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RESEARCH ARTICLE

Experimental Analysis of Thermal and Hydraulic Performance of Fiber-Reinforced Polymer Fins in Compact Heat Exchangers

M.H. Alshameri[1], A.A Azizuddin[1]*, A. Arshad[1], H. G. Mohammed[2], S. Sadeq[3]

[1]Faculty of Mechanical and Automotive Engineering Technology, Universiti Malaysia Pahang Al-Sultan Abdullah, 26600 Pahang, Malaysia
[2]Department of Mechanical Engineering, Universiti Teknologi PETRONAS, 32610 Bandar Seri Iskandar, Perak, Malaysia
[3]School of Engineering, Design and Built Environment, Western Sydney University, Kingswood, Penrith, NSW 2747 Australia

ABSTRACT – At present, the significance of fiber-reinforced polymer (FRP) material in fins-and-tube heat exchangers (HEs) has become critical in the thermal industrial sector. This is due to its advantages, such as lightweight nature, ease of manufacturing, and superior resistance to environmental conditions compared to metal materials. The utilization of FRP fins-and-tube heat exchangers has generated considerable interest among researchers, given their pivotal role in various thermal engineering systems. However, the use of carbon nanotube-reinforced polymers (CNTRP) fins in fin-and-tube heat exchangers has not been thoroughly examined, highlighting the need for a comprehensive study. This research aimed to design and fabricate a compact heat exchanger by integrating (CNTRP) fins with copper tubes to create a fully functional heat exchanger unit. The study also assessed the performance of the CNTRP fins by analyzing various geometric and process parameters and investigating their thermal and hydraulic characteristics. Experiments were conducted to evaluate the influence of parameters, such as the number of fins (no fin, six, eight, twelve fins), tube diameter (6.5, 8, 9.5 mm), inlet air flow velocity (2.4, 2.9, 3.2, 3.4, and 3.6 m/s), and inlet temperature (40, 60, and 80 °C) on the thermal-hydraulic performance of the heat exchanger. The major findings showed that (CNTRP) fins have a positive impact on heat transfer enhancement, the greater the number of fins, from no fins to six, eight, and twelve fins, the heat transfer coefficient improved by 6%, 10.77%, and 17%, respectively. Meanwhile, the pressure drop rose by 5.8%, 7.4%, and 9% with the same increase in the number of fins. However, this improvement in heat transfer coefficient is accompanied by a rise in pressure drop. Consequently, it is crucial to identify the trade-off in heat exchanger design whereby the increase in pressure drop penalty is weighed against the improvement in heat transfer efficiency.

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1. INTRODUCTION

In the world of industrial and engineering applications, designing heat exchangers with improved thermal properties and better fluid dynamics is crucial for boosting energy efficiency and reducing costs. How well a heat exchanger performs has a big impact on the overall heat transfer process, making its optimization vital for effective energy conservation. As global energy use continues to grow, we face environmental issues like ozone depletion, which call for smarter and more efficient energy use in all sectors [1][2]. This increasing demand highlights the urgent need to manage energy resources effectively, especially given the pressing concerns about conserving resources and maintaining environmental sustainability. Energy conversion and transportation are vital in many areas, including power generation, manufacturing, automotive, and HVAC systems. As manufacturers aim to cut production costs while boosting energy efficiency, a lot of focus is placed on optimizing heat exchanger designs [3]. Recent studies have concentrated on improving exergy utilization and the performance of air-side heat transfer. These efforts emphasize the delicate balance between achieving efficiency and maintaining cost-effectiveness [3]. Several factors affect the efficiency of heat exchangers, including the materials used, the design and size of the tubes, the spacing between tubes and fins, and the flow dynamics [4][5].

Compact fin and tube heat exchangers come with several benefits. They are highly efficient because of their large surface area relative to their size, which makes them perfect for tight spaces. By using less material, they also help lower initial costs. These heat exchangers are versatile enough to meet different process needs and perform well even in high-pressure situations. Additionally, they boost energy efficiency by increasing fluid velocity and turbulence. Because of these advantages, they are a popular choice in industries like refrigeration, aerospace, and chemical processing, where thermal performance and space limitations are crucial [6], [7].

In the last decade, there has been a growing interest in reinforced polymer heat exchangers, thanks to the impressive benefits that polymers provide. These materials have outstanding qualities, such as resistance to corrosion, a lightweight

***CORRESPONDING AUTHOR | A.A. Azizuddin** | ✉ azizuddin@umpsa.edu.my

design, cost-effectiveness, and ease of manufacturing. As a result, they are becoming more appealing alternatives to traditional metal heat exchangers [8], [9]. Polymeric materials are well-known for their many advantages, but they do face a challenge when it comes to dissipating heat due to their low thermal conductivity. This often results in the buildup of thermal stress. Many studies have looked into ways to improve the thermal conductivity of reinforced polymers, showing that adding specific materials can lead to significant enhancements such as [10], [11] and [12]. Moreover, researchers have explored various high-conductivity fillers, with carbon nanotubes (CNTs), boron nitride (BN), aluminum oxide, diamond, and graphene standing out as popular options in numerous studies [13]. These materials are selected for their outstanding thermal properties, which can greatly enhance the overall performance of polymer composites when integrated into them. Adding these fillers not only boosts thermal conductivity but can also improve other characteristics, like mechanical strength and durability. Using carbon nanotube (CNT)-reinforced polymers to create fins for heat exchangers is an exciting area of research that could significantly enhance thermal performance. However, recent studies have pointed out some important gaps that need to be addressed to fully leverage the benefits of these advanced materials. One notable study by [14] examined the thermal and electrical properties of multi-walled carbon nanotube (MWCNT) reinforced polymer composites with different aspect ratios. The researchers used a three-roll milling technique to ensure a uniform distribution of CNTs in the polymer matrix. Their findings showed that composites with a higher aspect ratio had much better thermal and electrical performance. The study concluded that optimizing the aspect ratio of CNTs is crucial for maximizing thermal efficiency.

Researchers at [15] explored how adding carbon nanotube (CNT) fins could improve the cooling performance of shape memory alloy wires. They coated the wires with CNT fins and tested how well they transferred heat. The results showed a clear boost in cooling efficiency, thanks to the excellent thermal conductivity of CNTs compared to more traditional materials. This points to the strong potential of CNTs for use in areas where good heat management is critical, like heat exchangers. However, despite the promising properties of CNTs themselves, a review of CNT/polymer nanocomposites found that their actual thermal conductivity often does not live up to what is expected based on CNTs' intrinsic qualities. This gap in performance is largely due to interfacial thermal resistance, which makes it harder for heat to flow efficiently. The researchers pointed out that more work is needed to improve the connection between the CNTs and the surrounding polymer matrix. By optimizing this interface, we could significantly boost the overall thermal conductivity of these materials [16]. Integrating CNT-reinforced polymer fins into current heat exchanger designs still comes with some challenges. A recent study on carbon fiber-reinforced, corrugated polymer plate heat exchangers showed that careful design of composite materials can lead to better thermal conductivity. This highlights the importance of thoughtful material engineering when aiming to improve heat exchanger performance [17]. However, [18] noted that more research is needed to successfully integrate CNTs into conventional heat exchanger systems and fully unlock their performance benefits.

There are two main ways to tackle challenges in thermal and fluid dynamics: theoretical and experimental methods. Theoretical approaches use mathematical models and equations to predict how fluids and thermal systems will behave, while experimental methods involve hands-on testing to observe and measure real-world performance. When it comes to heat exchangers, their efficiency is heavily influenced by both geometric and process-related factors. For example, tube diameter and the choice of fin material play crucial roles. Among these, tube diameter is especially important in finned-tube heat exchangers. Studies have shown that reducing the diameter of the pipes can significantly enhance heat transfer. This happens because smaller tubes increase fluid velocity and turbulence, which boosts the convective heat transfer coefficient and improves overall thermal performance [19], [20]. Moreover, the performance of finned-tube heat exchangers also depends heavily on the type of fin material used, including polymer-based options. Material selection plays a key role, as it directly impacts thermal efficiency, durability, and resistance to corrosion. Different materials offer different advantages: aluminum, for example, is widely used for its high thermal conductivity, while copper provides excellent heat transfer and corrosion resistance.

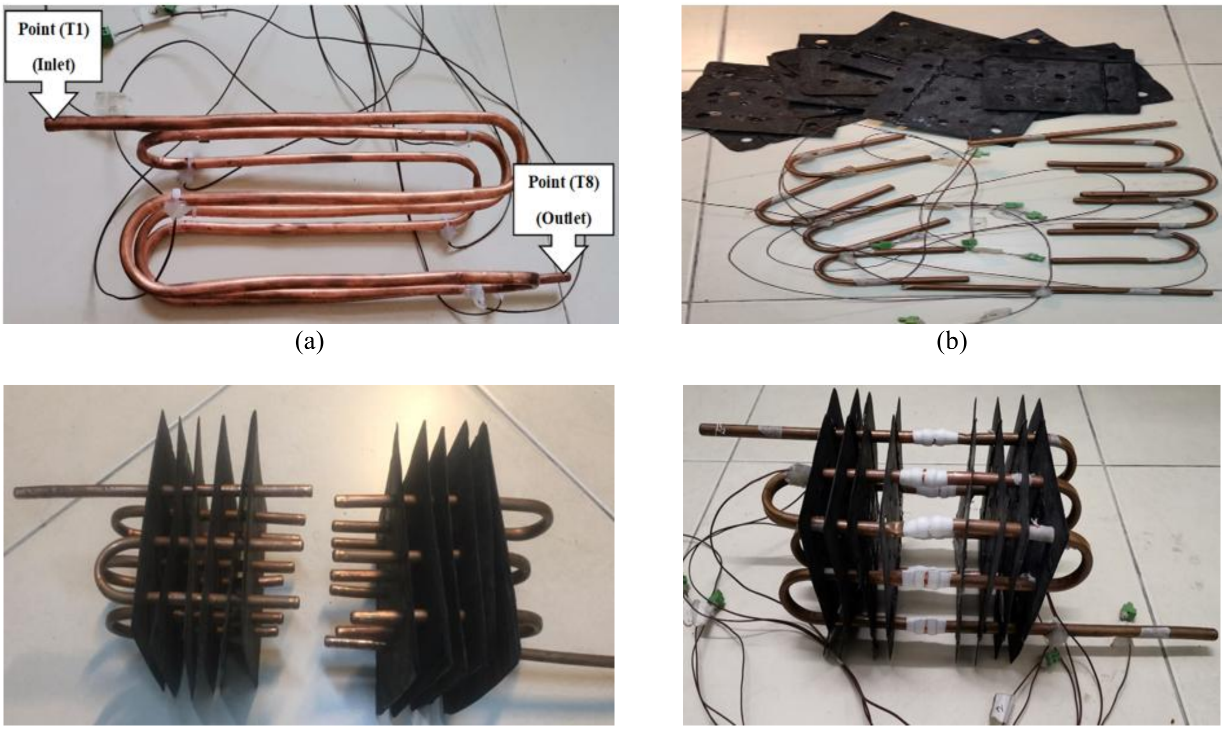
On the other hand, polymers, though typically lower in thermal conductivity, offer benefits like reduced weight and improved resistance to chemical corrosion, making them suitable for specific applications [21], [22]. As well, air velocity inside the wind tunnel and the temperature of the fluid entering the heat exchanger are key process parameters that greatly affect its performance. Generally, higher air velocities improve the convective heat transfer coefficient, which boosts the overall rate of heat exchange. This happens because faster airflow creates more turbulence, allowing heat to transfer more efficiently between surfaces and fluids. The convective heat transfer coefficient, in particular, is a critical factor; it determines how effectively heat moves from the heat exchanger's surface to the flowing air [23]. However, boosting air velocity isn't without drawbacks. It also causes a higher pressure drop across the heat exchanger, which means more energy is needed to maintain airflow. This added energy demand can offset the benefits gained from improved heat transfer. In other words, while higher air speeds can enhance thermal performance, they may also increase operating costs due to the extra power required to overcome the resulting pressure losses [24] [25]. The temperature of the fluid entering a heat exchanger plays a key role in determining its performance and efficiency. When the hot fluid enters at a higher temperature, it creates a larger temperature gradient between the fluid and the heat exchanger surface. This stronger thermal driving force boosts the rate of heat transfer. Additionally, higher fluid temperatures can trigger a shift from laminar to turbulent flow, which promotes better fluid mixing and further enhances heat transfer efficiency [26].

What makes this study unique is the development of a heat exchanger that uses fins made from polymers reinforced with 2% carbon nanotubes. This innovative approach aims to enhance thermal conductivity and address the limitations of traditional polymer heat exchangers. The study uniquely explores the impact of two key geometric parameters, tube diameter (6.5 mm, 8 mm, and 9.5 mm) and the number of fins (no fin, six, eight, and twelve fins) on heat transfer efficiency. Furthermore, it investigates the effects of two critical process parameters: air velocity within the wind tunnel (ranging from 2.6 m/s, 2.9 m/s, 3.2 m/s, 3.4 m/s, and 3.6 m/s) and the inlet temperature of the fluid (40 °C, 60 °C, and 80 °C). This comprehensive evaluation aims to assess the thermal and hydraulic performance of the heat exchanger, including the associated pressure drop penalties. Furthermore, the results of this research will assist heat exchanger designers in developing more efficient polymeric heat exchangers, thereby improving heat transfer and fluid flow characteristics.

2. METHODOLOGY

2.1 Preparation of the Polymer Heat Exchangers Unit

The heat exchanger was fabricated using a copper tube with a uniform thickness, and the total length of the tube was 2.5 m. The tube was then bent into a zigzag configuration using a copper pipe bender tool. This design allowed the fluid to enter at point T1, pass through the whole pipe section, and exit at point T8, as illustrated in Figure 1(a). The primary parameter investigated in this study was the use of fins made from a carbon nanotube-reinforced polymer. To fabricate the sample, a slow hardener was used with an epoxy resin-to-hardener ratio of 100:28.4 by weight. The epoxy resin and hardener weighed 110 g and 31.24 g, respectively. To enhance heat transfer properties, a 2% carbon nanotube (CNT) was added to the mixture. Assuming A represents the combined epoxy resin and hardener, the ratio of A to CNT was 98:2, resulting in a CNT weight of 2.88 g. To enable the insertion of these fins on both sides of the tube and preserve the heat exchanger's design integrity, the tubes were cut in half to facilitate the fin installation process. As shown in Figure 1(b), this depicts the tube after the cutting step. Subsequently, the assembly process took place between the tube and the fins, as illustrated in Figure 1(c). Copper conductors were used to reconnect the two separated sections of the copper tubes, and to preclude any potential liquid leakage during the experiments through the copper conductors, Teflon tape was applied as a preventative measure, as shown in Figure 1(d).



(a) (b) (c) (d)

Figure 1. The fabrication process of the heat exchanger unit

The research employed a strategic sensor placement approach to monitor the temperature within the heat exchanger. Eight J-type thermocouple sensors were positioned at key locations across the heat exchanger. The first sensor, labeled T1, was installed at the inlet of the heat exchanger, while the final sensor, T8, was placed at the outlet. The remaining six sensors, T3 through T7, were evenly spaced along the heat exchanger wall, creating equidistant measurement points between the inlet (T1) and outlet (T8). This sensor arrangement enabled the research to sequentially study the temperature differences from the inlet to the outlet of the heat exchanger. The final heat exchanger design used in this study is shown in Figure 2.

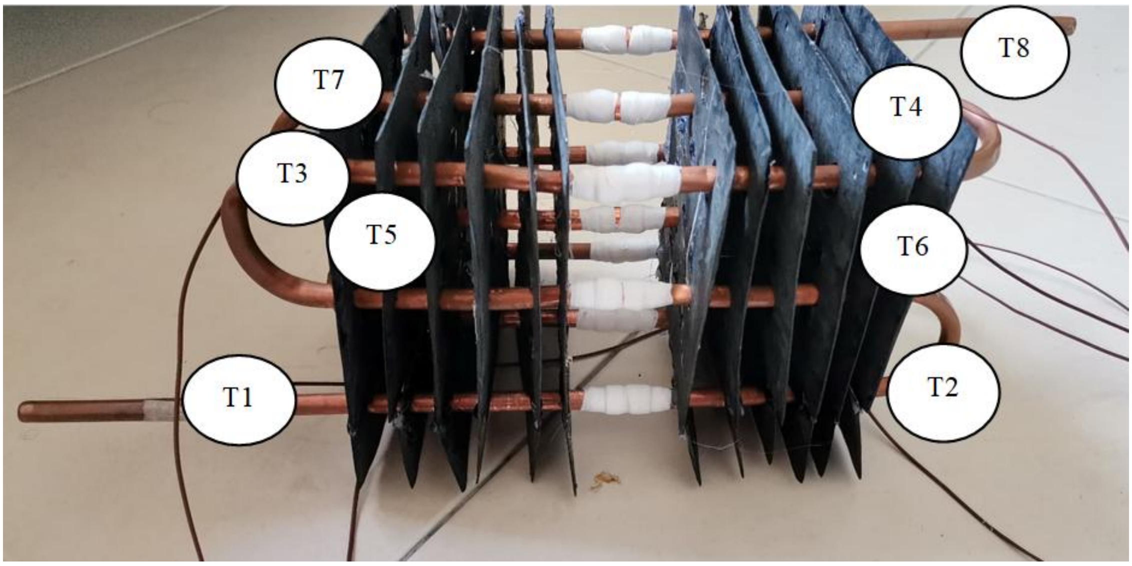


Figure 2. The final shape of the heat exchanger

2.2 Experimental Setup

The study used a carefully designed experimental setup featuring a single-phase airflow system. It operated on the principle of counterflow, allowing detailed investigation of the flow and heat transfer behavior in compact fin-and-tube heat exchangers. The setup included an open-loop rectangular wind tunnel, a flow straightener, and a test section. It was equipped with a variable-speed air blower and a 1500-Watt coil heater to control the temperature, along with a data acquisition system containing various measuring instruments. Together, these components made it possible to thoroughly analyze the heat exchangers' performance. Figure 3 shows the final layout of the experimental setup used in this study. Additionally, the geometric and process parameters of the tested heat exchangers were carefully recorded and summarized in Table 1, offering a detailed overview of the design features of the components.

Table 1. The geometric and process dimension of the test samples

Geometric & process parameters	Symbol	Dimensions
Tube wall thickness	δ	0.61 mm
Transverse tube pitch	P_t	30 mm
Longitudinal tube pitch	P_l	30 mm
Tube length	L	2500 mm
Tube outside diameter	D_{od}	6.5, 8, 9.5 mm
Tube inner diameter	D_{id}	5.89, 7.39, 8.89 mm
Number of fin	N_{row}	0, 6, 8, 12
Fin thickness	t	2 mm
Fin dimensions	$L \times W$	120×120 mm
Air velocity	V	2.4, 2.9, 3.2, 3.4, 3.6 m/s
Inlet temperature	T	40, 60, 80 ℃

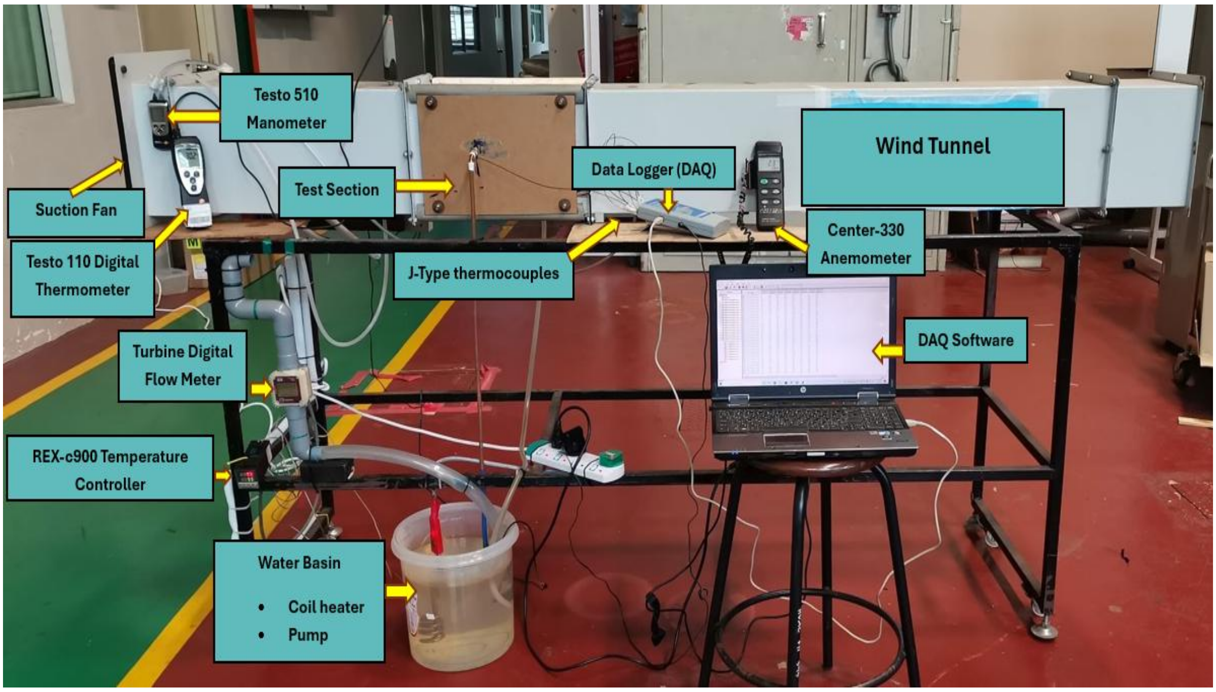


Figure 3. The final configuration of the complete experimental setup with the various measuring instruments

2.3 Experimental Procedure

The study used a wind tunnel setup to test the performance of a finned-tube heat exchanger under different operating conditions. Here's how the experiment was carried out: First, the water in the tank was heated to the target temperature (40°C, 60°C, and 80°C) using a 1500-Watt coil heater controlled by a REX C-100 temperature controller to keep the temperature steady. Then, a 50-Watt AC air suction fan was turned on to draw air into the test section above the heat exchanger. The air velocity was adjusted across several values (2.4 m/s, 2.9 m/s, 3.2 m/s, 3.4 m/s, and 3.6 m/s) depending on the specific test. Water was circulated continuously from the basin to the heat exchanger inlet using a 22-Watt hot water pump. After flowing through the heat exchanger, the water returned to the basin, maintaining a closed loop throughout the experiment. The wind tunnel itself had a square cross-section, with each side measuring 260 mm. A straightening device was positioned at the inlet to maintain uniform air flow velocity, with the test section placed 1.2 meters downstream to ensure the air was properly conditioned before entry. A center-330 anemometer was placed at the back of the wind tunnel to measure the air velocity and temperature before the test section, and a digital thermometer testo 110 was mounted on the upper wall of the wind tunnel to measure the air temperature after it exited the test section. Pitot tubes connected to a testo 510 manometer were inserted at the bottom of the wind tunnel, both before and after the test section, to measure the pressure drop. Eight J-type thermocouples were strategically positioned on the heat exchanger, as illustrated in Figure 2. Six of these thermocouples (T2, T3, T4, T5, T6, and T7) were installed within the test section on the heat exchanger, while two were located at the inlet (T1) and outlet (T8) of the heat exchanger. An OM-DAQPRO-5300 data logger was used to record the temperature readings at these eight locations over a two-minute period. During the experiments, the following parameters were recorded: air velocity, air temperature before and after the test section, and the pressure drop across the heat exchanger. This experimental setup allowed for the investigation of the heat exchanger's performance under various water temperatures and air flow conditions, with comprehensive data collection to analyze the system's thermal and hydraulic characteristics. More than that, the devices used to measure the experimental results were calibrated, and the accuracy percentages are presented in Table 2.

Table 2. Essential apparatuses for data collection and their accuracy percentages

Apparatus	Model	Qty	Capacity Range	Accuracy	Purpose
Data logger	(OM-DAQPRO-5300)	1	T, (-200-1200) ℃	± 0.5%	To collect the data from thermocouple (Type J)
Thermocouple Type J	-	8	T, (0-750) ℃	± 1.1 ℃ or 0.4%	To measure the temperature of outer tube surface
Anemometer	CENTER300	1	V, (0.6-20) $^{m}/_{s}$ T, (-50-500) ℃	Air flow ± 0.2 $^{m}/_{s}$ Temperature ± 0.8 ℃	To measure air velocity and air temperature
Digital temperature sensor	TESTO720	1	T, (-20-50) ℃	± 0.1 ℃	To measure air temperature
Digital manometer	TESTO 510	1	P, (0.41 to 40 in H_2O) hectopascal	± (0.04 in H_2O + 1.5%)	To measure the pressure drop difference
Frequency inverter	REE50	1	Speed (0-100) %	-	To control the fan speed
Temperature controller	REX-C100	1	T, (0-400) ℃	± 0.5%	To control the tube surface temperature
Brushless DC water pump	JT4502-1	1	Max 240 $^{L}/_{h}$	-	To pump the water through the tube
Turbine digital flow meter	K24	1	Volume (10-120) $^{L}/_{min}$	± 1%	To measure water flow rate
Coil heater	-	1	Max. 100 ℃	-	To heat up the tube surface

The main goal of data reduction is to analyze the experimental temperature data recorded during each test run. The purpose of this analysis is to determine the overall heat transfer performance on the air side and the pressure drop across the heat exchanger. This analysis involves calculating the following parameters:

$$U = \frac{Q}{A \times LMTD} \tag{1}$$

where Q represents the heat transfer rate in watts (W); A denotes the heat transfer surface area in square meters (m^2); $(LMTD)$ is the logarithmic mean temperature difference in degrees Celsius (℃).

The air-side heat transfer rate (Q), used to calculate the heat transfer coefficient, was obtained using the following equation:

$$Q = Q_{conv} = \dot{m}C_p(\Delta T) \tag{2}$$

where $\dot{m}$ is the mass flow rate (kg/s); ΔT is the temperature difference between the outlet and the inlet air temperature. The mass flow rate was calculated thus:

$$\dot{m} = \rho VA \tag{3}$$

The total surface area and log mean temperature difference:

$$LMTD = \frac{(T_w - T_{in}) - (T_w - T_{out})}{ln\frac{(T_w - T_{in})}{(T_w - T_{out})}} \tag{4}$$

where T_w is the tube wall temperature; T_{in} is the air inlet temperature, and T_{out} is the air outlet temperature.

The Reynolds number (R_e) was an important parameter to determine the type of flow exhibited by using the following equation:

$$R_e = \frac{\rho v D}{\mu} \tag{5}$$

where ρ is the density of the fluid, v is the velocity of the fluid, D is the hydraulic diameter, and μ is the dynamic viscosity of the fluid.

The friction factor (f) for both the finned and unfinned circular tubes was computed using the following equations:

$$f = \frac{D}{L}\frac{2\Delta P}{\rho V_{max}^2} \tag{6}$$

where Δp is the pressure difference between the inlet and the outlet, and it is expressed as follows:

$$\Delta p = P_{in} - P_{out} \tag{7}$$

The efficiency of a heat exchanger is a measure of how effectively it transfers heat from the hot fluid to the cold fluid, and it is expressed as follows:

$$\varepsilon = \frac{q}{q_{max}} \tag{8}$$

where q is the actual amount of heat transfer; q_{max} is the maximum amount of heat transfer.

Furthermore, the Thermal Performance Index (TPI) is calculated to evaluate the most important characteristics of heat transfer for the same pumping power requirements and is a valuable tool for improving the design and operation of heat exchangers by the following equation:

$$TPI = \frac{\varepsilon}{\Delta P} \tag{9}$$

where ε is the efficiency of heat exchangers.

3. RESULTS AND DISCUSSION

3.1 Thermal Performance

The main objective of this pilot study is to assess how well fiber-reinforced polymer fins perform at dissipating heat in compact heat exchanger applications. The study examines these fins with respect to two geometrical parameters and two process parameters. Specifically, the diameter of the tube heat exchanger was varied between 6.5 mm, 8 mm, and 9.5 mm, while the number of fins ranged from no fins to six, eight, and twelve fins. These geometrical parameters yielded distinct results while controlling for two process parameters: air flow velocity within the wind tunnel, which was set at 2.4 m/s, 2.9 m/s, 3.2 m/s, 3.4 m/s, and 3.6 m/s, and the water inlet temperature of the heat exchanger, which was varied between 40°C, 60°C, and 80°C.

3.1.1 Temperature profile

The experimental findings present temperature measurements taken at eight distinct points (T1 to T8) along the surface of the tube, captured by the data logger. There is a noticeable decrease in temperature from T1 to T8, influenced by various factors detailed later. Three randomized experiments were conducted, varying tube diameter, number of fins, inlet temperature, and air velocity. Figures 4(a), 4(b), and 4(c) illustrate the temperature gradient for the experiments with configurations (6.5 mm, twelve fins, 80 °C, 3.6 m/s), (8 mm, no fins, 60 °C, 3.6 m/s), and (9.5 mm, six fins, 40°C, 3.6 m/s).

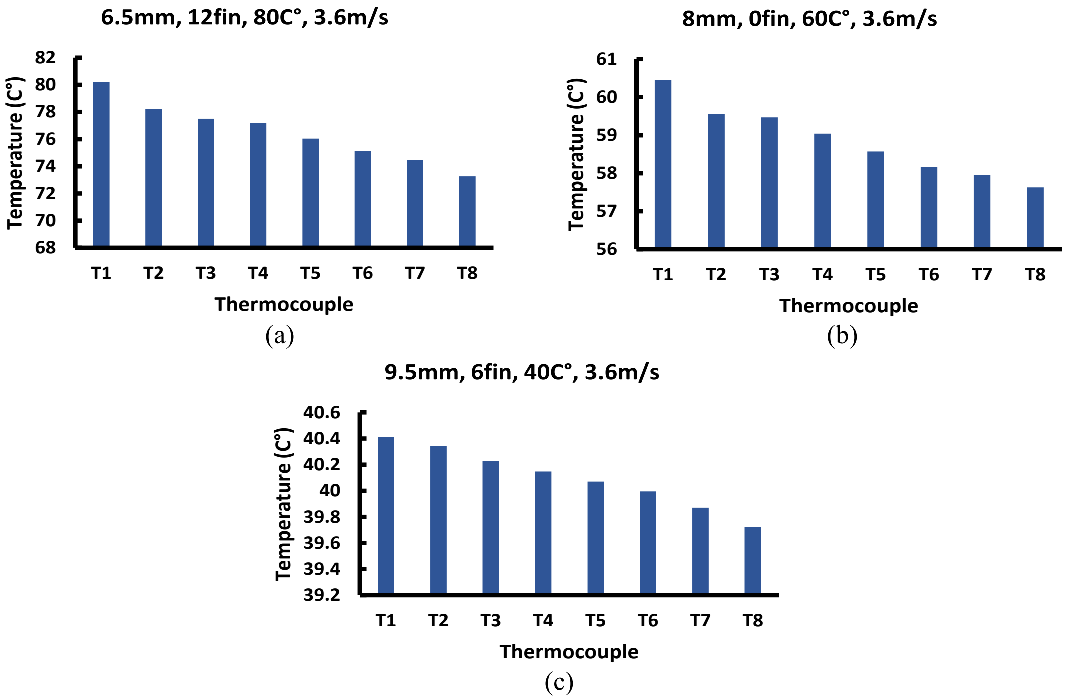
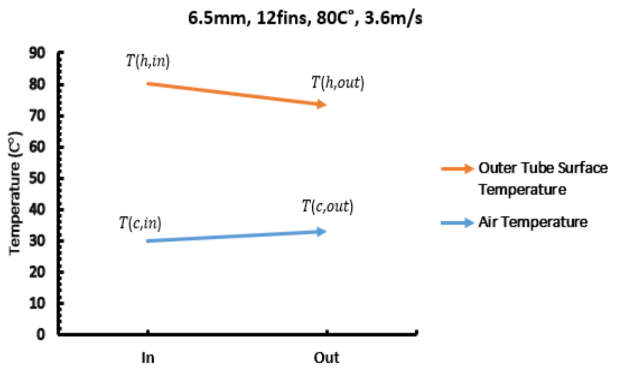


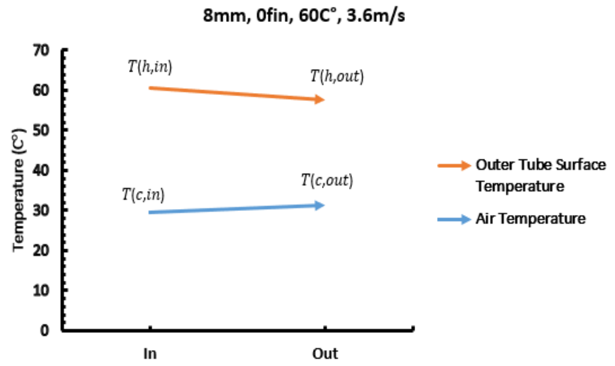
Figure 4. The values of temperature along the tube surface

In every case, the temperature of the tube steadily dropped, which caused the air leaving the tube to be warmer than the air entering it. This shows that as the air moved along, it absorbed heat from the hot surfaces set at (40°C, 60°C, and 80°C). Interestingly, the area behind the tubes, known as the wake region, showed a smaller temperature difference compared to the main flow, indicating less heat transfer in that zone [27]. Additionally, the wind tunnel design ensured unidirectional, parallel airflow across the test section Figure 5, a standard feature that facilitated consistent air movement around the heat exchanger. This uniform flow was essential for obtaining accurate and reliable experimental data [28].

One of the essential things in this study was to achieve a temperature steady state across all points (T1 to T8) on the heat exchanger's surface to accurately measure heat flux. While achieving a perfectly linear temperature-time relationship is challenging, the results showed straight lines with minor fluctuations within acceptable margins. These zigzag patterns were attributed to factors like thermocouple error, wind tunnel vibrations, and ambient noise, which had negligible effects on the overall results. The experiments successfully maintained this stability, with each point (T) displaying distinct temperatures while the overall system remained stable. Three randomized experiments were conducted, varying tube diameter, air speed, number of fins, and temperature, as illustrated in Figure 6. The experiments took 9.5, 22, and 34 minutes to heat the water to 40°C, 60°C, and 80°C, respectively. Once the target temperatures were reached, readings were taken for 200 seconds, revealing a linear temperature trend and a gradual decrease from T1 to T8 along the tube.



(a)



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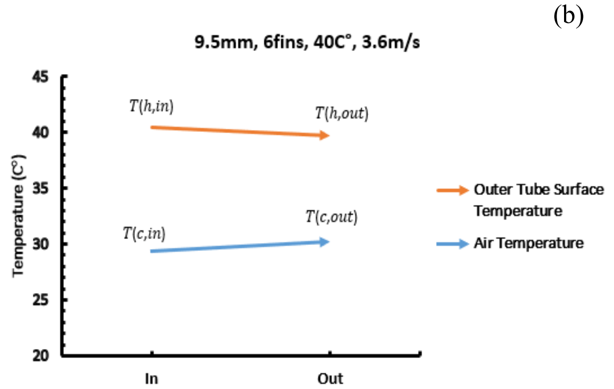
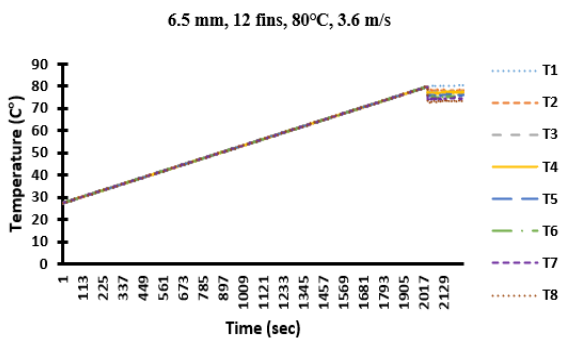


(c)

Figure 5. The temperature profile of the hot and cold fluids



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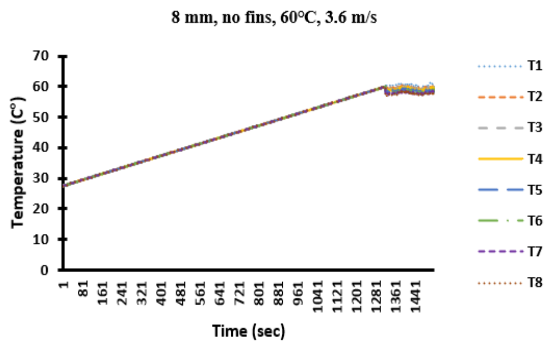

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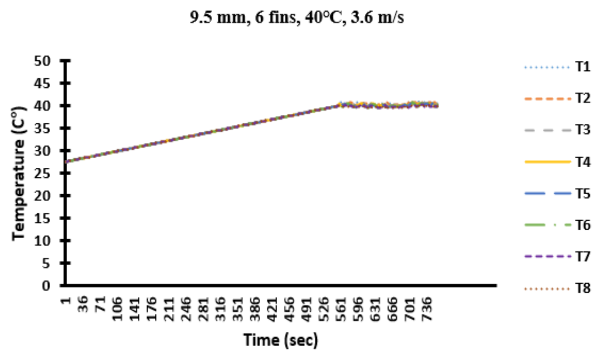


(c)

Figure 6. The temperature steady state of the experiments

3.1.2 Effect of tube diameter

The tube diameter significantly affects the heat transfer coefficient in a fin-and-tube heat exchanger. Generally, larger diameters decrease fluid velocity, reducing turbulence and mixing, which thickens the thermal boundary layer and lowers heat transfer rates. Conversely, smaller diameters increase velocity and turbulence, resulting in a thinner thermal boundary layer and higher heat transfer coefficients [29] [30]. In this study, three different tube diameters were tested, leading to varying average liquid velocities: 2.01 m/s for 6.5 mm, 1.33 m/s for 8 mm, and 0.941 m/s for 9.5 mm. These differences highlight the distinct fluid dynamics associated with each diameter. Experimental results showed in Figure 7 average heat transfer coefficients of 524 W/m^2 ℃ for 6.5 mm, 326.6 W/m^2 ℃ for 8 mm, and 200.6 W/m^2 ℃ for 9.5 mm, demonstrating a clear relationship between heat transfer coefficient and tube diameter.

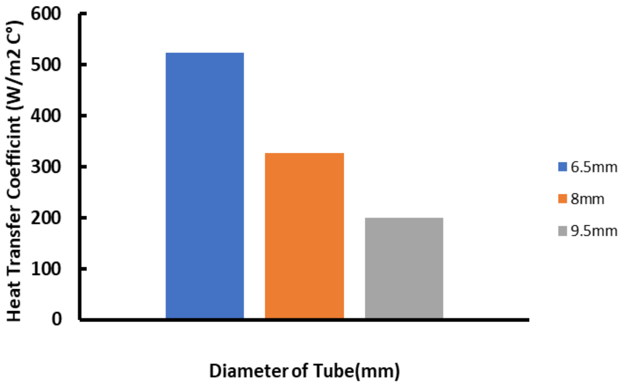


Figure 7. Variation in the heat transfer coefficient based on differences in tube diameter

The findings indicate that the 6.5 mm tube diameter is the most effective choice, as it maximizes the heat transfer coefficient compared to the larger 8 mm and 9.5 mm diameters.

3.1.3 Effect of the number of fins

This study primarily examined the impact of fins on the heat transfer coefficient, which increased slightly with the number of fins. This improvement is due to the larger surface area for heat transfer and enhanced interaction between the liquid and the fins. Adding more fins boosts the surface area, leading to better convective heat transfer [31] [32]. As shown in Figure 8(a), the heat transfer coefficient increased by 6%, 10.77%, and 17% for six, eight, and twelve fins, respectively. In Figure 8(b), the growth rates were 5.50%, 9.32%, and 14.95% with the same fin increases. Figure 8(c) also demonstrated increases of 9%, 16.78%, and 17.52% for six, eight, and twelve fins. These results highlight that a greater number of fins enhances the heat transfer coefficient, indicating improved efficiency in heat exchange between the fluid and the fins.

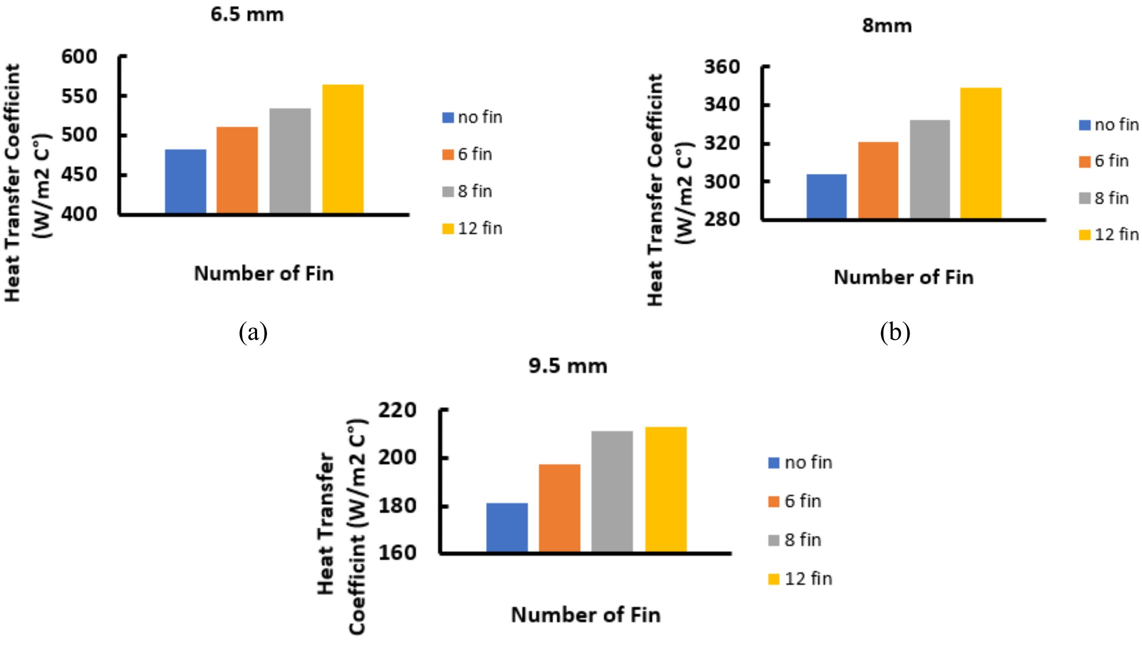
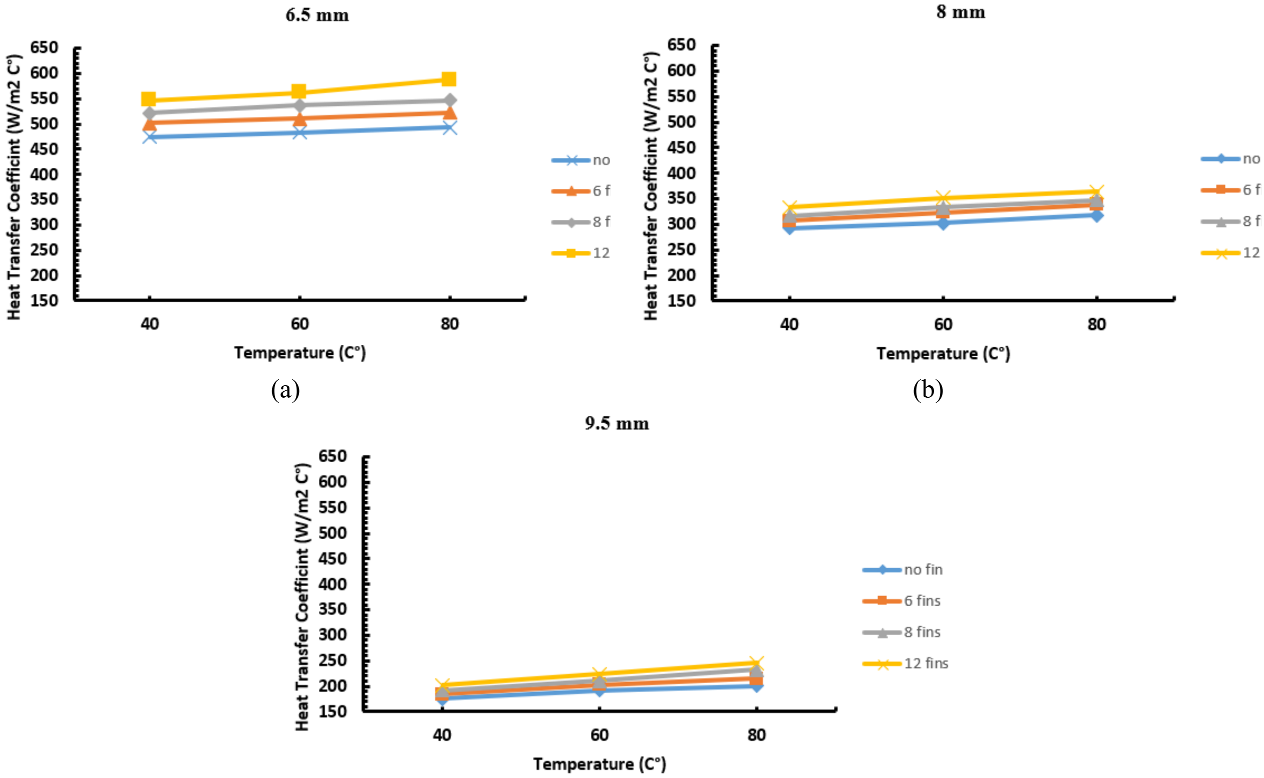


Figure 8. Variation in heat transfer coefficient as a function of the number of fins

The results show that twelve fins provided the best performance, maximizing the heat transfer coefficient. This increased surface area significantly enhanced heat transfer compared to six or eight fins. Thus, twelve fins are the optimal configuration for maximizing the heat transfer coefficient and improving overall heat exchange efficiency.

3.1.4 Effect of inlet temperature

This study examined the impact of inlet temperature on the heat transfer coefficient, finding that higher inlet temperatures enhance heat transfer performance. This improvement is due to several factors: increased thermal driving force from a larger temperature difference, reduced fluid viscosity, and enhanced fluid mixing, all contributing to a higher heat transfer coefficient. Additionally, higher temperatures can lead to a transition from laminar to turbulent flow, further increasing heat transfer rates [26]. Figure 9 shows a slight increase in the heat transfer coefficient with rising inlet temperatures. For the 6.5 mm diameter tube (Figure 9(a)), the average heat transfer coefficient increased by 2.27% from 40 ℃ to 60 ℃ and by 2.64% from 60 ℃ to 80 ℃. A similar trend is seen in Figure 9(b), with increases of 4.34% and 4.5% for the same temperature ranges. The most significant increase occurred in Figure 9(c), with rises of 8.22% and 10.64%.



(a) (b) (c)

Figure 9. Variation in the heat transfer coefficient as a function of inlet temperature

The findings in Figure 9 indicate that 80 ℃ is the optimal inlet temperature for maximizing heat transfer performance in the heat exchanger designs of this study.

3.1.5 Effect of air velocity

This section examines the impact of inlet air velocity on the thermal performance of fin-and-tube heat exchangers. Experiments demonstrated a positive correlation between air inlet velocity and heat transfer coefficient, consistent across different tube diameters, inlet temperatures, and fin counts. Figure 10 illustrates this relationship. The study examined the relationship between heat transfer coefficient and air velocity for three different tube sizes 6.5 mm, 8 mm, and 9.5 mm in a wind tunnel. For the 6.5 mm tube Figure 10(a), it rose from 0.47% to 1.38% as air velocity increased from 2.4 m/s to 3.6 m/s. The 8 mm tube Figure 10(b) showed an increase from 0.59% to 1.56%, while the 9.5 mm tube Figure 10(c) saw a rise from 0.82% to 2.28%. These trends indicate that higher air velocities enhance fluid mixing and turbulence, improving contact with heat transfer surfaces and facilitating convective heat transfer. Additionally, increased velocities reduce the thickness of the thermal boundary layer, lowering heat transfer resistance [23].

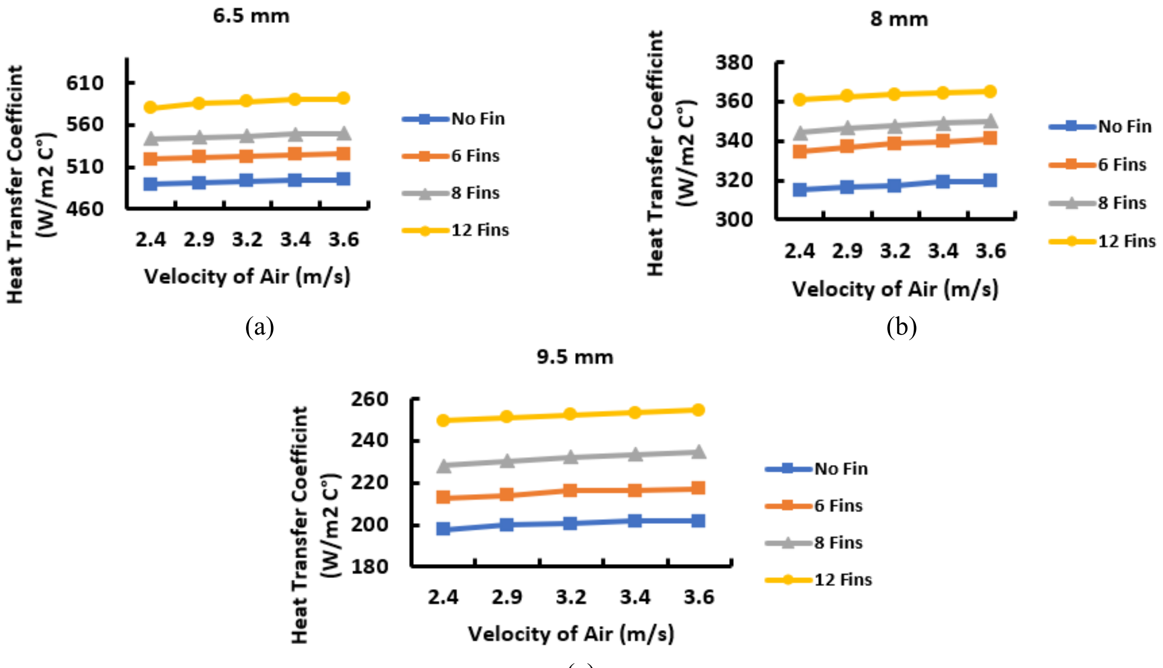


Figure 10. Variation in the heat transfer coefficient based on the velocity of air

The study identified 3.6 m/s as the optimal air velocity for maximizing heat transfer across all three tested tube sizes. At this velocity, the heat transfer coefficients were highest for the 6.5 mm, 8 mm, and 9.5 mm tubes.

3.2 Hydraulic Performance

As is generally known, any heat transfer enhancement is usually accompanied by an additional penalty, such as a pressure drop. In this study, the most important factors affecting the pressure drop are the number of fins and the air velocity. The results presented in this section for the Reynolds numbers 39,000, 47,100, 52,000, 55,250, and 58,500 correspond to the different air velocities of 2.4, 2.9, 3.2, 3.4, and 3.6 m/s, respectively.

3.2.1 Effect of pressure drop

The increase in pressure drop with added fins in a wind tunnel is primarily due to the larger surface area, resulting in higher friction, turbulence, and flow separation. These factors heighten airflow resistance, leading to a measurable pressure drop [33]. Figure 11 shows that as the number of fins increases, the pressure drop rate also rises significantly: a 52.63% increase with six fins, 94.74% with eight fins, and 136.34% with twelve fins, indicating that more fins create greater resistance.

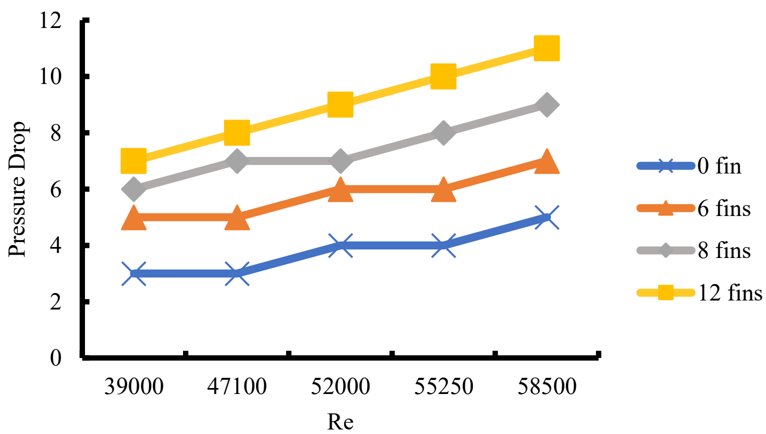


Figure 11. Pressure drop against the Reynolds number

Therefore, to achieve the optimum pressure drop based on the information in Figure 11, the number of fins would need to be reduced.

3.2.2 Effect of friction factor

The heat transfer rate is primarily limited by the thermal resistance on the air side of the heat exchanger. Experiments focused on air-side heat transfer, analyzing dimensionless numbers like Reynolds number and friction factor based on fin count and air velocity in the wind tunnel. As expected, the friction factor increased with more fins but decreased with higher air velocity. Figure 12 illustrates that the friction factor ratio rose with the number of fins, reaching 50.67% with six fins, 85.1% with eight fins, and 123.1% with twelve fins. This rise in the friction factor ratio reduces the overall heat transfer coefficient due to airflow obstruction. The Reynolds number ranged from 39,000 to 58,500, indicating turbulent flow, which typically sees a decreasing friction factor. At the lowest Reynolds number of 39,000, the friction factor ratio was 54.8%, decreasing to 22.3%, 7.4%, and finally 3.79% at the highest Reynolds number of 58,500. This inverse relationship is characteristic of turbulent flow; as air velocity and Reynolds number increase, the boundary layer thins, leading to lower frictional losses and improved heat transfer performance.

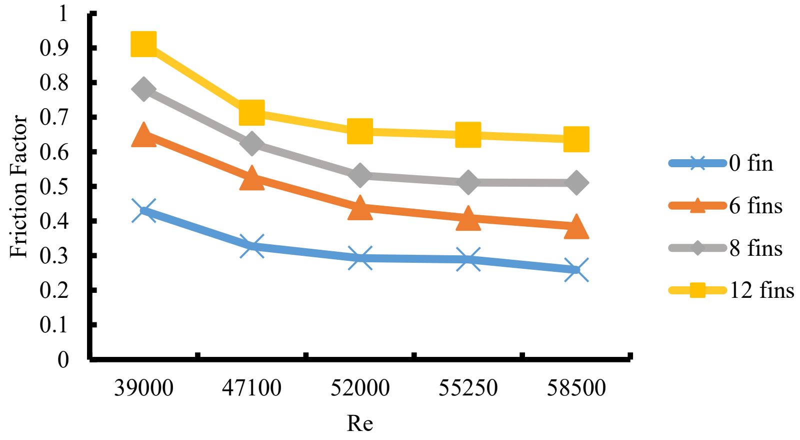


Figure 12. Friction factor against the Reynolds number

3.3 Heat Transfer Performance

3.3.1 Efficiency of heat exchangers

This study evaluated the heat transfer performance of the system by analyzing its effectiveness alongside pressure drop. Experimental tests in a wind tunnel investigated the effects of varying fin numbers and air velocities on pressure drop and heat transfer efficiency. Conducted at a constant temperature of 80°C, twelve experiments varied tube diameters, fin counts, and air velocities.

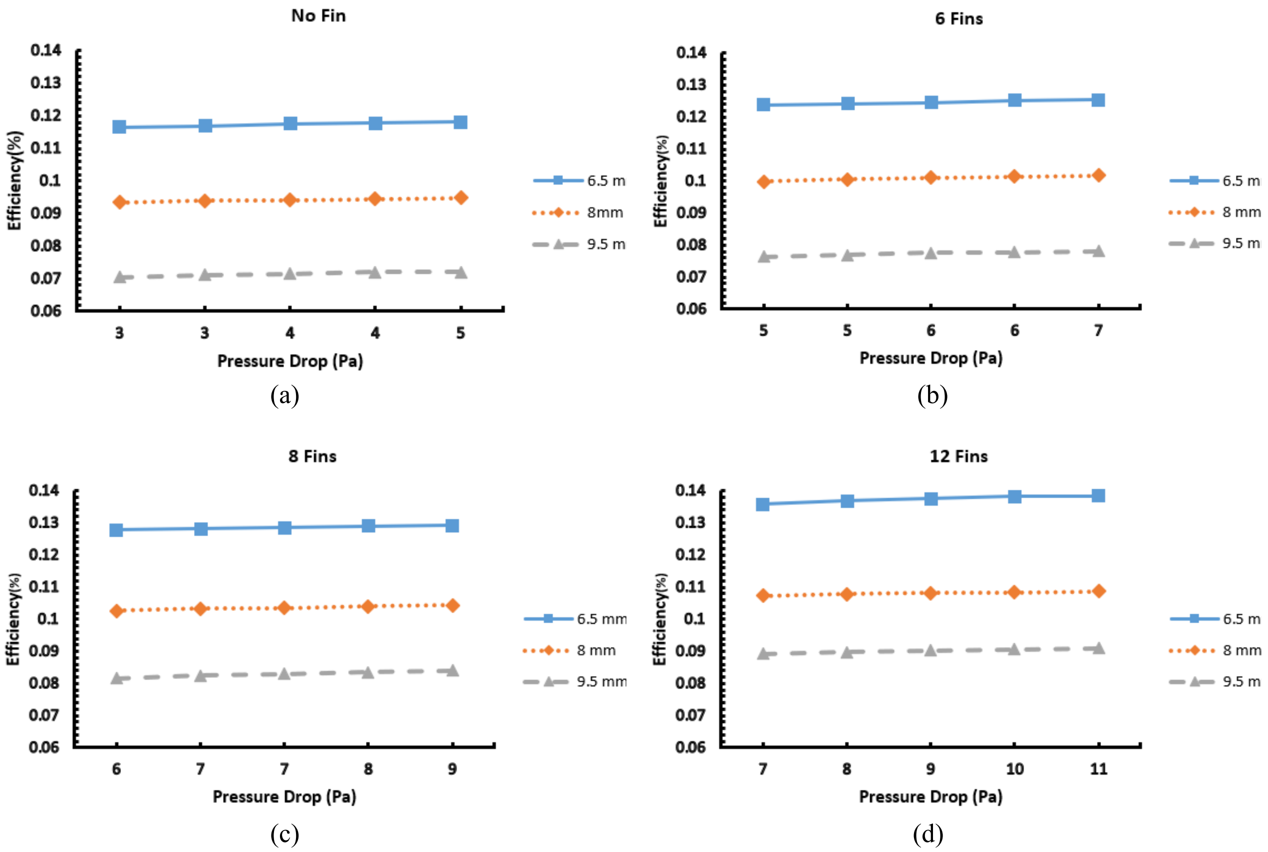


Figure 13. Efficiency of heat exchangers against the pressure drop

Figure 13(a) depicts the percentage change in heat transfer efficiency for experiments performed without fins, revealing efficiencies of 11.74%, 9.41%, and 7.15% for tubes measuring 6.5 mm, 8 mm, and 9.5 mm, respectively. The pressure drop during these experiments was noted to be between 3 and 5 Pascals. When six fins were added, as illustrated in Figure 13(b), there was a relative increase in heat transfer efficiency at the same air speeds, with efficiencies recorded at 12.46%, 10.09%, and 7.74% for the same tube sizes. However, this modification resulted in a higher pressure drop, ranging from 5 to 7 Pascals. The introduction of eight fins, shown in Figure 13(c), further enhanced heat transfer efficiency across five different air speeds, with efficiencies of 12.85%, 10.36%, and 8.3% for the 6.5 mm, 8 mm, and 9.5 mm tubes, respectively. This improvement was accompanied by an increase in pressure drop to a range of 6 to 9 Pascals. Finally, Figure 13(d) presents the results from experiments using twelve fins, which achieved the highest heat transfer efficiencies across the same air speeds, with efficiencies of 13.74%, 10.81%, and 9.02% for the respective tube sizes. This configuration also exhibited the greatest pressure drop, ranging from 7 to 11 Pascals.

The increase in heat transfer efficiency with the addition of fins can be attributed to several factors. Fins enhance the surface area available for heat transfer, which improves heat dissipation from the heated surface to the surrounding air, thus facilitating more effective convective heat transfer. Additionally, fins disrupt boundary layers, thin layers of stagnant air that form next to heated surfaces, thereby reducing thermal resistance and improving efficiency. However, the increased number of fins also leads to higher pressure drops due to the additional resistance encountered by the airflow, resulting from the larger surface area and flow obstructions created by the fins. The data suggests that twelve fins is the best choice to improve heat transfer efficiency while having no fins (no fins) is the optimal configuration to reduce pressure drop in this system.

3.3.2 Thermal performance index

The thermal performance index (TPI) was used in this study to evaluate heat transfer efficiency while maintaining the same pumping power. The goal was to design an optimal heat exchanger that reduces pressure drops and maximizes heat transfer efficiency for energy conservation. Increasing the number of fins from 0 to 12 decreased the minimum flow area and accelerated fluid flow, resulting in a significant rise in pressure drop compared to heat transfer enhancement. As shown in Figure 14, the TPI ranged from 0.01 to 0.032, decreasing with more fins. The lowest TPI value of 1.27% occurred with twelve fins due to high pressure drop, while configurations with six and eight fins had TPI values of 1.44% and 1.77%, respectively. In contrast, the highest TPI value of 2.58% was observed at no fins, indicating that the enhancement in heat transfer provided by the absence of fins outweighs the pressure drop introduced by the finned configurations.

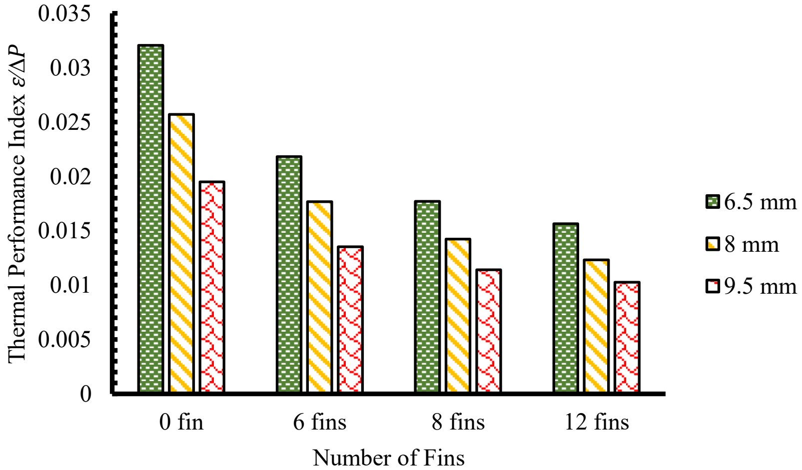


Figure 14. Thermal performance index in relation to the number of fins

Therefore, the results indicate a fundamental trade-off between heat transfer efficiency and the energy needed for pumping. If the main focus is on reducing pumping power, it might be better to operate with fewer fins, potentially even none at all. However, if the aim is to enhance heat transfer, increasing the number of fins would be advantageous despite the likely increase in pumping power. The key takeaway is that the optimal fin configuration depends on the specific priorities and requirements of the system being designed or operated.

4. CONCLUSIONS

This project successfully designed and assembled a compact heat exchanger utilizing carbon nanotube-reinforced polymer fins. These fins were effectively integrated with copper tubes through careful installation and assembly, resulting in a fully consolidated heat exchanger. This design allowed for easy installation and replacement within the wind tunnel during experiments, enabling a thorough investigation of performance. This study revealed important insights into heat transfer and heat exchanger design. The highest heat transfer coefficient recorded was 524 W/m^2 °C, achieved with a tube diameter of 6.5 mm. Using twelve fins made from carbon nanotube-reinforced polymer led to a maximum 17% improvement in the heat transfer coefficient. Raising the inlet temperature to 80°C resulted in a 10.64% increase in heat transfer while increasing the air velocity to 3.6 m/s in the wind tunnel, it went up to 2.28%. On the hydraulic side, the study focused on balancing improved heat transfer with minimizing pumping power. Results showed that the Thermal Performance Index (TPI) decreased as the number of fins increased, with the lowest value of 1.27% at twelve fins and the highest of 2.58% when no fins were used. This indicates that the heat transfer benefits gained by removing fins outweigh the pressure losses caused by adding them. In other words, there's a trade-off between heat transfer efficiency and pumping power, and the best fin configuration depends on the specific needs and priorities of the system. Overall, this study's model stands out for its lightweight construction, corrosion resistance, easy fabrication, and high efficiency, making it suitable for a range of industries. Its compact design is especially beneficial for HVAC systems, while its corrosion resistance and ease of cleaning make it ideal for food processing applications where hygiene is critical.

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CONFLICT OF INTEREST

The authors declare that there are no conflicts of interest associated with this publication.

AUTHORS CONTRIBUTION

M.H. Alshameri: (Conceptualization, Literature, Investigation, Data Collection)

A.A. Azizuddin: (Supervision, Resources, Review)

A. Arshad: (Supervision, Review)

Haetham G. Mohammed: (Literature, Investigation)

Safwan Sadeq: (Literature, Investigation)

REFERENCES

[1] S Han, X Li, Z Liu, B Zhang, C He, and Q Chen, "Thermal-economic optimization design of shell and tube heat exchanger using an improved sparrow search algorithm," *Thermal Science and Engineering Progress*, vol. 45, p. 102085, 2023.

[2] S Boldyryev, T Gil, and M Ilchenko, "Environmental and economic assessment of the efficiency of heat exchanger network retrofit options based on the experience of society and energy price records," *Energy*, vol. 260, p. 125155, 2022.

[3] K. Rajski and J. Danielewicz, "Heat transfer and heat recovery systems," *Energies*, vol. 16, no. 7, p. 3258, 2023.

[4] M. Hamood, A. N. Oumer, A. Abd Aziz, M. Ishak, J. P. Siregar, and O. Mamat, "Heat transfer analysis in compact heat exchanger with fibre reinforced plate," *IOP Conf Ser Mater Sci Eng*, vol. 1078, no. 1, p. 012036, 2021.

[5] M. Chaanaoui, K. Ettahi, S. Abderafi, S. Vaudreuil, and T. Bounahmidi, "Comparative analysis between optimum configurations of finned tube heat exchanger: Application for solar drying," *Case Studies in Thermal Engineering*, vol. 22, p. 100750, 2020.

[6] K. Barquín and A. Valencia, "Comparison of different fin and tube compact heat exchanger with longitudinal vortex generator in CFU-CFD configurations," *International Journal of Heat and Technology*, vol. 39, no. 5, pp. 1523–1531, 2021,

[7] Q Li, G Flamant, X Yuan, P Neveu, and L Luo, "Compact heat exchangers: A review and future applications for a new generation of high temperature solar receivers," *Renewable and Sustainable Energy Reviews*, vol. 16, no. 2, 721–733, 2012.

[8] M. A. Arie, A. H. Shooshtari, R. Tiwari, S. V. Dessiatoun, M. M. Ohadi, and J. M. Pearce, "Experimental characterization of heat transfer in an additively manufactured polymer heat exchanger," *Applied Thermal Engineering*, vol. 113, pp. 575–584, 2017.

[9] M. Nakada and Y. Miyano, "Accelerated testing for long-term fatigue strength of various FRP laminates for marine use," *Composites Science and Technology*, vol. 69, no. 6, pp. 805–813, 2009.

[10] X. He and Y. Wang, "Recent advances in the rational design of thermal conductive polymer composites," *Industrial & Engineering Chemistry Research*, vol. 60, no. 3, pp. 1137–1154, 2021.

[11] A. R. Cardenas, C. S. Davis, and A. M. Marconnet, "Liquid metal concentration effects on thermal and mechanical properties in elastomers," in *2024 23rd IEEE Intersociety Conference on Thermal and Thermomechanical Phenomena in Electronic Systems (ITherm)*, pp. 1–2, 2024.

[12] X. Meng, H. Yu, L. Wang, X. Wu, and B. U. Amin, "Recent progress on fabrication and performance of polymer composites with highly thermal conductivity," *Macromolecular Materials and Engineering*, vol. 306, no. 11, p. 2100434, 2021.

[13] A. Li, C. Zhang, and Y. F. Zhang, "Thermal conductivity of graphene-polymer composites: Mechanisms, properties, and applications," *Polymers (Basel)*, vol. 9, no. 9, pp. 1–17, 2017.

[14] D. K. Lee, J. Yoo, H. Kim, B. H. Kang, and S. H. Park, "Electrical and thermal properties of carbon nanotube polymer composites with various aspect ratios," *Materials*, vol. 15, no. 4, p. 1356, 2022.

[15] A. Pathak, J. AuBuchon, D. Brei, J. Shaw, J. Luntz, and S. Jin, "Carbon nanotube (CNT) fins for enhanced cooling of shape memory alloy wire," In *Behavior and Mechanics of Multifunctional and Composite Materials 2008,* vol. 6929, p. 69291K, 2008.

[16] Z. Han and A. Fina, "Thermal conductivity of carbon nanotubes and their polymer nanocomposites: A review," *Progress in Polymer Science*, vol, 36, no. 7, 914–944, 2011.

[17] B Kumanek and D Janas, "Thermal conductivity of carbon nanotube networks: A review," *Journal of Materials Science*, vol. 54. no. 13, 8929–8964, 2019.

[18] X. Zheng, B. R. Kim, S. J. Hong, J. G. Lee, and C. W. Park, "Heat transfer analysis of carbon fiber-reinforced corrugated polymer plate heat exchangers," *Applied Thermal Engineering*, vol. 244, p. 122684, 2024.

[19] S. G. Mavridou, E. Konstandinidis, and D. G. Bouris, "Experimental evaluation of pairs of inline tubes of different size as components for heat exchanger tube bundles," *International Journal of Heat and Mass Transfer*, vol. 90, pp. 280–290, 2015.

[20] J. Cobian-Iñiguez, A. Wu, F. Dugast, and A. Pacheco-Vega, "Numerically-based parametric analysis of plain fin and tube compact heat exchangers," *Applied Thermal Engineering*, vol. 86, pp. 1–13, 2015.

[21] J. Jabbour, S. Russeil, M. Mobtil, D. Bougeard, M. F. Lacrampe, and P. Krawczak, "High performance finned-tube heat exchangers based on filled polymer," *Applied Thermal Engineering*, vol. 155, pp. 620–630, 2019.

[22] S. A. Marzouk, M. M. Abou Al-Sood, E. M. S. El-Said, M. M. Younes, and M. K. El-Fakharany, "A comprehensive review of methods of heat transfer enhancement in shell and tube heat exchangers," *Journal of Thermal Analysis and Calorimetry*, vol. 151, no. 8, 5037–5065, 2023.

[23] A. Wibolo, A. A. I. Made, "Experimental study of air velocity effect of heat transfer in distillation condenser," *IOP Conference Series: Materials Science and Engineering*, vol. 1175, no. 1, p. 012014, 2021.

[24] J. Zhuang, M. Liu, H. Wu, and J. Wang, "Designing an environmental wind tunnel cooling system for high-speed trains with air compression cooling and a sensitivity analysis of design parameters," *Entropy*, vol. 25, no. 9, p. 1312, 2023.

[25] M. Khaled, "Enhancing thermal performance of water–air cross-flow heat exchangers through upstream nozzle design and unit division" *International Journal of Thermofluids*, vol. 27, p. 101223, 2025.

[26] D. A. Kamel, M. A. Gadalla, O. Y. Abdelaziz, M. A. Labib, and F. H. Ashour, "Temperature driving force (TDF) curves for heat exchanger network retrofit – A case study and implications," *Energy*, vol. 123, pp. 283–295, 2017.

[27] B. Gong, L. B. Wang, and Z. M. Lin, "Heat transfer characteristics of a circular tube bank fin heat exchanger with fins punched curve rectangular vortex generators in the wake regions of the tubes," *Applied Thermal Engineering*, vol. 75, pp. 224–238, 2015.

[28] F. Tagliaferri, L. Carrera, A. Albertini, M. Invernizzi, and S. Sironi, "The development and optimization of a new wind tunnel design for odour sampling," *Atmosphere*, vol. 15, no.10, p. 1181, 2024.

[29] L. Zhao, J. Chen, and G. Duan, "Turbulent flow in an I–L junction: Impacts of the pipe diameter ratio," *Physics of Fluids*, vol. 36, no. 2, p. 025122, 2024.

[30] R. A. Hussein, A. A. Alwan, and I. M. Abed, "Experimental investigation and analysis of heat transfer rate in conical tube heat exchanger: A novel enhancement approach," *Journal of Engineering Science and Technology*, vol, 16, no. 2, 61–74, 2021.

[31] K. Shahril, N. B. M. Kasim, and M. Sabri, "Heat transfer simulation of motorcycle fins under varying velocity using CFD method," *IOP Conference Series: Materials Science and Engineering*, vol. 50, no. 1, p. 012043, 2013.

[32] L. Miao, R. Wan, H. Wu, Z. Liu, and S. Wang, "Heat transfer and pressure drop characteristic research of sine wavy flying-wing fins," *Scientific Reports*, vol. 13, p. 15589, 2023.

[33] K. Rauthan, F. Guzzomi, A. Vafadar, K. Hayward, and A. Hurry, "Experimental investigation of pressure drop performance of smooth and dimpled single plate-fin heat exchangers," *Metals*, vol. 11, no. 11, p. 1757, 2021.