

Research Article

Evaluation of Conical Pin Fin Arrangements on Heat Sink Heat Transfer and Flow Characteristics

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Abstract:

This study experimentally evaluates the thermal and hydraulic performance of conical pin-fin heat sinks under varying airflow rates, power inputs, and fin spacing configurations. Four heat sink designs were fabricated from aluminum alloy 6061-T6 and tested in an open-loop wind tunnel. Key performance metrics, including the Nusselt number and apparent friction factor, were assessed across Reynolds numbers ranging from 10,480 to 23,900 and power inputs of 180, 285 and 480 W. Results indicate that the configuration with a longitudinal-to-transverse pitch ratio (R) of 1 achieved the highest thermal performance, enhancing the Nusselt number by up to 19.4% compared to other arrangements. In contrast, the $R = 2$ configuration exhibited the highest friction factor due to increased flow resistance. Experimental outcomes were validated using established correlations from Zukauskas, Colburn, and Jacob, showing strong agreement. These findings support the design optimization of heat sinks for efficient thermal management in electronic applications.

Keywords: Conical fins, heat sink, Nusselt number, pressure drop, thermal performance, forced convection

1. Introduction

A heat sink is a critical component in thermal systems, primarily used to manage excess heat generated by electronic devices and machinery. By increasing surface area and using materials with high thermal conductivity, heat sinks efficiently transfer heat away from heat-sensitive components to the surrounding environment. They are commonly made from metals like aluminum or copper. Heat sinks help maintain optimal operating temperatures, preventing thermal damage and ensuring the longevity and performance of devices. Various designs, such as finned or pin-type heat sinks, are employed depending on the specific thermal management needs of the system. Effective heat dissipation is essential for system reliability [1]. Many studies have been conducted to improve the thermal performance of a heat sink, which involves optimizing several design parameters that affect its ability to dissipate heat efficiently. The material's thermal conductivity is crucial. Metals such as copper and aluminum are frequently used due to their high thermal conductivity, which facilitates better heat transfer from the source to the heat sink [2], [3], [4]. A larger surface area enhances heat dissipation.

Fins, pins, or other extended surfaces increase the heat sink's area, thereby improving heat transfer to the surrounding fluid [5], [6], [7]. Fins' shape, spacing, and thickness significantly affect heat transfer.

Optimizing fin density and geometry improves airflow and heat dissipation without excessive bulk [8][9][10], [11]. Airflow over the heat sink significantly affects thermal performance. Proper orientation and maximizing convection

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enhance heat dissipation. Forced convection (using fans or pumps) dramatically improves heat transfer [12], [13]. Fin efficiency measures how well a fin transfers heat from the heat sink to the surrounding air, influenced by fin thickness, height, and material [14][15], [16]. Nilpueng et al. (2021) [17] investigated the effects of pin fin shape, orientation, and the S/Dp or S/Wp ratio on thermal performance, heat transfer coefficient, and pressure drop in plate pin fin heat sinks. The study examines variables such as pin fin shape (circular, square, and 45° square), size (2.5, 3.0, and 3.5 mm), air velocity, orientation, and S/Dp or S/Wp ratio. Findings indicate that square pin fin configurations (PSPFHS) enhance heat transfer by 12.52% compared to circular pin fins. Reducing the S/Dp ratio significantly improves heat transfer performance and pressure drop. Thermal performance factors exceed 1, showing that PPFHS outperform traditional plate-fin heat sinks. Xie et al. (2021) [18] analyzed the hydraulic and thermal performance of microchannel heat sinks with inserted pin fins at various inclined angles (0°, 30°, 45° and 60°) and arrangement methods (in-line and staggered). It employs numerical simulation to evaluate the flow and temperature fields, friction factors, and Nusselt numbers over a Reynolds number range from 10 to 300. The findings suggest optimal thermal performance occurs with in-line pin fins at a 30° incline, resulting in a higher friction factor. In contrast, changes in pin fin arrangement have a minimal effect on performance compared to the inclined angle. Abdelmohimen et al. (2021) [19] studied multi-slide configurations in plate-fin heat sinks, analyzing one to four staggered slides. They compared thermal and hydraulic performance under impinging and parallel flow. Numerical simulations were conducted for Reynolds numbers from 1333 to 5334. Results show that parallel flow has lower thermal resistance than impinging flow, but more slides in parallel significantly increase pumping power. The study finds that three slides affect flow balance, reduce thermal resistance, and cause minimal changes in pumping power. Abbas et al. (2025) [20] examined the thermal and hydrothermal behavior of conical and truncated conical fins in a staggered heat sink under laminar forced convection. Five fin geometries were tested with a tip-to-base diameter ratio ($\beta = 0$ to 0.8) while maintaining constant surface area, height, and pitch. Experimental and theoretical analyses showed that increasing β decreased heat transfer, convection coefficients, and steam condensation due to reduced fin-base contact. Results closely matched mathematical predictions. Design of Experiments (DOE) optimization identified optimal conditions, and empirical correlations were developed for heat transfer, pressure drop, and condensation rates based on steam pressure and coolant flow. Wang and Chen (2019) [21] analyzed the thermal and hydraulic performance of micro pin fin heat sinks (MPFHSs) with circular, square, diamond, and streamline pin fin shapes fabricated in copper through laser micro milling. Experiments with deionized water at two heat flux levels and various Reynolds numbers show that streamline fins produce the best heat transfer, while diamond fins yield the lowest pressure drop. Although diamond fins have weaker heat transfer, they offer the best overall thermal-hydraulic performance due to low thermal resistance at similar pressure drops. These findings guide the optimization of MPFHS designs for high-heat-flux electronic cooling applications. Previous studies have examined various heat sink parameters for pin and laminated fins; however, conical fins have received comparatively limited attention. Abbas et al. (2024) [22] conducted an experimental study to evaluate the thermohydraulic performance of heat sinks by replacing conical pin fins with truncated cone fins. Using a Design of Experiments (DOE) approach, 65 tests were carried out under varying steam pressures (500–11,500 Pa) and cold-water flow rates (0.5–2.25 LPM). Five fin configurations were analyzed, one conical and four truncated designs with tip-to-base diameter ratios (β) from 0.2 to 0.8, while keeping surface area and height constant. Performance metrics included heat transfer rate, Nusselt number, pressure drop, thermal resistance, and Thermal Performance Index (TPI). Results showed that the conical fin heat sink consistently achieved superior heat dissipation and lower thermal resistance. As β increased, heat transfer declined due to reduced base contact and increased boundary layer thickness. The conical design reached TPI values up to 4.6, while truncated designs fell below 1 in some cases, confirming that conical fins offer better overall thermal performance.

Previous studies have examined various heat sink parameters for pin and laminated fins; however, conical fins have received comparatively limited attention. To address this gap, the present study investigates the effect of the ratio of longitudinal pitch to transverse pitch on the performance of a conical pin fin heat sink under forced convection and varying heat flux conditions

2. Experimental investigation

An open-loop experimental test channel is fabricated to conduct experiments on the pin fin heat sink. The rectangular test channel is open to the atmosphere on one side, while the other end is connected to the suction side of the blower. Air is directed to flow through the test channel at different flow rates to gather data on heat transfer and pressure drop.

2.1 Experimental setup

The experimental setup comprises a rectangular cross-section duct with dimensions of 15 cm × 12 cm and a length of 50 cm. A heat sink is installed at the center of the duct, where the duct is constricted to dimensions of 18 cm × 12 cm × 8 cm to ensure that the airflow is fully directed over the heat sink surface, as shown in Fig. 1. A centrifugal blower was installed at the inlet (left) side of the channel to induce airflow, while a hot wire anemometer type (HT_9892) was positioned at the front of it to measure the volumetric flow rate through the channel. Three electric heaters, each with a capacity of 140 watts, were installed in the heat sink and connected in series to a 1000 VA variac. Also, the test rig was equipped with measuring instruments: a data acquisition unit (Applent AT4516) with 16 channels, each connected by a thermocouple type T, was utilized to record temperatures at 16 positions in the test rig. A manometer of type H290 was connected to the test rig in order to measure the pressure differential across it. Figure 2 presents the assembled components of the test rig and the accuracy and resolution of the measuring devices are illustrated in Table 1.

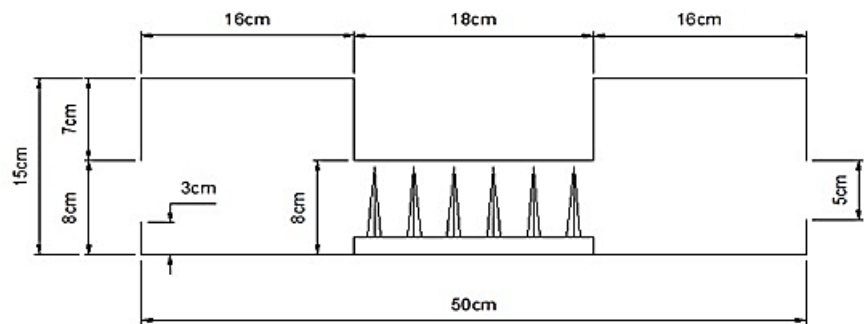


Fig. 1. Sketch diagram of the test rig

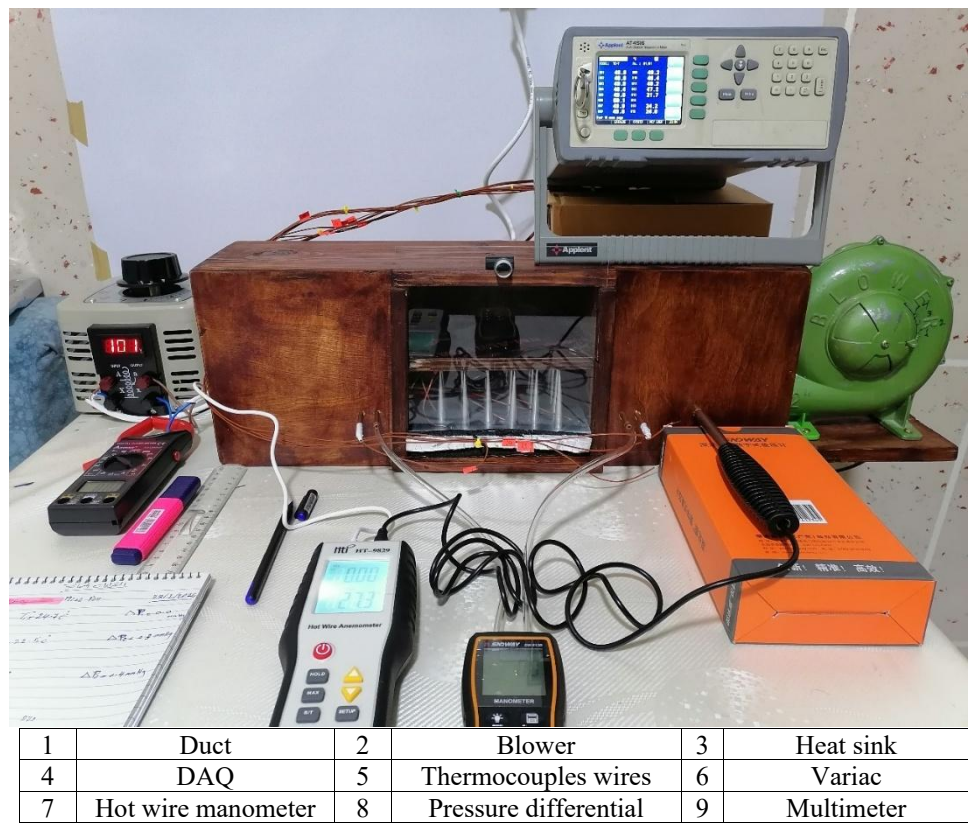


Fig. 2. Photo of the test rig

Table 1: The accuracy and resolution of measurement instruments

Instruments name	Accuracy	Resolution
Data acquisition DAQ	($\pm 0.2\% + 1$) °C	0.1 °C
Hot wire manometer	($\pm 5\% + 0.1$) m/s	0.1 m/s
Pressure differential	± 2 Pa	0.1 Pa
Multimeter		
Voltage	$\pm (0.8 \% + 5)$ V	0.1 V
Current	$\pm (2.0 \% + 5)$ A	0.01 A

2.2 Heat sink fabrication

Four conical pin-fin heat sinks were designed and fabricated using aluminum alloy 6061-T6. Each heat sink features a base plate measuring 18 cm in length, 12 cm in width, and 1.5 cm in thickness. Mounted on each base plate are 21 conical fins, each with a base diameter of 1 cm and a height of 6 cm. Additionally, three electric heaters, each rated at 140 W, are embedded within the base plate, as shown in Fig. 3. The fins are arranged on the base plates according to four different X_l/X_t ratios. Figure 4 and Table 2 present the complete dimensions of each heat sink.



Fig. 3. Photo of heat sink sample

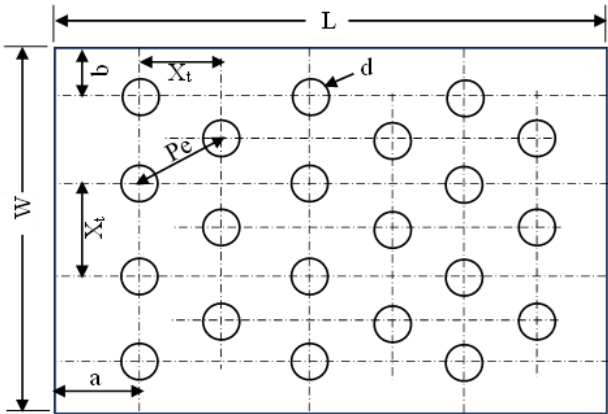


Fig. 4. Sketch of the general heat Sink design

Table 2: Details of the investigated heat sink dimensions

R	θ (Degree)	L (cm)	W (cm)	H (cm)	a (cm)	b (cm)	X_t (cm)	X_l (cm)	Pe (cm)
1	26.6°	18	12	8	1.5	1.5	3	3	10.5
1.25	32°	18	12	8	3	1.5	3	2.4	7.26
1.5	36.87°	18	12	8	4	1.5	3	2	5.5
2	45°	18	12	8	5.25	1.5	3	1.5	3.75

3. Data processing

The analytical component encompasses the determination of heat sink geometry, heat transfer, and flow characteristics:

3.1 Determining the characteristics of heat sink geometry

The geometrical characteristics of the heat sink include the following parameters:

- Minimum free flow area is calculated using the following expression [23]:

$$A_o = \left\{ \left[\left(X_t - \frac{d_f}{2} \right) \times \left(\frac{W}{X_t} - 1 \right) + 1 \right] \right\} \times l \quad (1)$$

- Total fin area, which contributes to the overall heat dissipation, is determined by

$$A_f = N_f \frac{\pi d_f}{2} \sqrt{l^2 + \left(\frac{d_f}{2} \right)^2} \quad (2)$$

- Unfinned surface area is also considered in the total heat dissipation analysis, and calculated as:

$$A_p = L * W - N_f \frac{\pi d^2}{4} \quad (3)$$

Accordingly, the total heat dissipation area is determined by summing the finned surface area (A_f) and the non-finned surface area (A_{No_fin}), as expressed below:

$$A_T = N_f \frac{\pi d}{2} \sqrt{l^2 + \left(\frac{d}{2} \right)^2} + L * W - N_f \frac{\pi d^2}{4} \quad (4)$$

- Frontal area is defined as a projected frontal area of the heat sink determines the extent of exposure to the incoming air flow; it is expressed as;

$$A_{fr} = W \times H \quad (5)$$

- Hydraulic diameter is calculated as:

$$d_H = \frac{4A_o L}{A_T} \quad (6)$$

3.2 Determination of heat transfer and flow characteristics

- Heat transfer characteristics

The actual heat dissipation from the heat sink to the flowing air is calculated using the following expression [24], [25]:

$$Q = \dot{m} c_p (T_o - T_i) \quad (7)$$

where

$$\dot{m} = \rho u A_f \quad (8)$$

alternatively, the convective heat transfer rate is defined by the convective heat transfer coefficient, as given by

$$h = \frac{Q}{A_t(\bar{T}_w - T_b)} \quad (9)$$

where

$$T_b = \frac{T_i + T_o}{2} \quad (10)$$

and

$$\bar{T}_w = \frac{\sum_{i=1}^{i=n} \bar{T}_{w,i}}{n} \quad (11)$$

The heat transfer coefficient can be expressed in terms of the Nusselt number as follows:

$$Nu = \frac{h d_H}{k} \quad (12)$$

- Air flow characteristics

The Reynolds number, which indicates the flow regime (laminar or turbulent), is given by

$$Re = \frac{\rho u_m d_H}{\mu} \quad (13)$$

The apparent friction factor quantifies the pressure loss from friction within the heat sink channels and is determined as [26]:

$$f_{app,ave} = 2 \frac{d_H}{L} \frac{\Delta P}{\rho u_m^2} \quad (14)$$

4. Results and discussion

4.1 The Nusselt number

Figures 5 to 7 illustrate the variation of the Nu number with Re number at input power levels of 180 W, 285 W, and 480 W, respectively. In all cases, a positive correlation between the Nu number and Re number is evident. This trend is consistent with the physical expectation that increasing the airflow rate enhances convective heat transfer due to increased turbulence and improved mixing near the heat sink surfaces. Moreover, a noticeable increase in the Nu number is observed with rising power input. Higher power levels elevate the thermal gradient between the heat source and the surrounding air, thereby intensifying the rate of heat transfer. This enhancement is further supported by the role of the R factor; defined as the ratio of X_l pitch to X_i pitch in the fin array, which influences airflow uniformity and the effective surface area exposed to the flow. As the R factor increases, it tends to improve heat dissipation by optimizing flow distribution and reducing flow resistance across the fins. These results collectively highlight the interdependence of geometric design and operating conditions in determining the thermal performance of heat sinks. The combined effects of Re number, power input, and R factor must be carefully considered to maximize heat transfer efficiency in practical applications. It was found that the Nusselt number values in all experiments were initially clustered within a narrow range of approximately 80 to 100. However, with increasing Reynolds number, a significant divergence in these values was observed. The figures indicate that the Nusselt number for a R factor of 1 exhibits the highest values compared to other R factors, ranging from 100 to 150, 96 to 158 and 93 to 154 at power inputs of 128 W, 258 W and 480 W, respectively, respectively. In contrast, the lowest Nusselt number values are observed for a R factor of 1.25, ranging from 87 to 135, 84 to 134 and 82 to 133 at input power levels of 128 W, 258 W and 480 W, respectively. To further analyze the previous results, the average Nusselt number was calculated at each power input across three airflow rates, as illustrated in Figure 8. The results show that the R factor of 1 yields the highest average

Nusselt numbers across all power inputs, ranging from 130.47 to 133.32. In contrast, the R factor of 1.5 exhibits the lowest average values, ranging from 105 to 114.6.

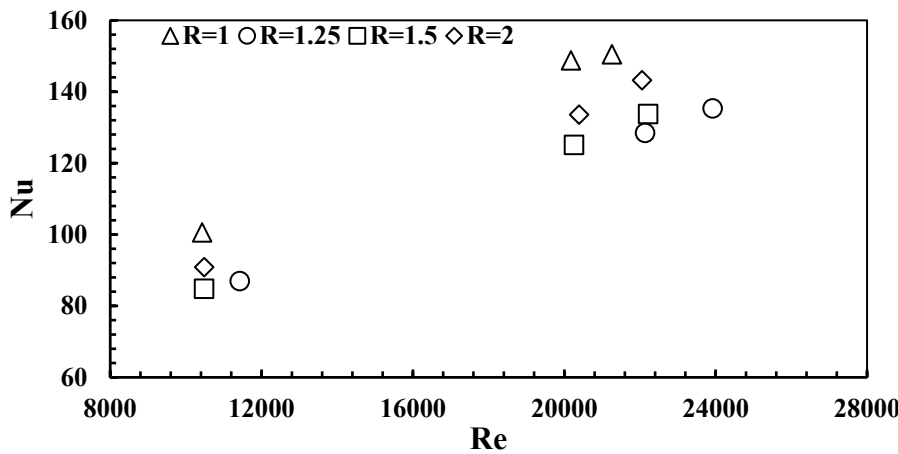


Fig. 5. Variation of Nu number with Re number at a power input of 185 W

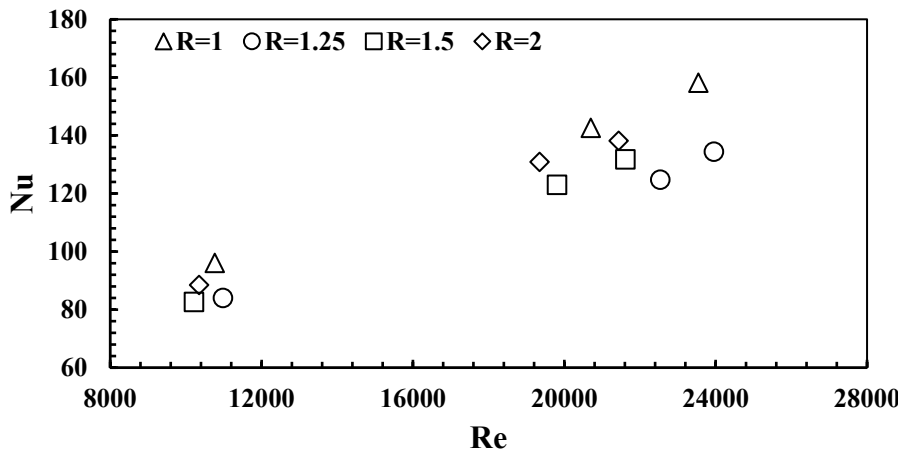


Fig. 6. Variation of Nu number with Re number at a power input of 285 W

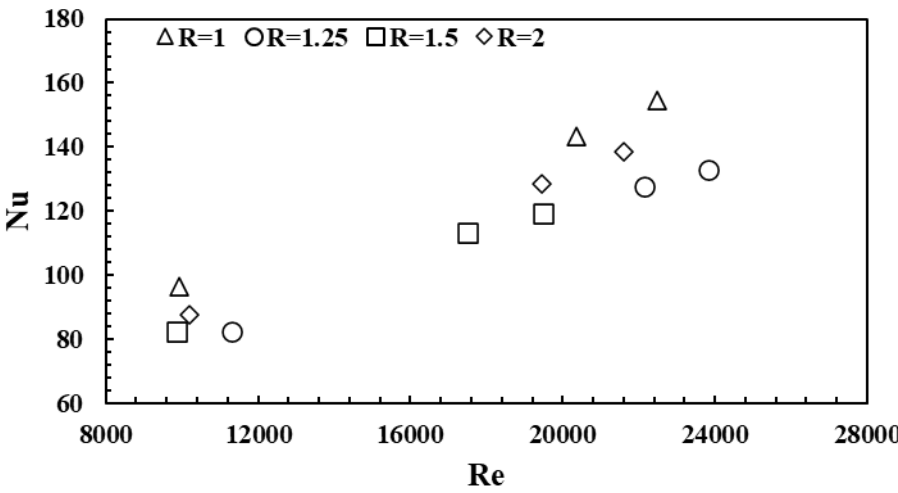


Fig. 7. Variation of Nu number with Re number at a power input of 480 W

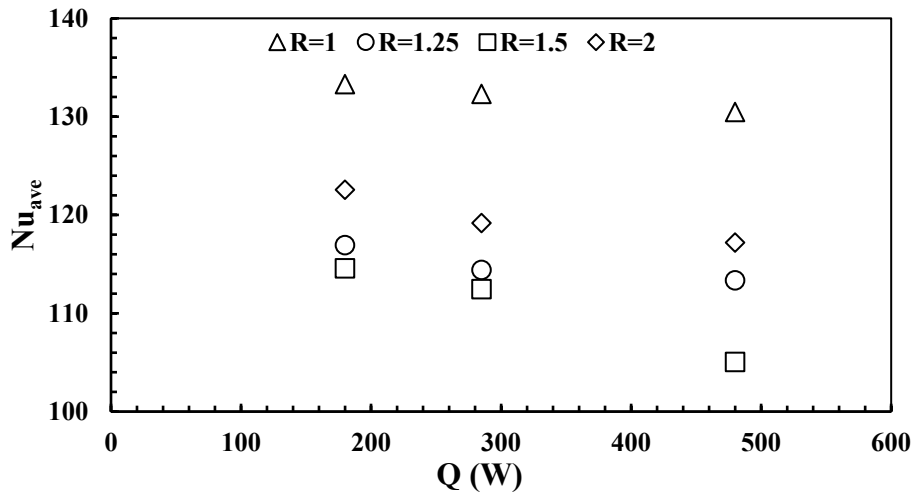


Fig. 8. Variation of Average Nusselt Number with Power Input

4.2 Evaluation of the average apparent friction factor

The average apparent friction factor ($f_{app,ave}$) was determined from the experimental pressure drop measurements conducted at three input power levels and corresponding airflow rates. The results, presented in Figures 9 and 11 for input powers of 180 W, 285 W and 480 W, respectively, indicate that the R factor, together with the Re number and input power, significantly influences the average apparent friction factor, $f_{app,ave}$. Among the tested configurations, the heat sink with a Rfactor of 2 exhibits the highest friction factor due to the narrowing of the flow passages, which increases flow resistance. At an input power of 180 W, the $f_{app,ave}$ ranges from 0.0054 to 0.0046 as Re increases from 10,480 to 22,053. For input powers of 285 W and 480 W, it varies from 0.0056 to 0.0050 and from 0.0056 to 0.00511, respectively. The lowest $f_{app,ave}$ was observed at a Rfactor of 1 in the experiments with 285 W and 480 W input power. In contrast, at higher Re values under 180 W input, the minimum $f_{app,ave}$ was recorded for both R factors of 1.5 and 2. The average apparent friction factor ($f_{app,ave}$) for a R value of 2 increased by 1.7%, 5% and 5.7% compared to a R value of 1 at input powers of 180 W, 285 W, and 480 W, respectively. For a Rvalue of 1.25, the increases were 1.6%, 2.4% and 3.7%, while for R = 1.5, the corresponding increases were 3.5%, 9.8% and 12.7%.

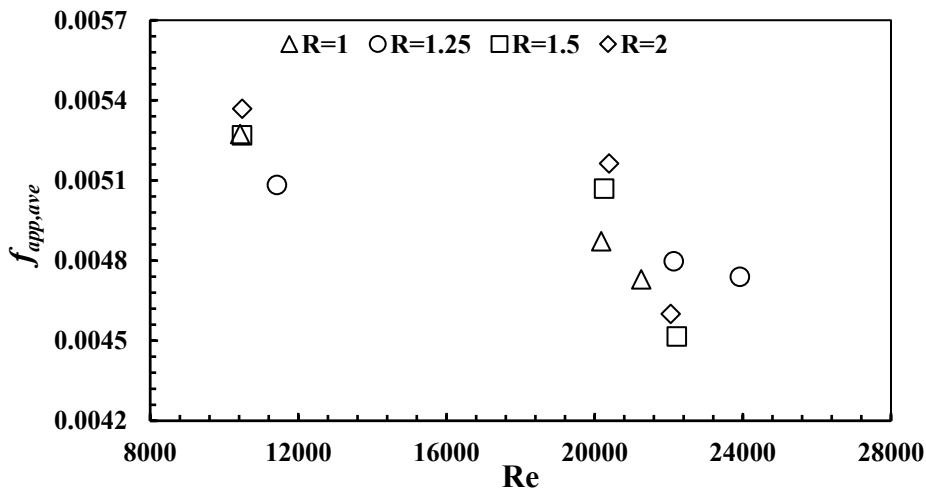


Fig. 9. Variation of $f_{app,ave}$ with the Re number for different R factors at an input power of 180 W

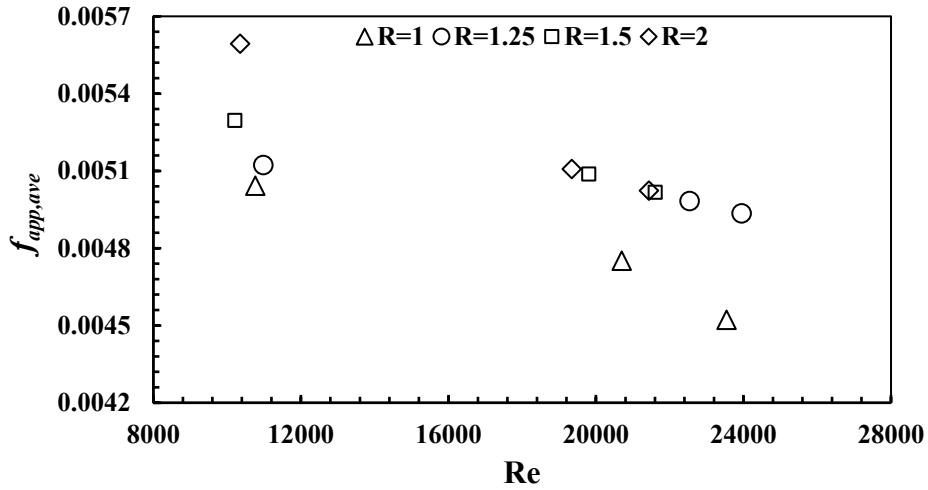


Fig. 10. Variation of $f_{app,ave}$ with the Re number for different R factors at an input power of 285 W

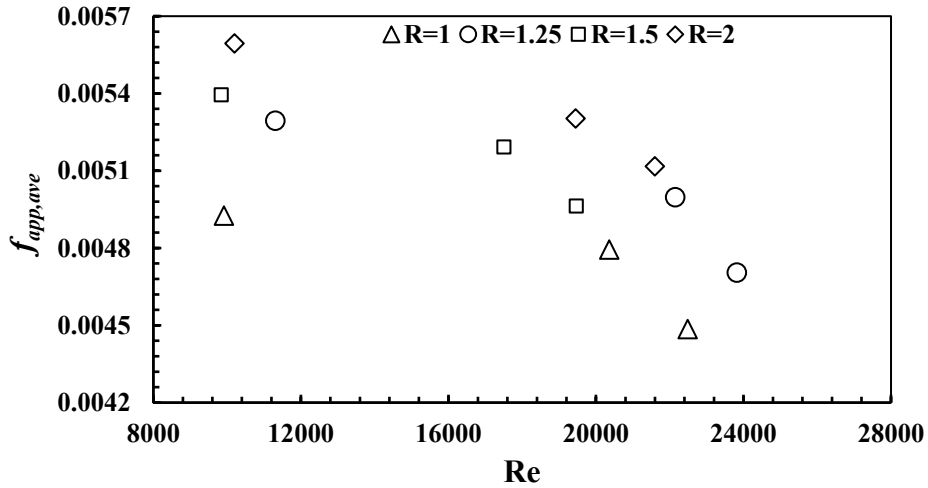


Fig. 11. Variation of $f_{app,ave}$ with the Re number for different R factors at an input power of 480 W

5. Results validation

In these experiments, the average apparent friction factor has been determined from the experimental pressure drop, as shown in Figs. 9 to 11. These figures indicate that higher levels of the friction factor were observed in experiments conducted at the R factor of 2. The experimental results were validated by correlations suggesting a staggered bundle tube arrangement for the Nu number and the friction factor. The validation was conducted on the Nu number and friction factor for experiments at 480 W using the following correlations:

- Correlation, which Zukauskas presented for calculating the Nu number from a staggered bundle of tubes [27]

$$Nu = 0.35 \left(\frac{x_t}{x_l} \right)^{0.2} Re_{max}^{0.6} Pr^{0.36} \left(\frac{Pr}{Pr_w} \right)^{1/4} \quad (15)$$

- Correlation, which Colburn presented for calculating the Nu number from a staggered bundle of tubes [28]

$$\overline{Nu} = 0.33 Re_{max}^{0.6} Pr^{0.36} \quad (16)$$

- Correlation, which Jacob presented for calculating the f factor from a staggered bundle of tubes [29]

$$f = \left[0.025 + \frac{0.1175}{\left[\left(\frac{X_t - d}{d} \right)^{1.08} \right]} \right] R_{max}^{-0.16} \quad (17)$$

5.1 Validation of the Nusselt Number

The experimental Nusselt number was validated using two correlation models proposed by Zukauskas and Colburn. The results are presented in Figures 12 to 14 for R factors ranging from 1 to 2, respectively. The experimental Nusselt number showed better agreement with Zukauskas' correlation than with Colburn's across all tests. This is attributed to the broader applicability of Zukauskas' relation, which incorporates key parameters such as the transverse-to-longitudinal pitch ratio (X_t/X_l) and the density ratio (ρ/ρ_w). These parameters significantly influence both the Nusselt number and the friction factor. It was found that the minimum and maximum errors between the experimental Nusselt number and that predicted by Zukauskas' correlation did not exceed 0.8% and 3.5%, respectively. In contrast, when compared with the Colburn correlation, the error ranged from 7% to 23%.

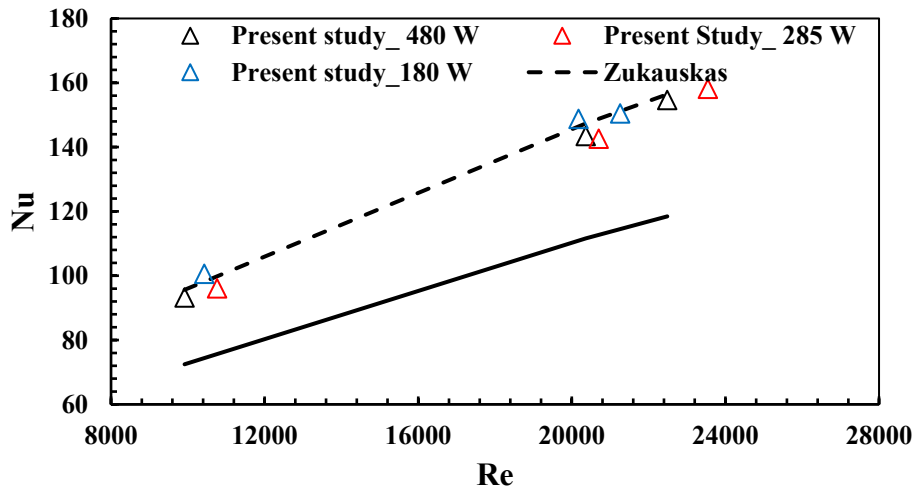


Fig. 12. Validation of experimental Nu number against correlations at R = 1 for varying input powers

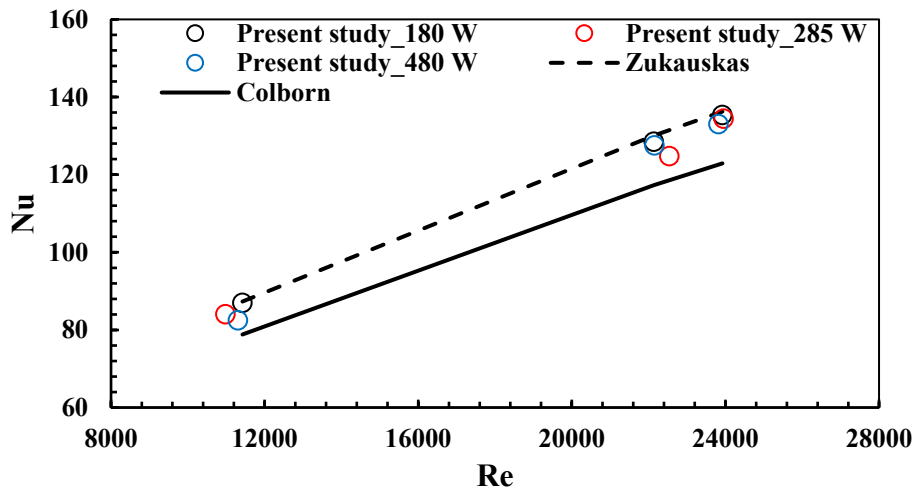


Fig. 13. Validation of experimental Nu number against correlations at R = 1.25 for varying input powers

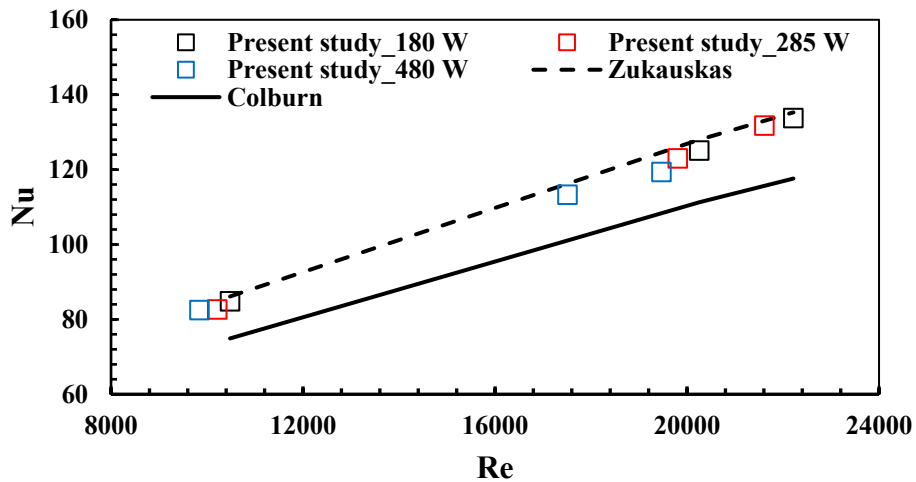


Fig. 14. Validation of experimental Nu number against correlations at $R = 1.5$ for varying input powers

5.2 Validation of apparent friction factor

The experimental apparent friction factor in all tests was validated using the Jacob correlation, which is commonly employed for friction factor estimation. The results are presented in Figures 15 to 18. The experiment conducted at $R = 1$ showed a significant deviation between the experimental results and the Jacob correlation, as shown in Figure 15. Similar significant deviations were observed for R factors of 1.25 and 1.5, except in the experiment at 285 W, where the results at high Reynolds numbers (22,150 to 23,950) closely aligned with the Jacob correlation, as shown in Figs. 16 and 17. The best agreement with the Jacob correlation was observed in the experiments at $R = 2$, where most of the experimental friction factors closely matched the correlation. The minimum and maximum errors ranged from 0.11% to 5.2%.

6. Quantification of experimental uncertainty

Errors in experimental data recording typically arise from three main sources: calibration, data collection, and data reduction. Each of these categories encompasses multiple sources of uncertainty, which may originate from the measuring equipment or instrumentation systems. These sources are often documented in the equipment specifications, which include parameters such as linearity, accuracy, drift, response time, and repeatability [30]. The Engineering Equation Solver (EES) software was employed to evaluate the relative uncertainties of the independent and dependent parameters. The corresponding results are summarized in Table 3.

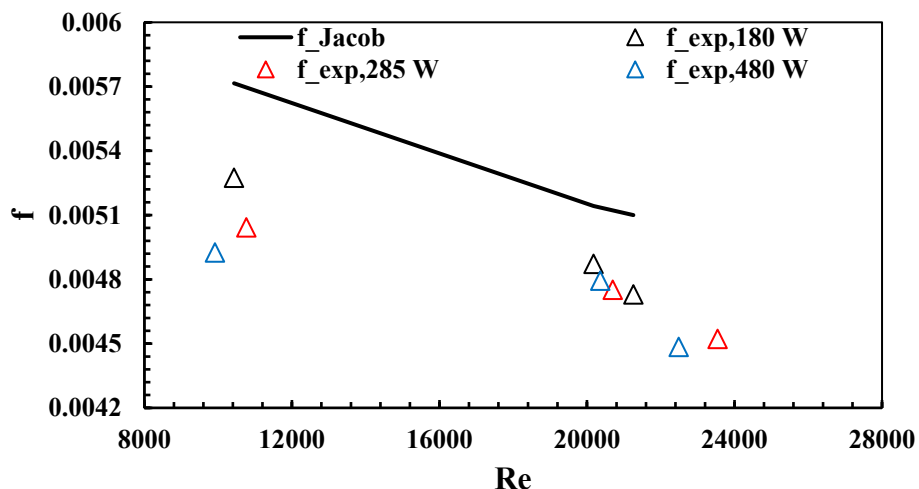


Fig. 15. Validation of experimental friction factor against correlations at $R = 1$ for varying input powers

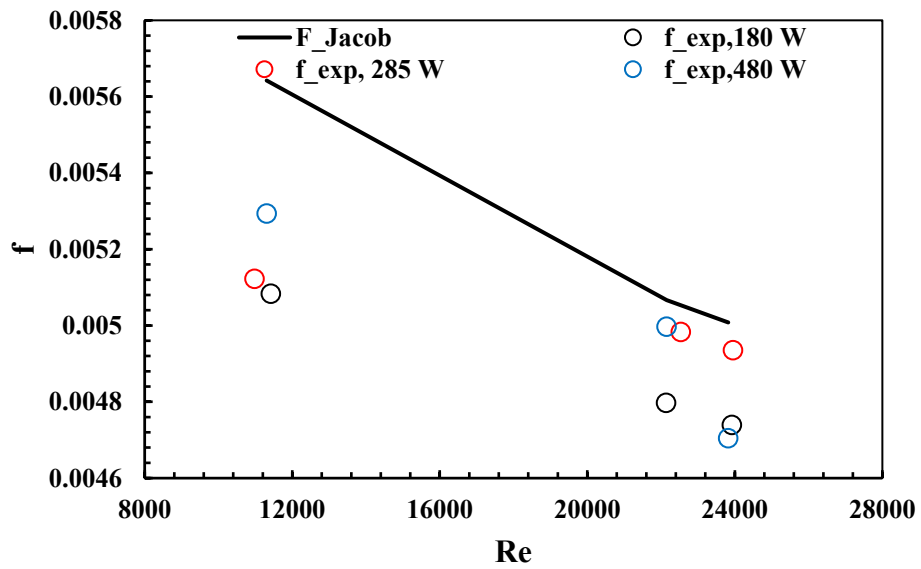


Fig. 16. Validation of experimental friction factor against correlations at $R = 1.25$ for varying input powers

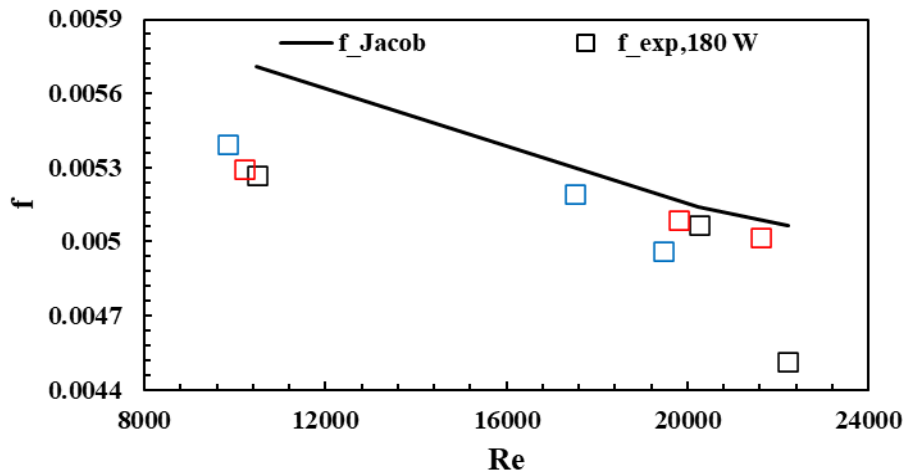


Fig. 17. Validation of experimental friction factor against correlations at $R = 1.5$ for varying input powers

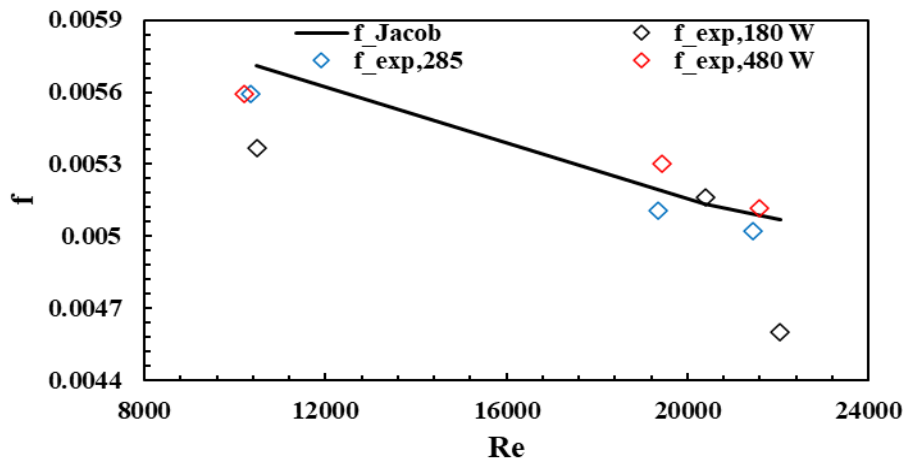


Fig. 18. Validation of experimental friction factor against correlations at $R = 2$ for varying input powers

Table 3: Summary of Relative Uncertainties for Independent and Dependent Variables

Parameter	Relative uncertainty %	Parameter	Relative uncertainty %	Parameter	Relative uncertainty %
1. Independent parameters					
A_c	± 0.89	A_f	± 0.316	A_p	± 1.1
A_t	0.57 m^2	T_i	± 1.75	T_o	$\pm 1.47 \text{ }^\circ\text{C}$
T_w	$\pm 0.66 \text{ }^\circ\text{C}$	u	± 0.14	ΔP	± 0.3
2. Dependent parameters					
cp	± 0.075	f	± 0.67	h	± 3.49
k	± 0.069	$\dot{m}$	± 0.905	Nu	± 3.47
P	± 0.158	Pr	± 0.45	Re	± 0.3
Q	± 13.84	ρ	± 0.215	μ	± 0.635

7. Conclusion

The heat transfer and flow characteristics of conical pin-fin heat sinks were experimentally analyzed under varying airflow rates, input powers, and fin spacing ratios. The following conclusions can be drawn:

- The configuration with a R factor of 1 consistently demonstrated the highest thermal performance, enhancing the Nusselt number by 15.6%, 19.4%, and 10.4% compared to R = 1.25, 1.5, and 2, respectively.
- The R = 1.25 configuration showed the lowest thermal performance across all tested conditions.
- Increasing the input power led to a higher apparent friction factor, indicating increased flow resistance with rising thermal loads.
- The highest friction factor was observed for the R = 2 configuration, with increases of 4.13%, 2.56%, and 8.67% compared to R = 1, 1.25, and 1.5, respectively.

Overall, the results suggest that a R factor of 1 offers the most favorable balance between heat transfer enhancement and flow resistance, making it a suitable design choice for applications requiring efficient thermal management under forced convection.

Nomenclature

A	Area, m^2
a	Distance from the fin center to each end of L, m
b	Distance from the fin center to each end of W, m
cp	Specific heat, $\text{J/kg}\cdot^\circ\text{C}$
d	Diameter, m
f	Friction factor
h	Coefficient of convection heat transfer, $\text{W/m}^2\cdot^\circ\text{C}$
H	Duct height, m
l	Fin length, m
L	Heat sink length, m
k	Thermal conductivity, $\text{W/m}\cdot^\circ\text{C}$
ΔP	Pressure drop, Pa
$\dot{m}$	Mass flow rate, kg/s
P	Input power, W
Pe	Fin pitch, m
T	Temperature, $^\circ\text{C}$
Q	Heat transfer rate, Q
u	Velocity, m/s
W	Heat sink width, m
X	Distance, m

Greek symbols

ρ	Density, kg/m ³
μ	Dynamic viscosity, Pa.s

Dimensionless Groups

Nu	Nusselt number
Pr	Prandtl number
Re	Reynolds number

Subscript symbols

app	Apparent
ave	average
b	bulk
f	fin
fr	frontal
H	Hydraulic
i	inlet
m	Mean
max	Maximum
l	Longitudinal
t	Transvers
T	Total
o	Outlet
O	Minimum free flow
w	Wall

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