfer rates compared to notched fins, and previous literature has predominantly emphasized the industrial significance of thick fins and wide fin spacing to address specific industrial requirements [14], underscoring the importance of notch designs in optimizing heat transfer efficiency and guiding our exploration of various notch configurations in our study. The findings from research paper [16] fin array with inverted hybrid square-semicircular notch demonstrated higher performance with 7.82 W/m^2K is the average coefficient of heat transfer as demonstrated by the use of SolidWorks Flow Simulation software. This represents an eight percent improvement over plain rectangular arrays. This computational study informs our research by demonstrating the potential benefits of innovative fin designs in enhancing heat transfer efficiency. In our research, we conducted experimental studies to validate and expand upon findings by Anjali Rai,

exploring various fin configurations: plain fins, rectangular notch fins, semicircular notch fins, triangular notch fins, inverted rectangular notch fins, inverted semicircular notch fins, and inverted triangular notch fins. This methodology allows us to apply and verify computational insights in practical heat transfer applications.

Our research work attempt is made to study and analyze experimental heat transfer enhancement passively using un-notched and notched plate fins. We carried out a thorough investigation in this work to ascertain the heat transfer coefficients for various fin shapes. Our goal is to fill in these research gaps and make significant contributions to the field of convectional heat transfer occurring naturally in fin arrays by means of thorough investigation and analysis.

2.Literature review –

MAYANK JAIN [1] employs parametric models in CREO Parametric 2.0 to study thermal efficiency across various fin shapes (triangles, circles, rectangles). They upgraded from aluminum alloy 204 to 6061 for enhanced thermal conductivity, optimizing parameters like cross-sectional area and length. Triangular fins in 6061 outperformed other materials, showing significant temperature reductions and higher heat transfer rates.

Sachin R. Pawar [2] examines how triangle notches affect the heat transfer via natural convection in horizontal fin arrangements, finding static areas which prevent warmth from dissipating in the bottom centre of the array. The study investigates three configurations - unnotched, 20% notched, and 40% notched—maintaining consistent fin weight. Results indicate a reduction in heat transfer coefficients with increased notching, showing decreases of 7% and 10% for 20% and 40% notched fins, respectively. Despite varying heat inputs, all arrays exhibit a consistent chimney flow pattern, suggesting limited effectiveness in enhancing heat transfer through notching.

Y Joo [3] contrasted base plates that were positioned vertically and assessed heat sinks with pin- and plate-fins utilising convection in nature. Depending on the application, heat sinks with plate fins demonstrated better overall heat transfer than heat sinks with pin fins. Their study introduced a novel correlation demonstrating enhanced the loss of heat per mass per unit heat sinks with pin.

Abbas Jassem Jubear [4] explored, using both computational modelling and real-world experimentation, the impact of fins height on the rectangle-shaped fins array's heat transfer by convection performance. In the experiment, fins with lengths of 300 millimetres and heights of ten millimetres, twenty-five millimetres and 45 millimetres were utilised. According to their research, fin height increased with rising Rayleigh Numbers, Nusselt Numbers and convection heat transfer coefficients; the greatest values were found at 45 mm.

S. Sadrabadi Haghighi [5] investigated how to improve heat transmission through natural convection in plate-based heat sinks by adding fins. Based on experimental data, their study demonstrated an empirical correlation between the Rayleigh Number, fin gap to proportion of height, the quantity of plates, and Nusselt number.

B. Freegah [6] created a new thermal design using half-round pins to raise plate-fin heat sinks' heat efficiency. Their symmetry-enhanced half-round hollow pins and waved half-round pin shapes improved heat dissipation, according to their analysis, which was based on numerical simulations. These designs increased the Nusselt number by 34.48% while decreasing base temperature and resistance to heat by roughly Twenty-five percent and twenty-nine percent respectively. Potential uses for improving electronic equipment heat management are highlighted by the research.

Ali Daliran , Yahya Ajabshirchi[7] examine how adding rectangular fins to flat plate air collectors improves thermal performance by creating airflow turbulence. According to their research, fin attachment raises outlet air temperatures and decreases overall heat loss by increasing heat transfer coefficients over the absorber plate while also lowering Nusselt numbers. The study highlights the significant impact of fin attachment on collector efficiency and validates the numerical model through experimental comparisons.

Anurag Dahiya Singh [8] evaluated three manifold configurations for Microchannel Heat Sinks (MCHS) using CFD analysis. The greater performance of the Divergent Convergent (DC) manifold was demonstrated by its improved Nusselt numbers and 10% higher heat transfer coefficient when compared to the Rectangular and Rectangular with Semi-Circular designs.
Mehran Ahmadi [9] investigated the constant external natural convection heat transfer of steeply positioned rectangular intermittent fins using a combination of numerical calculations and experimental tests. According to their research, adding breaks to the fins greatly increases thermal efficiency; the ideal break length has been found. This compact correlation aids in improving natural heat transfer in electronics, power electronics, and telecom applications.
V. I. Terekhov, A. L. Ekaid, and K. F. Yassin [10] examined laminar natural convection across isothermal vertical plates over a range of Rayleigh numbers ($10–10^5$) and channel lengths ($1–500$). The study found that while convective draft and Reynolds number increase with aspect ratio, mean heat transfer decreases, with a modified Rayleigh number providing broader data alignment with established correlations.
Mehdi Karami, Mahmood Yaghoubi, and AmirrezaKeyhani [11] examined natural convective Heat transfer in heat exchangers with square fins that are finned spaced at different input voltages (8.4 to 56.7 watts) and square fin spacings (5, 9, and 14 mm). They discovered that around 80% of heat transfer occurs through natural convection, and they constructed an empirical relationship to predict the Nusselt value with regard to fin spacing in the range of Rayleigh numbers go from 6.5 to 1,335.
R. C. Chikurde [12] examines the transport of heat across horizontal rectangular fin arrays with different styles of knurling. An 11% increase in heat transmission is demonstrated by full-height knurled fins with 10 mm spacing and 0.5 mm depth; knurling depth and spacing have a major impact on performance. Overall, knurled fins can enhance heat transfer by up to 15% compared to plain fins, primarily by creating turbulence that disrupts laminar boundary layers.
Roody Charles [13] The effectiveness of heat sinks using trapezoidal, rectangular, and inverted trapezoidal fin shapes under natural convection is examined in this work. According to the results, inverted trapezoidal fin perform better than other designs, outperforming rectangular and trapezoidal layouts by 10% and 25%, respectively, in terms of heat transfer coefficient. Numerical simulations support these findings, highlighting larger temperature differences, lower fin profiles, and reduced airflow drag. These factors collectively contribute to superior heat transfer efficiency in the inverted trapezoidal geometry.
Kharche and Farkade[14] Results of experiments with and without notches on a vertical rectangular finned array are reported. It was discovered that the unnotched fins' heat transmission rate was almost 18% lower than that of the notched fins.
Manikanda PRABU N., Venkateshwaran P [15] Outlined research, both computational and experimental, on three different forms of pin-fin profiles: stepped, elliptical, and rectangular. The experiments were performed in vacuum environment so as to consider only radiation heat transfer. Thermal performance of elliptical shaped pin fins was found to be the best.
Anjali Rai[16] carried out a numerical study to investigate different vertical plate fin array configurations using SolidWorks Flow Simulation. They found that the mean heat transfer rate of the inverted hybrid square-semicircular notched fin arrays is 8% higher than that of regular rectangular arrays, at 7.82 W/m^2K. Additionally, inverted plate fin arrays consistently outperform non-inverted designs, while hybrid arrays show superior heat transfer coefficients compared to uniform notched fins. This research highlights the efficacy of computational methods in reducing experimental costs and underscores the importance of considering radiative heat loss in computational studies relative to experimental results.

An overview of current fin design efforts that are susceptible to natural convection -

Paper	**Experimental/numerical**	**Orientation**	**Fin**	**Height**	**Thikness**	**Fin Sap**	**Material**	**Fins with Good**

				(mm)	**(mm)**	**acing**		**heat_transfer_rate**
[1]	Numerically & analytically	Vertical	Rectangle Fin	12	1.5	4.8	Aluminum	Rectangular, Triangular, Circular
			Triangular Fin	12	1.5	4.8	Aluminum	
			Circular Fin	12	1.5	4.8	Aluminum	
[2]	Numerically & analytically	Vertical	Rectangle Fin	85	1.5	12	Aluminum	Rectangular Fin
			Fin with 20% of the notch	102	1.5	12	Aluminum	
			Fin with 40 % of the notch	119	1.5	12	Aluminum	
[3]	Numerically & analytically	Vertical	Heat sink with plate fins	30	1	NA	Aluminum	Plate Fin heat sink
			Heat sink with pin fins	30	1	NA	Aluminum	
[4]	Experimentally, Numerically & analytically	Vertical	Rectangular Fin Array	10	4	10	Aluminum	Rectangular Fin Array with 45 mm
				25	4	10	Aluminum	
				45	4	10	Aluminum	
[5]	Experimentally &Numerically	Vertical	Pin-fin Plate	45	4	5 - 12	Aluminum	Cubic plate pin-fin having 8.5 mm widths.
			Cubic plate pin-fin	45	4	5 - 12	Aluminum	
			Heat sinks with plate fins but no fillet profile	20	1	3.3	Aluminum	
			heat sinks featuring fillet and plate-fin profiles	24.5	1	3.3	Aluminum	
			Symmetric half-round pins arranged vertically	21.5	1	3.3	Aluminum	

[6]	CFD Experimentally & Numerically	Vertical	Positioning corrugated half-round pins vertically	21.5	1	3.3	Aluminum	Positioning corrugated half-round pins vertically
			Placing symmetrical half-round pins horizontally	21.5	1	3.3	Aluminum	
			Half-round, horizontally oriented corrugated pins	21.5	1	3.3	Aluminum	
[8]	CFD Experimentally & Numerically	Vertical	Rectangular	24.5	0.5	0.5	Copper while the Cover of Acrylic Sheet	Divergent Convergent
			Rectangle with Semi-Circular	24.5	0.5	0.5		
			Divergent Convergent	24.5	0.5	0.5		
[11]	Experimentally &Numerically	Vertical	Square Fins	50	2	5	Aluminum	Square Fins with 9 mm fin spacing
						9	Aluminum	
						14	Aluminum	
[13]	Experimentally, analytically &Numerically	Vertical	Rectangular Fins	30	1.375	3.425	Aluminum	Inverted Trapezoidal fins
			Trapezoidal Fins	30	1.375	3.425	Aluminum	
			Inverted Trapezoidal fins	30	1.375	3.425	Aluminum	

Table number 1

Despite extensive research in this area, several research gaps remain:

- Lack of study on different types of geometrical fins and inverted fins.
- Most of the papers contained experimentation using computational fluid dynamics (CFD) methods, with limited experimental validation.
- The results reported in previous papers were primarily focused on industrial applications, often featuring thick fins and large distances between fins.

Lack of study on the heat losses occurring between experiments, which was not adequately addressed in the literature.

3.Experimental Setup –

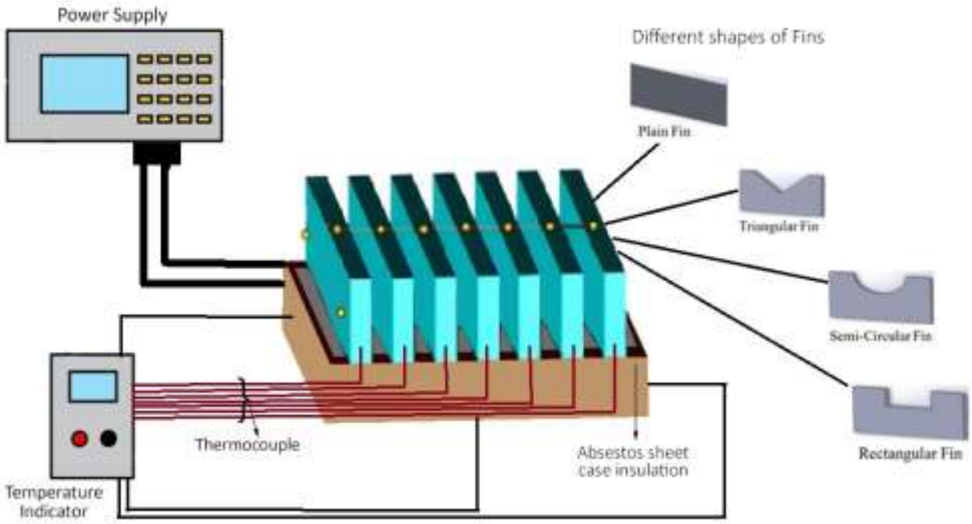

Fig Number 1: Layout of Experimental Setup

4.Experimental Model

4.1.BasePlate

Experimental setup have procured a rectangular aluminum base plate measuring 142 mm × 80 mm with a thickness of 25 mm. This size was selected to balance adequate surface area for heat dissipation with practical economic considerations. The base plate serves as the primary structure for mounting the fins and facilitates uniform heat distribution. Grooves machined into the base plate securely fit the fins, enhancing the contact surface and maximizing heat transfer efficiency. This setup allows for experimentation with varying voltage supplies (20V to 80V) to assess their impact on heat transfer coefficients, following insights from previous research.

4.2.Fins Combination

To maximize heat transfer efficiency, we included four different fin combinations based on literature insights. Each fin's dimensions—80 mm for length, 40 mm for width, and 2 mm for thickness—meet the requirements commonly utilised in heat transfer research.

- **Standard Rectangular Plain Fins**: These fins serve as the benchmark, given their extensive investigation in previous research, providing a control group for performance comparison.
- **Triangular Notch Fins**: Incorporating triangular notches into the rectangular fins was based on previous studies indicating their potential for excellent heat transfer coefficients.
- **Semicircular Notch Fins**: Semicircular notches were introduced to assess the effects of different shapes on heat dissipation, as literature suggests their effectiveness in improving heat transfer.
- **Rectangular Notch Fins**: Rectangular notches were also included to examine their impact on heat transfer performance, aiming to determine how various notch shapes could enhance efficiency.

  By using these fin combinations, we aimed to identify the design offering the best heat transfer efficiency under consistent experimental conditions.

4.3.Orientation

The fin array is oriented vertically to maximize heat transfer efficiency, informed by literature insights. Previous research highlights that vertical configurations significantly enhance natural convection and improve heat dissipation. Studies indicate that vertical orientations promote chimney flow patterns, which effectively enhance convective heat transfer processes. By adopting a vertical fin array configuration, we aim to optimize heat transfer performance and overall system efficiency under consistent experimental conditions.

4.4 Insulation

As insulation for the experimental setup, asbestos sheets were chosen due to their excellent thermal resistance properties and cost-effectiveness. While safety precautions are taken to mitigate health risks associated with asbestos, its thermal performance and availability make it a practical choice for this setup.

5. Calculation and Result

5.1.Heat Transfer Coefficient Calculation and Result

To calculate the coefficient of heat transfer :

$$h_{exp} = \frac{Q\_convection}{A_e \times (T_s - T_\infty)} \quad (1)$$

The heat transfer coefficient may be calculated using the area that is exposed of the finned array, the temperature difference among the surface of the fins and the air around it, and the total convective heat transfer.

Explanation of Terms

1. **Total Convective Heat Transfer ($Q_{convection}$) :**
   To calculate the total convective heat transfer, the total heat supplied to the system is deducted from the total conductive heat transfer:

$$Q_{convection} = Q_{total} - Q_{conduction} \quad (2)$$

Here, Q_{total} is the total heat supplied and $Q_{conduction}$ is the total conductive heat transfer.

2. **Total Heat Supplied (Q_{total}):**
   The total quantity of heat delivered to the system is calculated upon the electrical power input to the heating element:

$$Q_{total} = V \times I \quad (3)$$

Here, V is the voltage supplied and I is the current.

3. **Total Conductive Heat Transfer ($Q_{conduction}$):**
   The entire amount in the heat transmission across the different sides of the insulated base plate is referred to as the conductive heat transfer.

$$Q_{conduction} = Q_{rh} + Q_{lh} + Q_{fs} + Q_{bs} + Q_{bottom} \quad (4)$$

Each term represents the heat transfer through a specific side of the base plate:

- **Right-hand side:**

$$Q_{rh} = \frac{(T_{base} - T_{rh})\, x\, k\, x\, l\, x\, h}{b} \quad (5)$$

- **Left-hand side:**

$$Q_{lh} = \frac{(T_{base} - T_{lh})\, x\, k\, x\, l\, x\, h}{b} \quad (6)$$

- **Front side:**

$$Q_{fs} = \frac{(T_{base} - T_{fs})\, x\, k\, x\, l\, x\, h}{b} \quad (7)$$

- **Back side:**

$$Q_{bs} = \frac{(T_{base} - T_{bs})\, x\, k\, x\, l\, x\, h}{b} \quad (8)$$

- **Bottom side:**

$$Q_{bottom} = \frac{(T_{base} - T_{bottom})\, x\, k\, x\, l\, x\, h}{b} \quad (9)$$

Here, T_{base} is the temperature at the base plate, T_{rh}, T_{lh}, T_{fs}, T_{bs} and T_{bottom} are the temperatures at the respective sides, K is the asbestos sheet's thermal conductivity, l and h are the base plate's dimensions, and b is the insulation thickness.

4. **Exposed Area of the Finned Array (A_e):**
   The total surface area of the exposed base plate plus all of the fins makes up the exposed area of the finned array:

$$A_e = n\,(A_1 + B_a) \tag{10}$$

Here, n is the how many fins, A_1 is area of convective heat transfer for one fin and B_a is base plate's total exposed area.

   - **Area of Convective Heat Transfer for One Fin (A_1):**
     $A_1 = 3.2 \times 10^{-3}\, m^2$
     This is the region of one fin's surface that is open to the air and helps in heat transfer.
   - **Total Exposed Area of the Base Plate (B_a):**
     $B_a = S_f \times L \times N$
     Here, S_f is the the space within two fins, L fin channel's length and N a number of fin channels.

5. **Temperature Difference ($T_s - T_\infty$):**
   The difference in temperature across the air surrounding the fins and their surface:

   - T_s is the surface temperature of the fins.
   - T_∞ is the ambient temperature.

6. **Grashof Number (G_r):**
   The buoyancy-driven flow in a fluid is quantified by the dimensionless Grashof number. It is provided by:

$$G_r = \frac{g\,\beta\,l^3\,\Delta T}{\vartheta^2} \tag{11}$$

Here, β is the thermal expansion coefficient, g Gravitational acceleration, l is the length of the fin channel , ΔT is the difference in temperature between the surrounding air and the surface and ϑ is the kinematic viscosity of the fluid.

7. **Coefficient of Thermal Expansion (β):**
   To compute the thermal expansion coefficient :

$$\beta = \frac{1}{(T_{mf} + 273)} \tag{12}$$

   Here, T_{mf} is the mean film temperature.

8. **Temperature Difference (ΔT):**
   The difference in temperature between the surrounding air and the surface is :

$$\Delta T = T_s - T_\infty \tag{13}$$

   Here, T_s temperature of the surface and T_∞ is the external temperature.

9. **Nusselt Number (N_u):**
   The dimensionless Nusselt number is a representation of the proportion of convective to conductive heat transmission. It is computed as :

$$N_u = \frac{h_{exp} \times L_c}{k} \tag{15}$$

   Here, L_c characteristic length and k fluid's thermal conductivity.

By using this detailed methodology, we can accurately calculate the experimental heat transfer coefficient, Nusselt Number (N_u), Grashof Number (G_r) and gain insights into the efficiency of different fin configurations under consistent experimental conditions.

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Result Table of Experimental Heat Transfer Coefficient:

Table 1 . Result of Non-Inverted Fins

Voltage	Plain Fin			Semicircular Notch Fin			Rectangular Notch Fin			Triangular Notch Fin		
	h_{exp}	N_u	G_r	h_{exp}	N_u	G_r	h_{exp}	N_u	G_r	h_{exp}	N_u	G_r
20	0.98	1.47	46370.5797	1.35	2.03	48650.828	0.74	1.5	48571.77	1.79	2.6	42437.82
35	3.18	4.77	69867.8989	3.26	4.82	87588.1972	3.34	5.10	74856.94	3.19	4.7	86612.199
50	5.00	7.39	124562.9259	4.82	7.12	140690.95	5.83	8.62	141822.9	5.60	8.2	135220.79
65	5.34	7.78	150817.6106	5.12	7.47	203589.727	6.16	8.98	197431.4	7.76	11.3	176959.43
80	5.59	7.94	254332.8221	8.11	11.5	252657.587	9.56	13.7	241002.6	10.1	15.1	219688.23

Table 2 . Result of Inverted Fins

Voltage	Inverted Semicircular Notch Fin			Inverted Rectangular Notch Fin			Inverted Triangular Notch Fin		
	h_{exp} (W/m²K)	N_u	G_r	h_{exp}	N_u	G_r	h_{exp}	N_u	G_r
20	0.41	0.622	42473.82	0.13	0.196	47421.26	0.66	0.99	43713.56
35	3.80	5.7	67438.83	6.32	9.476	81731.69	3.43	5.13	82898.12
50	8.65	12.779	123891.81	7.04	10.404	125039.65	6.59	9.75	126166.05
65	7.45	10.874	177987.27	8.62	12.568	180054.27	7.40	10.79	182974.14
80	9.49	13.486	259723.80	9.72	13.993	242389.80	10.74	15.45	213997.44

6. Result and Discussion

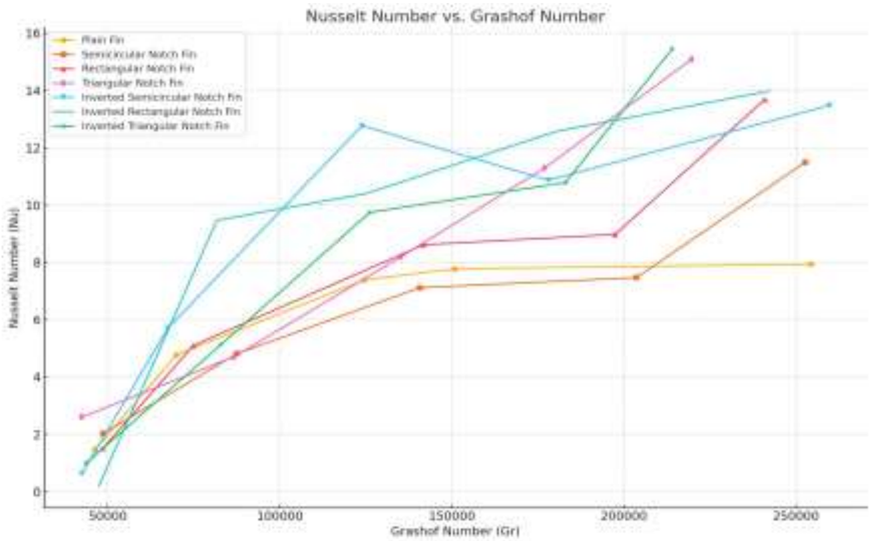

Figure 2 . Nusselt Number Vs Grashof number for fin types

The relationship between the Grashof number (G_r) to the Nusselt value (N_u) across various fin types is depicted in the graph. The Nusselt number, indicating the efficiency of heat transfer, generally increases proportion to higher Grashof numbers, which represent the buoyancy-driven flow. Each line represents a different fin type, with different markers to distinguish them. The triangular and inverted triangular notch fins show the highest Nusselt numbers at higher Grashof numbers, indicating superior heat transfer efficiency in these configurations.

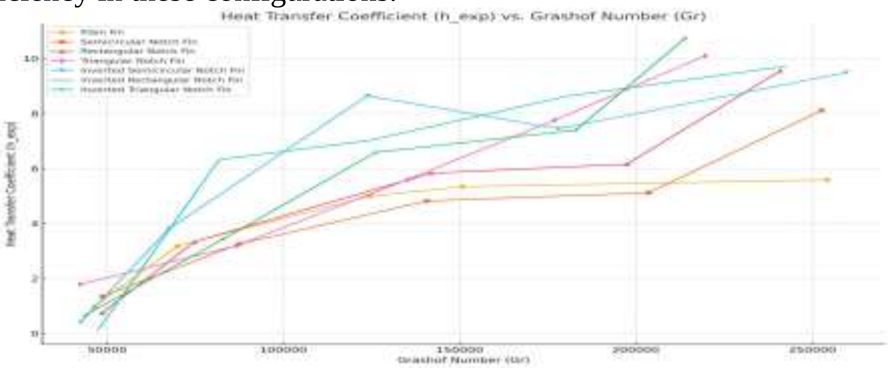

Figure 3. Heat Transfer Coefficient (h_{th}) Vs Grashof Number (G_r)

The graph shows the link for different fin types among the Grashof number (G_r) and the heat transfer coefficient (h_{th}). Different fin types are represented by each line, which illustrates how the heat transfer coefficient varies as the Grashof number increases. In general, the heat transfer coefficient tends to increase with higher Grashof numbers for all fin types. Notably, triangular notch fins and inverted triangular notch fins exhibit the highest heat transfer coefficients at higher Grashof numbers, indicating their superior heat transfer performance compared to other fin types.

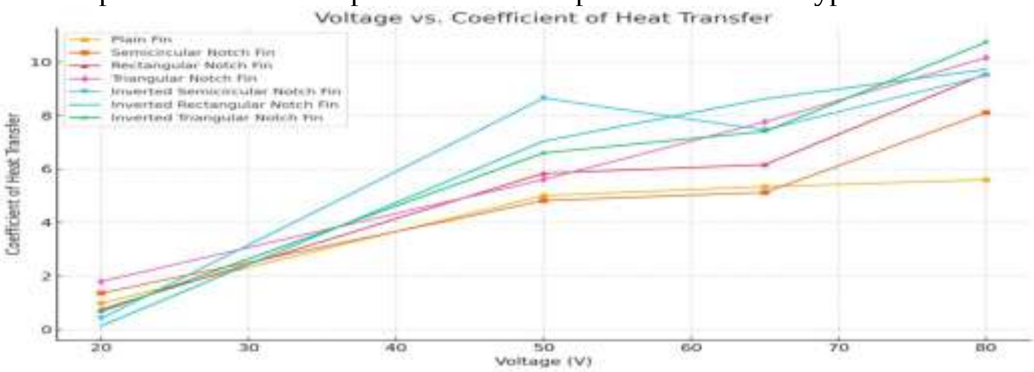

Figure 4. Heat Transfer Coefficient (h_{exp}) Vs Voltage (V)

The link between voltage and the coefficient of heat transfer for different fin types is shown in the graph. As voltage increases, the coefficient of heat transfer generally rises for all fin types, indicating improved heat transfer efficiency at higher voltages. The "Triangular Notch Fin" and "Inverted Triangular Notch Fin" show the highest coefficients at 80V, while the "Inverted Rectangular Notch Fin" and "Inverted Semicircular Notch Fin" exhibit significant increases at mid-range voltages. The "Plain Fin" demonstrates a more moderate and steady increase compared to the notched fins. Overall, notched fins, particularly triangular ones, tend to have better heat transfer performance at higher voltages.

7. Conclusion

- Out of all the configurations, triangle notch fins had the greatest heat transfer coefficient (10.74 W/m^2K) at 80 V, indicating the best heat enhancement.
- Heat transfer coefficients increased with higher voltages for all fin types. Notched fins, particularly triangular and inverted triangular ones, showed the highest efficiencies at 80V..

- Higher Grashof numbers correlated with increased heat transfer coefficients across all fin types, with triangular and inverted triangular notch fins performing the best at elevated Grashof numbers.
- Despite equal surface areas, the geometric design and orientation significantly influenced heat transfer rates. Inverted triangular and rectangular configurations provided superior performance due to better air channeling.
- The Nusselt number, representing heat transfer efficiency, generally rose with higher Grashof numbers. Triangular and inverted triangular notch fins showed the highest Nusselt numbers, confirming their superior heat transfer efficiency.

8. Scope for Future Work

- Future studies could explore using copper or other high-conductivity materials for the fins.
- Varying the notch height from the lowest to highest across the base plate could provide further insights.
- Experiments with forced convection could offer higher input parameters and better results.
- Coating fins with nanomaterials may enhance heat transfer and cooling rates, increasing efficiency.

9. References

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