Jurnal Polimesin
Department of Mechanical Engineering
State Polytechnic of Lhokseumawe
http://e-jurnal.pnl.ac.id/polimesin

e-ISSN	: 2549-1999	No.	: 6	Month	: December
p-ISSN	: 1693-5462	Volume	: 22	Year	: 2024

Article Processing Dates: Received on 2024-08-16, Reviewed on 2024-11-23, Revised on 2024-12-09, Accepted on 2024-12-12 and Available online on 2024-12-30

Numerical study on the effect of box and polygon geometry in fin and tube heat exchanger on fluid flow and heat transfer

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Abstract

Currently, fin and tube heat exchangers are widely used in various engineering applications, including modern heat exchangers, automotive radiators, and Air Conditioning (AC) systems such as evaporators, and condensers. Enhancing their performance necessitates innovative designs, advanced application, and optimizes geometries to improve heat transfer efficiency. This study investigates the effect of box and polygon geometries on fluid flow and heat transfer in a split Air Conditioner (AC) fin and tube heat exchanger using simulation software. The research examines two tube arrangement-inline and staggered-across different fluid velocities (0.5 m/s, 1 m/s, 1.5 m/s, and 2.5 m/s) and heat flux values (100 W/m^2, 125 W/m^{2,} 125 W/m^{2,} and 150 W/m^2). The numerical study revealed that the best thermal and hydraulic performance of the fin and tube heat exchanger, based on geometry variations between box and polygon tubes, was achieved with the polygon tube geometry, which resulted in a lower temperature around 23.41°C. This temperature confirmed an increase in heat transfer coefficient by approximately 5% and Nusselt number by about 3%. The best performance overall, considering both thermal and hydraulic aspects, was observed in the inline arrangement, especially for the polygon tube, which resulted in a lower temperature of around 26.38°C. This confirmed an improvement in the heat transfer coefficient by about 4% and the Nusselt number by 2.5%.

Keywords:
Fin, tube, heat exchanger, box, polygon, inline, staggered.

1 Introduction

Fin and tube heat exchangers are widely used in various engineering applications, such as modern heat exchangers, automotive radiators, air conditioning evaporators, and condensers. To improve their performance, there is a continuous need for innovative designs, new applications, and advanced geometries that can enhance heat transfer efficiency (Bhuiyan & Islam, 2016) (Min et al., 2014; Sariyusda, 2009). Many researchers have investigated ways to enhance the performance and efficiency of fin and tube heat exchangers for practical applications (Bhuiyan et al., 2013).

For instance, several studies have focused on different strategies to improve the efficiency and effectiveness of these heat exchangers. One commonly applied approach is the use of Vortex Generators (VG), which create swirling flow patterns that improve heat transfer. However, while vortex generators enhance heat transfer, they also lead to increased turbulent pressure, which requires additional power to maintain system performance (Fiebig, 1998) (Yang et al., 2021). Furthermore, optimizing the geometry and material selection plays a significant role in improving performance. Research on flat fins combined with staggered tube configurations has shown that increasing the thermal conductivity of materials can significantly enhance heat transfer efficiency. Additionally, studies have demonstrated that air velocity has a considerable impact on heat transfer efficiency, with lower air velocities often yield better results in certain conditions (Nakkaew et al., 2019).

On the other hand, research on shell and tube heat exchangers is also relevant, as it has been found that adhering to construction standards, such as TEMA and ASME, can improve the performance of these systems. The addition of fins to the tubes has been proven effective in enhancing heat transfer, although the resulting increase in turbulent pressure must be carefully considered (Posner et al., 2003). Therefore, to further improve the efficiency and performance of fin and tube heat exchangers, it is essential to combine various approaches, including innovations in geometric design, material selection, and a detailed analysis of the factors influencing heat transfer. Mathematical modeling is often employed to assess the performance of these heat exchangers across different geometric configurations, such as hexagonal designs, to evaluate their reliability and effectiveness (Cárdenas et al., 2017).

A cam-shaped tube bundle with a staggered arrangement was characterized experimentally with a pitch ratio comparison of 1.5 and 2 at Reynolds numbers ranging from 27,000 to 42,000. The results of this study showed a 93% increase in the Nusselt number (Bayat et al., 2014). A numerical study on fin and tube heat exchangers, varying the tube spacing to 16 mm and fin thickness to 0.8, 0.12, 0.16, and 0.2, was conducted to compare the heat transfer rate and pressure drop (Q/ΔP). The numerical study results indicated a 6-8% comparison in the heat transfer rate and pressure drop (Q/ΔP) (Lu et al., 2011).

A numerical study on flat tube banks was conducted to investigate the heat transfer performance at low Reynolds numbers, specifically in the range of 500 to 800. The results indicated that the Nusselt number increased by approximately 50% (Sahel et al., 2019) (Fullerton & Anand, 2017). Experimental and numerical investigations of the flow and heat transfer characteristics around egg-shaped tubes, with varying angles of 0.2°, 0.4°, 0.6°, 0.8°, and 10°, were carried out to determine the Performance Evaluation Criteria (PEC) values (Dowson, 1998) (Fiebig, 1998). The optimal results from this study were observed for the egg-shaped tube with a 10° angle, yielding a PEC value of 2. This suggests that further increasing the variation in the egg-shaped angle could enhance the PEC value (Munawir et al., 2017)., 2017) (Min et al., 2014).

Fin and tube heat exchangers have been analyzed to assess the flow and heat transfer characteristics under varying fluid flow velocities of 3 m/s, 4 m/s, 5 m/s, 6 m/s, and 7 m/s. The best performance was observed at a fluid velocity of 7 m/s, resulting in an 18.4% increase in the Nusselt number (Eleiwi et al., 2020). Heat flux variations of 15 W/m², 20 W/m², and 25 W/m² were characterized in the context of mixed convection from horizontal isothermal elliptic cylinders, applied across different tube angle configurations to investigate heat transfer performance. The findings showed a 16% and 20% increase in heat transfer coefficient and Nusselt number, respectively (Alnakeeb et al., 2014; Rosidi et al., 2022).

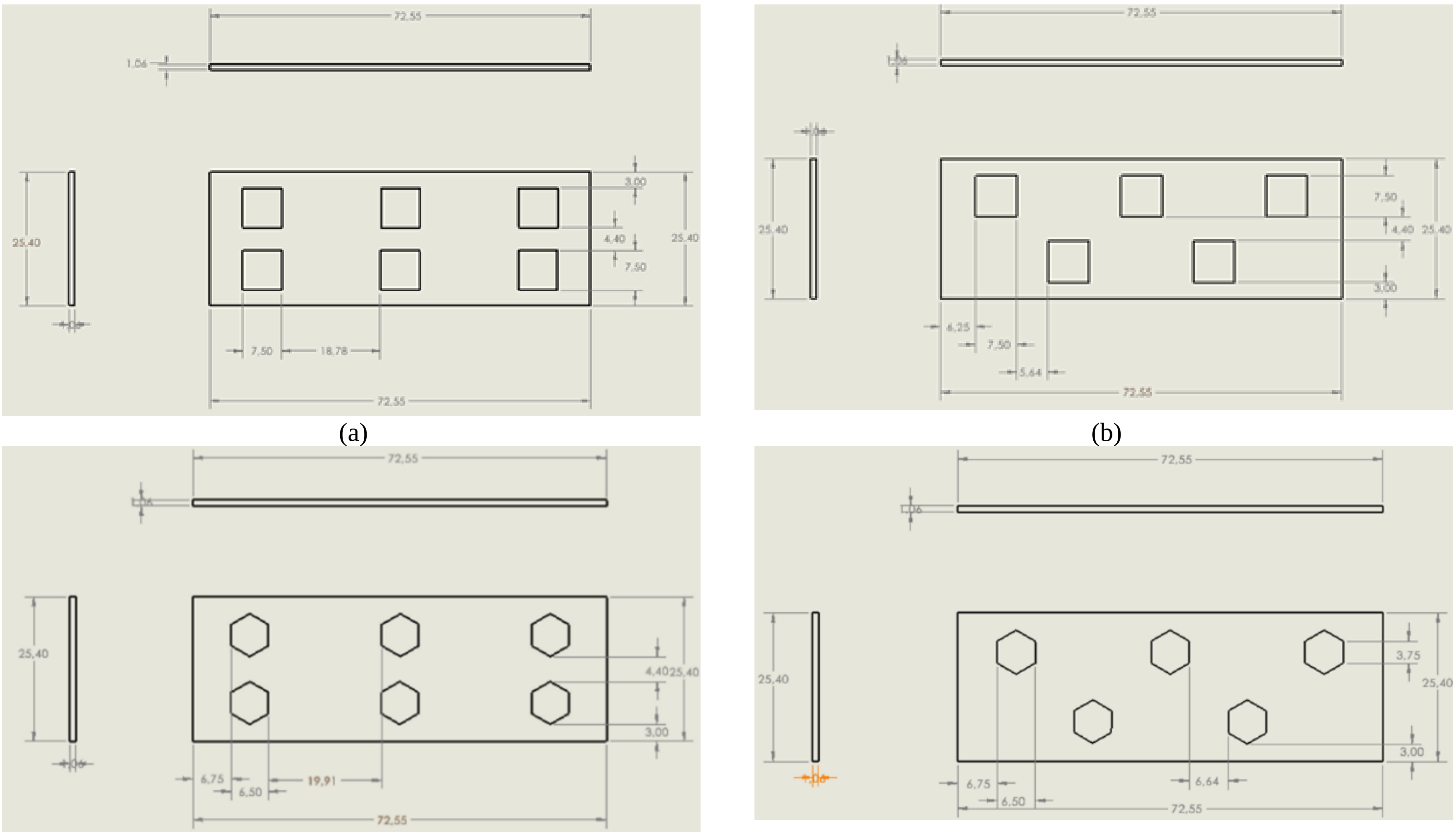
Various experimental and computational studies have been conducted on fin and tube heat exchangers to evaluate their heat transfer efficiency, performance, and reliability under different conditions, including variations in geometry configurations, geometric ratios, flow velocities, and heat flux values. This study will utilize simulation techniques with Computational Fluid Dynamics (CFD) software (Sariyusda, 2009; Versteeg, 2007) to investigate these parameters. The simulations will explore different tube geometries, such as tube box and polygon shapes, arranged in both inline and staggered configurations. Fluid flow velocities of 0.5 m/s, 1 m/s, 1.5 m/s, 2 m/s, and 2.5 m/s, along with heat flux values of 100 W/m², 125 W/m², and 150 W/m², will be considered to examine the fluid flow and heat transfer characteristics (Syuhada & Edhy, 2023) (Gaos et al., 2024).

The selection of tube box and polygon geometries in this study is motivated by several critical engineering factors. Polygon tubes are known for their superior heat dissipation properties compared to tube box designs, which allow for more efficient cooling by promoting better airflow around the tubes (Rosidi et al., 2022). This results in a reduced average surface temperature and enhances the overall performance of the heat exchanger. Additionally, the polygon geometry fosters greater turbulence, which improves fluid mixing and, consequently, heat transfer efficiency. Studies also indicate that polygon tubes contribute to lower pressure drops, thus improving hydraulic efficiency. Finally, the combination of inline and staggered tube arrangements for both geometries offers flexibility in design, allowing for the optimization of performance across various configurations. Therefore, the choice of tube box and polygon geometries is based on a balance of thermal performance and hydraulic efficiency considerations.

2 Research Methodology

2.1 Optimization Method for Fin and Tube Heat Exchanger

Fig. 1: (a) displays the top, front, and side views of the fin and tube heat exchanger geometry with an inline tube box configuration; (b) shows the geometry of a fin and tube heat exchanger with a staggered tube box configuration; (c) illustrates the geometry of a fin and tube heat exchanger with an inline polygon tube configuration; and (d) depicts the geometry of a fin and tube heat exchanger with a staggered polygon tube configuration. These geometries were developed using SolidWorks software, following the guidelines outlined in the study by Bhuiyan & Islam (Bhuiyan & Islam, 2016).



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

Fig. 1. (a) Geometry of the fin and tube heat exchanger with tube box inline arrangement, (b) geometry of the fin and tube heat exchanger with tube box staggered arrangement, (c) geometry of the fin and tube heat exchanger with polygon tube inline arrangement, (d) geometry of the fin and tube heat exchanger with polygon tube staggered arrangement.

2.2 Design of Fin and Tube Heat Exchanger

The design in this study is a modification of the conventional fin and tube heat exchanger, which initially used circular tube geometry. The tube geometry was then modified to box and polygon shapes, arranged in two configurations: inline and staggered. The geometric parameters of the fin and tube heat exchanger were specified: fin thickness (t = 1.06 mm), fin length (L = 72.55 mm), fin width (l = 25.40 mm), and the number of fins (n = 6 pieces). These specifications are illustrated in Fig. 1 (a), (b), (c), and (d).

2.3 Computational Domain Numerical Study

The computational domain is established using a control volume approach, which discretizes the governing equations with a first-order upwind scheme to improve accuracy. The upstream region of the computational domain is defined with dimensions of 1.6 mm × 29.20 mm × 25.40 mm, while the downstream region measures 1.6 mm × 217.65 mm × 25.40 mm. The boundary conditions for the walls are illustrated in Fig. 2, which presents the numerical analysis of single-phase flow through a box-shaped tube under steady-state conditions. The computational domain in this study is based on a Cartesian coordinate system, where the X-axis corresponds to the flow direction (streamwise), the Y-axis is perpendicular to the flow direction (spanwise), and the Z-axis is normal to the XY plane.

The Reynolds-averaged Navier-Stokes equations and the continuity equation are solved using the Realizable k-ε turbulence model, chosen for its efficiency and accuracy in solving the governing equations while considering the assumptions of the selected viscous model (Safi'i et al., 2021). The governing equations within the three-dimensional computational domain for the fin and tube heat exchanger are convection equations, expressed in Cartesian coordinates. The Reynolds-averaged continuity and Navier-Stokes equations are written in tensor form as Eq. 1-Eq. 5.

Inlet

$$u = u_{in}, v = w = 0, T = T_{in} \tag{1}$$

Outlet

$$\frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial w}{\partial x} = \frac{\partial T}{\partial x} = 0 \tag{2}$$

Wall

$$u = v = w = 0, q = 0 \tag{3}$$

Hot tube

$$u = v = w = 0, T = T_{tube} \tag{4}$$

Symmetry

$$w = \frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = \frac{\partial T}{\partial y} = 0 \tag{5}$$

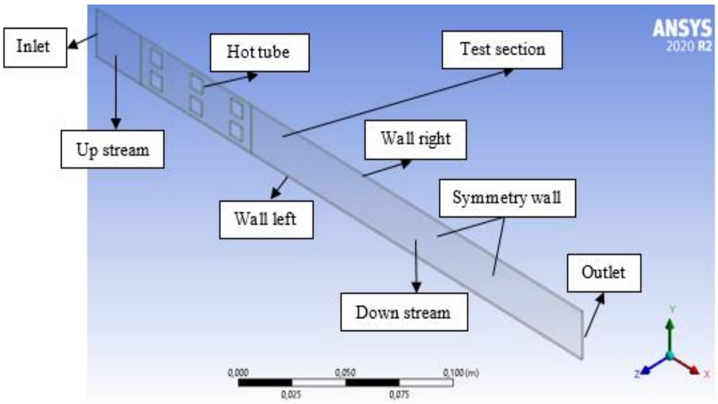


Fig. 2. Computational domain.

2.4 Grid Independence Test

The grid independence analysis was conducted to determine the optimal point for the average tube temperature based on the experimental results. This analysis involved varying the number of mesh cells in the computational domain, with values of 41,070, 52,355, 61,027, 70,422, and 81,659 cells. The simulation setup procedure followed the steps outlined by Bhuiyan et al. (2013) (Bhuiyan et al., 2013). The results of the grid analysis are presented in a graph illustrating the relationship between grid number and heat transfer coefficient, as shown in Fig. 3. The numerical results from the grid analysis revealed the following relative errors: ±0.03% between grids 41,070 and 52,355, ±0.02% between grids 52,355 and 61,027, ±0.0012% between grids 61,027 and 70,422, and ±0.001% between grids 70,422 and 81,659.

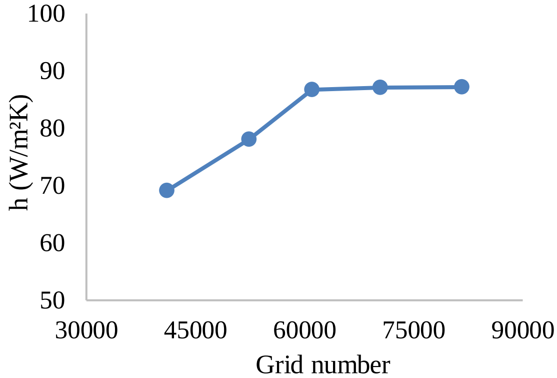


Fig. 3. Graph of h values versus the number of grids.

2.5 Validation

The validation was conducted using a numerical approach and compared with experimental data from A. A. Bhuiyan et al. (2016) (Bhuiyan et al., 2013) on a fin and tube heat exchanger under forced convection. The purpose of this validation was to determine the Colburn factor (j) for the proposed system with varying airflow velocities. The validation results are shown in Fig. 4, which illustrates the relationship between the Reynolds number and the Colburn factor (j).

The experimental data revealed a Colburn factor (j) of 0.0964, while the simulation provided a value of 0.0979, with the largest relative error in thermal resistance being 1.56%. This comparison indicates that the numerical model demonstrates good accuracy and can be trusted for predicting the system's performance.

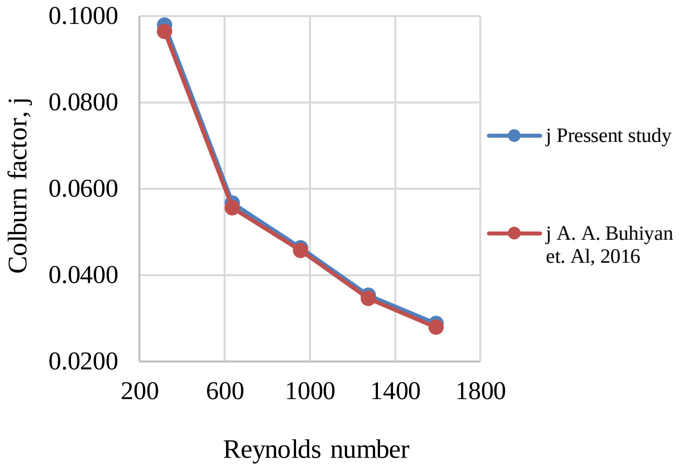


Fig. 4. Colburn factor, j vs. Reynolds number.

For various Reynolds numbers, the largest relative error was found to be 1.56%, with a Colburn factor, jjj, of 0.0979. This result indicates good agreement between the experimental data from the literature and the ongoing numerical study being validated. The validation of the fin and tube heat exchanger against the experiment by Bhuiyan & Islam (Bhuiyan & Islam, 2016), as well as the numerical study of the fin and tube heat exchanger with varying flow velocities, demonstrates that the Colburn factor, jjj, is relatively low but the heat transfer rate is high, as seen from the surface temperature values on the tube.

The heat transfer simulation using the forced convection method is conducted under steady air conditions with the assumption of constant thermal physical properties. In addition, the convective heat transfer between the fin-and-tube heat exchanger interface and the air is considered in this study. A convergence criterion of 10^{-4} for flow and 10^{-6} for energy is applied. The air properties, acting as the coolant in this numerical study, are: density ρ = 1.2096 kg/m³, specific heat capacity Cp = 1005 J/kg.K, viscosity $\mu = 1.915\times10^{-5}$ kg/ms, thermal conductivity k = 0.0261 W/m.K, and molecular weight 28.966 kg/kmol.

The temperature data collection on the tube is explained in Fig. 5, which presents the report plot definition during the simulation, focusing on the average temperature across the entire tube. During the simulation, the inlet temperature is set at 25°C, with a turbulence intensity of 5% and a turbulent viscosity ratio of 10. To ensure effective forced convection conditions in the fin-and-tube heat exchanger, the test region on the heat exchanger may be influenced by airflow disturbances. This means that flow disturbances around the fin-and-tube heat exchanger can affect the temperature distribution, as unstable or turbulent air flow may cause higher temperature fluctuations in certain parts of the system. This influence is critical to note because irregular air circulation can affect the overall system performance, particularly in terms of heat transfer efficiency.

In this simulation, an aluminum alloy 6061 is used for the fin-and-tube heat exchanger module, with an assumed thermal conductivity of k = 168 W/m.K and input power values of Q = 100 W, 125 W, and 150 W. For boundary conditions, no-slip walls are defined on the fin surfaces. To ensure that the computational domain is adequately resolved and numerical uncertainty is minimal, a grid independence test is performed. This test helps refine the grid size progressively until an acceptable convergence is achieved. This ensures that the maximum temperature in the fin-and-tube heat exchanger is reached with sufficient accuracy.

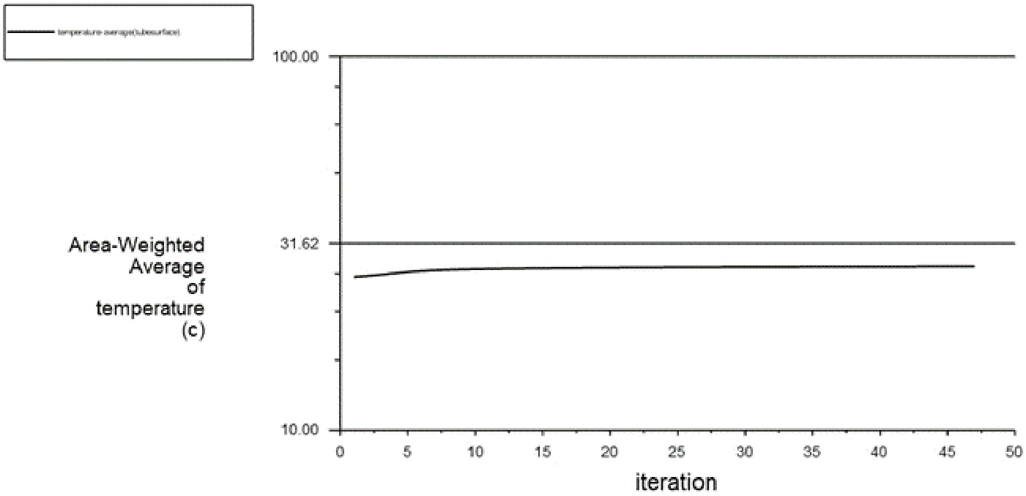


Fig. 5. Plot of the average wall tube temperature measurement.

2.6 Numerical Data Reduction

The simulation results and contour plots were obtained during the post-processing stage in the results menu. The kinematic viscosity is determined based on the correlation between dynamic viscosity divided by the air density, as shown in the Eq. 6.

$$\mu = \frac{v}{\rho} \quad (6)$$

Thermal diffusivity is the ratio of the time derivative to temperature. Thermal diffusivity can also be referred to as a measure of thermal inertia. In materials with high thermal diffusivity, heat moves quickly because the material conducts heat relative to its volumetric heat capacity. The thermal diffusivity value used in the CFD software is 0.0000218684 m²/s at a temperature of 25°C. The Prandtl number is a dimensionless number that represents the relationship between kinematic viscosity and thermal diffusivity, as expressed in the Eq. 7.

$$P_r = \frac{\mu}{\alpha} \quad (7)$$

The data required for the heat transfer performance analysis of the fin-and-tube heat exchanger includes the average temperature of the surface of the fin-and-tube heat exchanger to determine the heat transfer coefficient (h) and pressure drop (ΔP). The average air velocity in the fin-and-tube heat exchanger duct is set between 0.5-2.5 m/s, and the number of oblique fins consists of 7 rows. The average Reynolds number is Eq. 8.

$$R_e = \frac{\rho . D_h . U_\infty}{\mu} \quad (8)$$

The hydraulic diameter is defined by the Eq. 9.

$$D_h = \frac{4s.H}{2(H+s)} \quad (9)$$

The surface area of the fin-and-tube heat exchanger is defined across the entire surface of the tube. This area can be calculated using the Eq. 10.

$$A_s = (nLH)+(PL) \quad (10)$$

Pressure drop through the OFHS is calculated by taking the difference between the static pressure at the air inlet and outlet, thus the pressure drop equation is Eq. 11.

$$\Delta P = P_{in} - P_{out} \quad (11)$$

Thermal resistance and the heat transfer coefficient can be calculated using the Eq. 12 and Eq. 13.

$$R_{th} = \frac{T - T_\infty}{Q} \quad (12)$$

$$h = \frac{Q}{A_s(T_{ave} - T_\infty)} \quad (13)$$

Nusselt number can be calculated using the Eq. 14.

$$N_u = \frac{h.D_h}{k} \quad (14)$$

3 Results and Discussion

This study aims to investigate the characteristics of fluid flow and heat transfer in a fin-and-tube heat exchanger with tube geometries in the form of a box and polygon, using both inline and staggered tube arrangements. The box and polygon tube geometries are expected to generate vortices and swirl flow, thereby enhancing the convective heat transfer rate on the air side.

3.1 Effect of Box and Polygon Tube Shapes with Inline and Staggered Arrangements on Thermal Aspects

This analysis aims to discuss the impact of tube shapes, namely box and polygon tubes, with inline and staggered arrangements on the thermal aspects of a fin-and-tube heat exchanger. This device is designed to transfer heat between two fluids at different temperatures, with the hot fluid flowing through the tubes while the cold fluid flows through the gaps in the fins. The shape and arrangement of the tubes play a crucial role in heat transfer efficiency, where the typical perforated fin design can increase surface porosity and boundary layer dissipation, thereby enhancing heat transfer between the fluids.

In this context, the analysis examines how the tube shape, whether box or polygon, and the inline and staggered arrangements influence the thermal performance of the device. The numerical study we conducted shows that the polygon tube, with a specific arrangement, can provide better performance than the box tube, making this analysis important for understanding the factors that affect heat transfer efficiency. The presence of box and polygon tube geometries in the fin-and-tube heat exchanger arranged either inline or staggered, results in periodic secondary flows that repeatedly refresh the thermal boundary layer.

Consequently, flow mixing and heat advection increase. Furthermore, the influence of box and polygon tube geometries on the fin-and-tube heat exchanger can generate high turbulence intensity, leading to excellent fluid mixing in both primary and secondary flows, thereby significantly reducing the thermal load on the heat exchanger, especially in the inline arrangement. This is evidenced by the uniform temperature distribution across the tubes.

Fig. 6 shows the graph of the relationship between the Reynolds number and the average surface temperature of the box and polygon tube geometries, arranged in both inline and staggered configurations. The best overall result from this study, in terms of average surface temperature, was found in the polygon tube geometry with both inline and staggered tube arrangements, with the highest average temperature of 26.55°C and the lowest of 26.06°C.

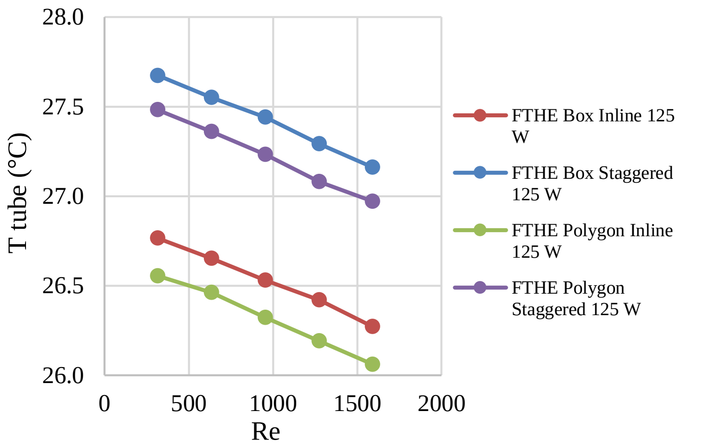


Fig. 6. Graph of the relationship between Reynolds number and tube temperature in a fin-and-tube heat exchanger with box and polygon tube geometries, arranged inline and staggered, at various air flow velocities with Q = 125 W.

This is due to the polygon tube geometry having a much better heat dissipation surface area compared to the inline or staggered box tube geometry, which allows fresh air to enter more freely to cool the tubes. The airflow velocity plays a significant role in the cooling process of the fin-and-tube heat exchanger, as evidenced by the significant impact of increasing the flow rate from 0.5 m/s to 2.5 m/s, which results in a notable increase in the heat transfer coefficient.

Fig. 7 shows the relationship between Reynolds number (Re) and heat transfer coefficient (h) in a fin-and-tube heat exchanger with two types of tube geometries, namely box and polygon, and two tube arrangements, inline and staggered, at varying air flow velocities with a heat input of 125 W. The graph shows that the value of h increases as the Reynolds number increases. This indicates that the faster the airflow, the more efficient the heat transfer.

Regarding the tube geometry, the polygon-shaped tube shows higher h values compared to the box-shaped tube at all tested Re values. This suggests that the polygon geometry is more effective in enhancing heat transfer. In terms of tube arrangement, the staggered configuration yields slightly higher h values compared to the inline arrangement due to the more turbulent fluid flow, which increases heat transfer efficiency.

Thus, it can be concluded from Fig. 7 that the combination of polygon geometry and staggered arrangement provides the best performance in terms of heat transfer among all the configurations tested. Therefore, to improve heat transfer in a fin-and-tube heat exchanger, the use of polygon geometry with a staggered arrangement is recommended. This configuration can enhance the heat exchanger's efficiency and achieve higher heat transfer targets.

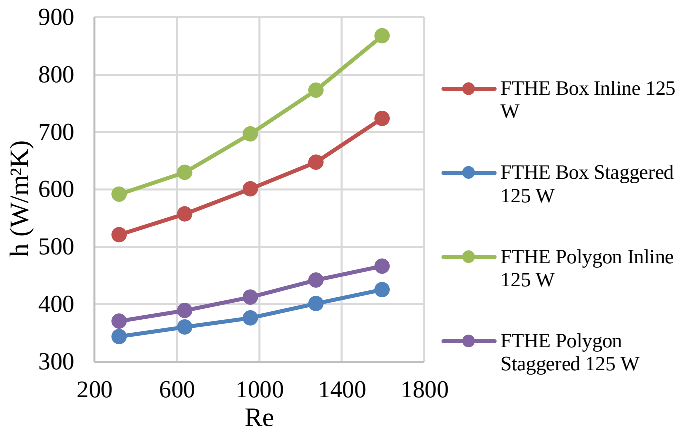


Fig. 7. Graph of the relationship between Reynolds number and heat transfer coefficient parameter in a fin-and-tube heat exchanger with box and polygon tube geometries, arranged inline and staggered, at various air flow velocities with Q = 125 W.

Fig. 8 shows that the temperature of the fin-and-tube heat exchanger is nearly uniform across the entire surface. The constant distance between tubes, 4.4 mm, allows for higher flow velocity transitions, enabling the free air to periodically and repeatedly initiate flow around the tubes. The nearly uniform temperature distribution of the fin-and-tube heat exchanger can serve as validation for determining the most optimal design for the heat exchanger, a validation typically marked by an increase in the Nusselt number. Fig. 9 shows the graph of the relationship between Reynolds number and Nusselt number in the fin-and-tube heat exchanger with box and polygon tube geometries at Q = 125 W.

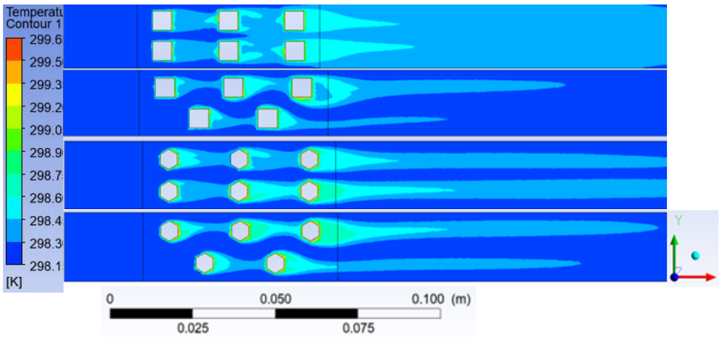


Fig. 8. Temperature contour of the fin-and-tube heat exchanger with box and polygon tube geometries, arranged inline and staggered, at an air flow velocity of 2.5 m/s with Q = 125 W.

Fig. 9 shows the relationship between the Reynolds number (Re) and Nusselt number (Nu) in a fin-and-tube heat exchanger with various fin geometries (box and polygon), as well as tube arrangements (inline and staggered), at a constant heat power of 125 W. In general, it can be observed that an increase in the Reynolds number, indicating a higher air flow velocity, leads to a consistent increase in the Nusselt number (Nu) for all tested configurations. This indicates that heat transfer becomes more efficient with the increase in airflow velocity.

When comparing fin geometries, polygon fins show higher Nu values than box fins at the same Reynolds number, both in inline and staggered arrangements, indicating that polygon fins are more efficient in heat transfer. In terms of tube arrangement, the inline configuration with polygon fins results in higher Nu values compared to box fins, signifying that the inline arrangement is more effective in enhancing heat transfer. On the other hand, in the staggered arrangement, although polygon fins perform slightly better in heat transfer compared to box fins, the difference is not as significant as that observed in the inline arrangement.

From these results, it can be concluded that polygon fin geometry is superior in terms of heat transfer efficiency compared to box fins, and the inline arrangement provides better heat transfer performance than the staggered arrangement, especially with polygon fin geometry. The Colburn factor (j) is known as a modified Reynolds analogy widely used to correlate heat, momentum, and mass transfer. The basic mechanism and mathematics of heat, mass, and momentum transfer are essentially the same and have been developed to directly link heat transfer coefficients, mass transfer coefficients, and friction factors. The Chilton and Colburn factor analogy, j, has been shown to be the most accurate (A. A. Buhiyan et al., 2016).

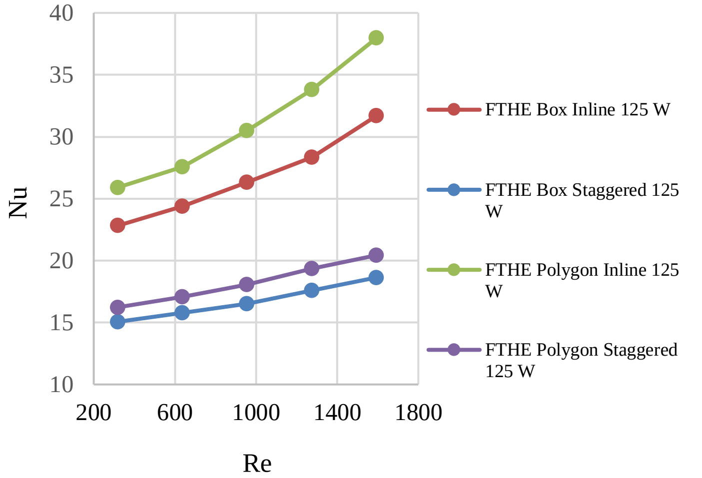


Fig. 9. Graph of the relationship between Reynolds number and Nusselt number in a fin-and-tube heat exchanger with box and polygon tube geometries, with inline and staggered tube arrangements at various air flow velocities and Q = 125 W.

Fig. 10 shows the graph of the relationship between the Reynolds number and the Colburn factor (j) in a fin-and-tube heat exchanger with box and polygon tube geometries, with inline and staggered tube arrangements, at various airflow velocities with Q = 125 W.

In light of these results, to optimize heat transfer efficiency, it is recommended to use polygon fins with an inline arrangement. However, it is important to consider other factors such as pressure drop and manufacturing complexity when selecting the appropriate geometry and configuration. For a more comprehensive understanding, future studies could explore the effects of varying heat input (Q) and other parameters on the overall heat transfer performance.

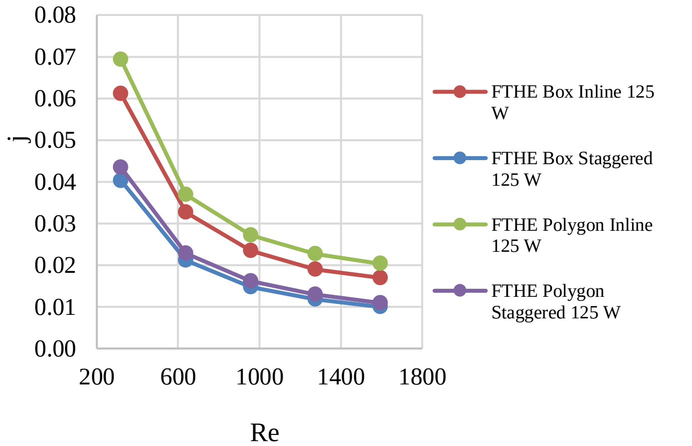


Fig. 10. Graph of the relationship between Reynolds number and Colburn factor (j) in a fin-and-tube heat exchanger with box and polygon tube geometries, and inline and staggered tube arrangements at various airflow velocities with Q = 125 W.

The effect of flow differences between box and polygon tubes with inline and staggered arrangements is significant. There is a clear distinction between box and polygon tubes with inline and staggered arrangements in terms of the recirculation zone, which acts as a differentiating factor. The recirculation zone area for polygon tubes with an inline arrangement is larger than that of the box tube configuration. On the other hand, in the staggered tube arrangement, the recirculation zone is smaller than in the inline arrangement, as the flow is only obstructed on one side of the domain. Polygon tubes with an inline arrangement show better heat transfer performance, as evidenced by the higher Colburn factor (j) value of 0.07.

3.2 Effect of Box and Polygon Tube Shapes with Inline and Staggered Arrangements on Hydraulic Aspects

The variation in shape, arrangement, and fluid flow velocity contributes to an increased ratio between the inertia force of tangential velocity and the viscous force. Fig. 11 illustrates the vortex intensity on the flow's cross-sectional plane at the same location as the velocity vectors. It was found that the box tube geometry, in both inline and staggered arrangements, exhibits higher vortex intensity. In the fin-and-tube heat exchanger with box tubes, the wake region is larger, accompanied by a higher countercurrent circulation. However, this vortex intensity diminishes downstream due to viscous dissipation. Typically, higher wake and vortex intensity in a heat exchanger system would lead to more uniform cooling. Interestingly, in this study, despite the polygon tube exhibiting smaller wake and vortex intensities, several key parameters related to heat transfer performed better compared to the box tube.

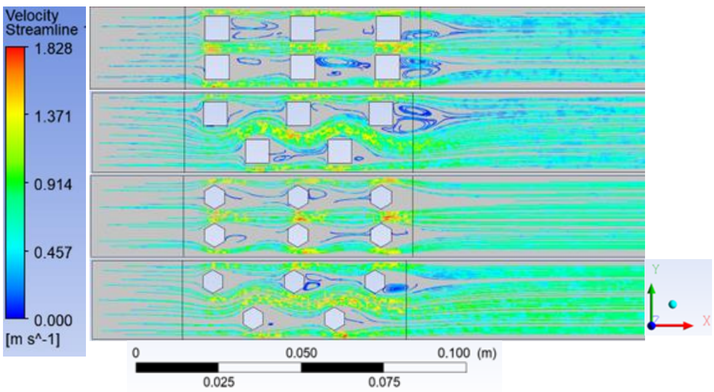


Fig. 11. Velocity streamline contour of fluid flow in fin and tube heat exchanger with tube box and polygon geometry arranged inline and staggered at flow velocity of 2.5 m/s and Q=125 W.

The growing fluid flow, which increases with the fluid velocity, plays a crucial role in the performance of the fin-and-tube heat exchanger. The periodic flow development in the test section of the fin-and-tube heat exchanger has an average velocity of 1.371 m/s. The velocity information of the fluid flow passing through the upstream domain, test section, and downstream creates a stronger secondary flow, which can enhance heat transfer but also increase the friction factor (f). The impact of the friction factor (f) leads to uneven hydrodynamic flow, affecting the cooling performance of the fin-and-tube heat exchanger and potentially reducing heat transfer efficiency. Fig. 12 illustrates the relationship between the Reynolds number and the friction factor (f) in the fin-and-tube heat exchanger with box and polygon tube geometries, arranged in both inline and staggered configurations, at an airflow velocity of 2.5 m/s and W = 125 W.

The graph in Fig. 12 shows the relationship between Reynolds number (Re) and friction factor (f) for four configurations of the Fin-and-Tube Heat Exchanger (FTHE) at a flow velocity of 2.5 m/s and Q = 125 W. The tested configurations include FTHE box inline 125 W, FTHE box staggered 125 W, FTHE polygon inline 125 W, and FTHE polygon staggered 125 W. All configurations exhibit a decreasing trend in friction factor as Reynolds number increases. At low Re, the friction factor for all configurations is nearly the same, but as Re increases, clear differences emerge between the configurations. FTHE Box Inline shows the highest friction factor, followed by FTHE box staggered, while FTHE polygon inline and FTHE polygon staggered exhibit lower friction factors. FTHE polygon staggered shows the lowest friction factor among all configurations, indicating the best performance in terms of flow resistance.

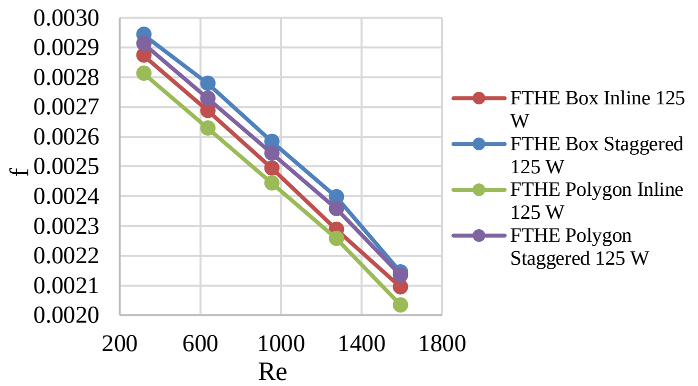


Fig. 12. Graph of the relationship between Reynolds number and friction factor (f) in a fin-and-tube heat exchanger with box and polygon tube geometries, arranged in both inline and staggered configurations, at a flow velocity of 2.5 m/s and Q = 125 W.

In terms of configuration comparison, the box configuration has a higher friction factor than the polygon due to the box geometry creating greater flow resistance. The staggered arrangement in both box and polygon configurations results in lower friction factors compared to the inline arrangement, allowing for smoother fluid flow and reducing turbulence. The combination of polygon geometry and staggered arrangement in FTHE polygon staggered provides the best result with the lowest friction factor, indicating more efficient fluid flow and minimal flow resistance.

In conclusion, FTHE polygon staggered demonstrates the best performance in terms of friction factor, meaning this configuration has lower flow resistance and higher efficiency in heat transfer. The staggered arrangement and polygon geometry prove to be key factors in minimizing the friction factor, which is crucial for enhancing heat transfer efficiency. Therefore, further research is needed to optimize the combination of geometry and arrangement in FTHE to achieve lower friction factors and higher heat transfer efficiency.

Fig. 13 illustrates the pressure distribution contours in a fin-and-tube heat exchanger with two different tube geometries: box and polygon, arranged in both inline and staggered configurations. The analysis was carried out at an air flow velocity of 2.5 m/s and a heat load of Q = 125 W. There are notable differences in the pressure distribution between the two geometry configurations. In the inline arrangement, the airflow is relatively steady, showing a more uniform pressure increase along the tube, with only slight pressure increases near the tube ends. In contrast, the staggered arrangement results in more turbulent airflow, with significant pressure fluctuations at specific points. This is due to the irregular flow pattern, which improves heat transfer but also increases pressure loss. The polygon geometry leads to a more intricate pressure distribution compared to the box tube, with a wider range of pressure variation along the tube surface. Overall, these pressure distribution results emphasize the importance of choosing the appropriate tube geometry and arrangement to optimize heat transfer efficiency without causing excessive pressure load on the system.

Fig. 14 illustrates the relationship between Reynolds number (Re) and pressure drop (ΔP) in a fin-and-tube heat exchanger with two different tube geometries, box and polygon, arranged in both inline and staggered configurations. The tests were conducted at a flow velocity of 2.5 m/s and a heat load of 125 W. In general, the pressure drop (ΔP) tends to rise as the Reynolds number increases. This suggests that as the fluid flow speed increases, the friction also increases, leading to a higher pressure drop.

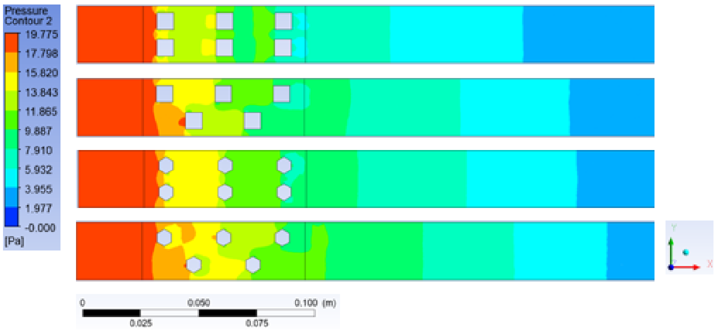


Fig. 13. Contour of pressure distribution in a fin-and-tube heat exchanger with box and polygon tube geometries, arranged in inline and staggered configurations, at a flow velocity of 2.5 m/s and Q = 125 W.

At lower Reynolds numbers, there is almost no noticeable difference in pressure drop across the four configurations. However, as the Reynolds number increases, a clear difference between the inline and staggered configurations becomes evident. The inline configuration typically results in a lower pressure drop. This is because the fluid flows more smoothly in the inline setup, reducing friction and, in turn, lowering the pressure drop. On the other hand, the staggered configuration creates more turbulence in the flow, which increases friction and leads to a higher pressure drop.

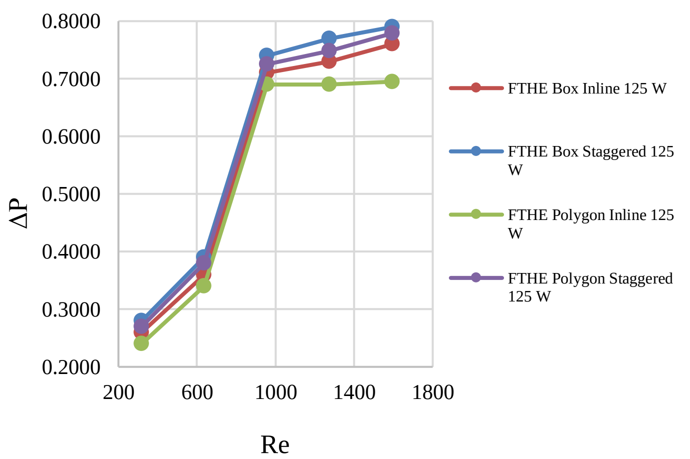


Fig. 14. Graph showing the relationship between Reynolds number and pressure drop in a fin-and-tube heat exchanger with box and polygon tube geometries arranged in inline and staggered configurations, at a flow velocity of 2.5 m/s and Q = 125 W.

Moreover, the box tube geometry results in a higher pressure drop compared to the polygon geometry. This can be attributed to the more intricate design of the box tube, which has a larger surface area, leading to increased friction and, consequently, a higher pressure drop. The staggered configuration with box tube geometry exhibits the highest pressure drop, while the inline configuration with polygon tube geometry generates the lowest pressure drop among all the configurations tested.

From the analysis in Fig. 14, it can be concluded that the selection of geometry and the arrangement of fins have a significant impact on the pressure drop in a fin-and-tube heat exchanger. The inline configuration with polygon geometry proves to be more efficient in terms of fluid flow, as it results in a lower pressure drop. However, the Inline configuration has a smaller surface area compared to the staggered configuration, which may limit its heat transfer performance. Therefore, while the Inline configuration is more effective at reducing pressure drop, the staggered configuration with polygon geometry may offer the best compromise between efficient fluid flow and optimal heat transfer performance.

4 Conclusion

The numerical study found that the fin and tube heat exchanger with variations in tube geometry (box and polygon) exhibited the best thermal and hydraulic performance when the polygon tube geometry was used, with a low temperature of approximately 23.41°C. This temperature value confirms a 5% increase in the heat transfer coefficient and a 3% increase in the Nusselt number.

Furthermore, the numerical study found that, in terms of both thermal and hydraulic performance, the fin and tube heat exchanger with variations in tube geometry (box and polygon) arranged in both inline and staggered configurations showed the best overall performance in the inline configuration, especially with the polygon tube geometry. This configuration had a low temperature of approximately 26.38°C, confirming a 4% increase in the heat transfer coefficient and a 2.5% increase in the Nusselt number.

Conflict of Interest Statement

The authors declare that they are unaware of any financial or personal relationships that could be perceived as influencing the work presented.

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