Heat transfer intensification with field synergy principle in a fin-and-tube heat exchanger through convex strip installation

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Abstract
Improvement of heat transfer using surface protrusion (convex strip) has been effective recently. Surface protrusion is able to improve flow mixing which increases the rate of heat transfer. Therefore, this study aims to improve the heat transfer in a fin-and-tube heat exchanger by fitting convex strips around the tubes. Three-dimensional modeling was carried out by placing four and eight convex strips around the staggered tubes at a constant temperature of 106°C. The turbulent k–ε model was applied at a Reynolds number range of 3438–15,926. The results of the study indicate that tubes with eight convex strips demonstrated a heat transfer improvement of 40.46%, compared to that with four convex strips. In this case, the TEF is 6.27% higher than the four convex strips. In addition, the synergy angle in the eight convex strips configuration was 0.13% lower than that of the four convex strips configuration. Meanwhile, the flow resistance in the tubes with eight convex strips configurations was 30.96% higher than that of the four convex strips.

KEYWORDS
convection heat transfer coefficient, convex strip, field synergy principle, friction factor, vortex intensity

JEL CLASSIFICATION
Mechanical engineering

1 | INTRODUCTION

Fin-and-tube type heat exchangers have been widely used in various manufacturing industries. In most cases, gas is used as the medium for heat exchange in fin-and-tube heat exchangers; however, the high thermal resistance on the gas side causes a low heat transfer rate. Therefore, the heat transfer rate on the gas side should be improved to increase the efficiency of this type of heat exchanger.1–3

Studies have been carried out experimentally and numerically to increase the gas side heat transfer rate in fin-and-tube heat exchangers. One of the methods to increase the heat transfer rate is by manipulating the surface geometry of the fins...
of the heat exchanger to expand their surface. Wavy fins, slotted fins, louvered fins, and vortex generators (VGs) on fins are commonly used fin geometries to increase the heat transfer rate. Mohanta et al.\textsuperscript{4} performed a numerical simulation by changing wavy fins to a hybrid slit–wavy fin in a staggered tube configuration at Reynolds numbers of 350–6500. Their simulation results demonstrated the increase of the heat transfer on the gas side from 20% to 39% compared to the fin without geometric manipulation. Yu et al.\textsuperscript{5} analyzed the heat transfer and flow resistance of two wavy fins at low Reynolds numbers. The increase in the Nusselt number was found to be proportional to the non-dimensional increase in the amplitude of the waviness and Peclet number. Sadeghianjahromi et al.\textsuperscript{6} combined a wavy fin with a delta winglet VG and slits on the fin surface. Fourteen variations were simulated with different waffle height ($h$), slit width ($W$), and slit length ($L$) values. The combination of a wavy fin with $h = 1.4$ mm, and two pairs of delta winglet VGs that provide a gap with $L = 12$ mm and $W = 1$ mm reduced the thermal resistance on the gas side by up to 16%. Zhang et al.\textsuperscript{7} studied the effect of modifying a wavy fin to a humped wavy fin on the heat transfer rate. The humped radius ($R$) was varied from 0.3, 0.5, and 0.7 mm, whereby the lowest average field synergy angle was observed at $R = 0.5$ mm, Guo et al.\textsuperscript{8} indicating the best heat transfer.

Li et al.\textsuperscript{9} experimentally and numerically compared the fluid flow structure and heat transfer characteristics of a flow-through butterfly type, x-type, and arc type slotted fins. Their study was carried out using seven flow velocity variations in the Reynolds number range of 558–2235. The butterfly type slotted fins increased the heat transfer by 20%–24% and exhibited a lower average field synergy angle than that of the other slotted fins. Kong et al.\textsuperscript{10} compared the effects of two types of slotted fins (continuous and alternating slots) on the heat transfer through simulations using tubes with staggered and in-line configurations. The highest heat transfer coefficient was observed using continuous slot fins with a staggered configuration. Li et al.\textsuperscript{11} investigated the effect of combining a slotted fin with a longitudinal VG on increasing heat transfer efficiency. The results of their numerical analysis noted the increase of the factor $j/(f^{1/3})$ by 15.8% compared to the plain fin. Meanwhile, Wang et al.\textsuperscript{12} investigated the effect of the slot parameters of the fins on the heat transfer performance. The slot lengths and widths were varied at 1.2–2.1 and 0.1–1.0 mm, respectively. They found that the optimal heat transfer was observed using a slot length and width of 1.38 and 0.27 mm, respectively. Deng and Qian\textsuperscript{13} noted the occurrence of thermal contact resistance, which had significant negative impacts on the heat transfer performance and minimal impact on reducing the pressure drop, using arc-slotted fins.

The numerical analysis of Huisseune et al.\textsuperscript{14} tested the combined use of a delta winglet VG with a louvered fin in the Reynolds number range of 140–1220, which resulted in an increase in the heat transfer by up to 16% at a Reynolds number of 660. Sadeghianjahromi et al.\textsuperscript{15} performed a numerical simulation to study the heat transfer characteristics and flow resistance of louvered fins, whereby the optimal louver angle of 21° was obtained to maximize the performance evaluation criteria. The effects of the fin pitch and longitudinal tube pitch on the Colburn $j$ factor and friction factor were negligible. In addition, Wan et al.\textsuperscript{16} investigated the thermal–hydraulic performance of louvered fins in a low-pressure environment. Numerical simulations were performed with variations in the pressure, fin pitch ($F_P$), louver angle ($\theta$), and fin length ($L_D$). The best performance was obtained at a pressure of 0 kPa, $F_P = 3$ mm, $\theta = 25^\circ$, and $L_D = 50$ mm. Meanwhile, Yadav et al.\textsuperscript{17} performed a size analysis of louvered fins using a genetic algorithm to obtain the optimal geometric parameters and their effect on increasing the heat transfer rate. The optimum results of an increase in the heat transfer rate by 22.56% were obtained using the following parameters: tube major axis ($T_{ma}$) = 1.706 mm, minor tube axis ($T_{ma}$) = 19.027 mm, tube length ($T_1$) = 285.997 mm, tube thickness ($T_i$) = 0.1945 mm, louver angle ($\theta$) = 28.17°, louver pitch ($L_P$) = 0.72 mm, fin pitch ($F_P$) = 1.292 mm, fin length ($F_i$) = 7.333 mm, louver length ($L_i$) = 5.868 mm, fin thickness ($\delta_i$) = 0.1725 mm, louver height ($L_h$) = 0.579 mm, and number of fins ($N_i$) = 221.

Song et al.\textsuperscript{18} observed the effect of curved delta winglet VGs on the efficiency of a fin-and-tube heat exchanger with varying dimensions of the VG, tube pitch, and fin pitch. Their observations indicated that the use of the VG was 18.79% greater than that of a plain fin. Syaiful et al.\textsuperscript{19} conducted experimental tests on a rectangular winglet (RWP) and concave rectangular winglet (CRWP) VG. The heat transfer coefficients of the RWP and CRWP VGs increased by 205% and 109%, respectively, compared to those of the plain fin. In another study, Syaiful et al.\textsuperscript{20} numerically analyzed the use of RWP and CRWP VGs on a rectangular channel, revealing the 102% increase in the heat transfer coefficient using CRWP, which was higher than that using RWP, at a Reynolds number of 364. Välikangas et al.\textsuperscript{21} observed the effect of the combination of herringbone fin and delta winglet VG on improving heat transfer. The delta winglet VG on the herringbone fin increased the heat transfer performance of the heat exchanger by up to 5.23%, compared to the heat exchanger without a delta winglet VG. This was attributed to the formation of longitudinal vortices in the downstream region of the VG. Meanwhile, Gupta et al.\textsuperscript{22} used a punched rectangular winglet VG to improve heat transfer with a lower pressure drop than that of a non-punched RWP.
Based on these previous studies, the geometric manipulation of fins is an effective method in improving the thermal performance of heat exchangers. Li et al.\textsuperscript{23} placed four convex strips around the tube of a fin-and-tube heat exchanger and noted an increase in the Nusselt number by up to 25\% compared to plain fins. In addition, the convex-striped fins increased the flow resistance by up to 16\% compared to plain fins. However, Li et al.\textsuperscript{23} did not consider the effect of the number of convex strips on the increase in heat transfer. Therefore, this numerical study aims to investigate the intensification of heat transfer in a fin-and-tube heat exchanger using different numbers of convex strips. Moreover, the flow phenomena in the gab fin of the fin-and-tube heat exchanger were observed in detail.

In this study, novelty is emphasized on the numerical study and addition of the number of convex strips and their effect on the thermal and hydraulic performance of the fin and tube heat exchanger. This makes it possible to determine the optimum number of convex strips in fin and tube in future studies.

2 | PHYSICAL MODEL

The numerical studies were performed by simulating a three-dimensional fluid flow in a fin-and-tube heat exchanger. The plain fin was compared with the fins with four and eight convex strips around the tube, respectively. The tubes were arranged in a staggered configuration with 12 rows and 6 columns. The dimensional variations on the configuration with four convex strips were based on the study of Li et al.\textsuperscript{23}; however, the current study increased the number of convex strips to eight to compare the intensification of heat transfer with the number of convex strips, as shown in Figures 1 and 2. The dimensions of the convex-striped fins are listed in Table 1.

![FIGURE 1](A) Top view of the geometry of the four convex-striped fin. (B) A–A cut of the geometry of the four convex-striped fin\textsuperscript{23}
The numerical studies were conducted under the assumption of an incompressible steady state fluid flow. The variations in the airflow velocity were based on the study by Li et al.\textsuperscript{23} The rectangular channel has a length, width, and height of 536.5, 42, and 2.3 mm, respectively. The flow direction, transverse direction, and distance between the fins are expressed as the $X$, $Y$, and $Z$ axes, respectively, as shown in Figure 3.

The dotted lines in Figure 3A represent the domain selected due to the periodic symmetry of the fin geometry. Figure 3B shows a three-dimensional view of the domain consisting of the upstream extended region, downstream extended region, and fin region. The upstream extended region was determined to obtain a fully developed flow on the inlet side, whereas the downstream extended region was determined to overcome the reverse flow.
3 | MATHEMATICAL MODEL

3.1 | Governing equations

In this study, an incompressible steady state fluid with constant physical properties was assumed. The flow in the channel was assumed to be turbulent with the flow velocity varied in the Reynolds number range of 3438–15,926. The fin thickness and conduction heat transfer to the fin were included in the modeling using conjugate heat transfer to determine the temperature distribution of the fin surface.

The governing equations used in the modeling calculations are:

Continuity equation

$$\frac{\partial}{\partial x_i} (u_i) = 0$$  \hspace{1cm} (1)

Momentum equation

$$\frac{\partial}{\partial x_i} (\rho u_i u_k) = \frac{\partial}{\partial x_i} \left( \mu \frac{\partial u_k}{\partial x_i} \right) - \frac{\partial p}{\partial x_k}$$  \hspace{1cm} (2)
Energy equation

\[
\frac{\partial}{\partial x_i} (\rho u_i T) = \frac{\partial}{\partial x_i} \left( \Gamma \frac{\partial T}{\partial x_i} \right) \tag{3}
\]

where \( \rho \) and \( \mu \) are the dynamic density and viscosity of air, respectively, and \( \Gamma \) is the diffusion coefficient, which can be described as:

\[
\Gamma = \frac{\lambda}{C_p} \tag{4}
\]

where \( \lambda \) and \( C_p \) are the specific thermal and thermal conductivities of air, respectively.

The equations for \( k \) and \( \varepsilon \) are

\[
\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_j}(\rho u_j k) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k + G_b + \rho \varepsilon + Y_M \tag{5}
\]

\[
\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_j}(\rho u_j \varepsilon) = \frac{\partial}{\partial x_j} \left[ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + \rho C_1 S - \rho C_2 \frac{\varepsilon^2}{k + \sqrt{\varepsilon}} + C_1 \frac{\varepsilon}{k} C_3 G_b \tag{6}
\]

where

\[
C_1 = \max \left[ 0.43, \frac{H}{H + 5} \right], H = \frac{k}{\varepsilon}, S = \sqrt{2S_{ij}S_{ij}} \tag{7}
\]

The eddy viscosity \( \mu_t \) is

\[
\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \tag{8}
\]

The coefficient \( C_\mu \) is

\[
C_\mu = \frac{1}{A_0 + A_s \frac{kU^*}{\varepsilon}} \tag{9}
\]

where

\[
U^* = \sqrt{S_{ij}S_{ij} + \bar{\Omega}_{ij} \bar{\Omega}_{ij}} \tag{10}
\]

and

\[
\bar{\Omega}_{ij} = \Omega_{ij} - 2 \varepsilon_{ijk} \omega_k \tag{11}
\]

\[
\Omega_{ij} = \bar{\Omega}_{ij} - 2 \varepsilon_{ijk} \omega_k \tag{12}
\]

The constants \( A_0 \) and \( A_s \) are

\[
A_0 = 4.04, A_s = \sqrt{6 \cos \varphi} \tag{13}
\]

where

\[
\varphi = \frac{1}{3} \cos^{-1}(\sqrt{6W}), W = \frac{S_{ij}S_{jk}S_{ki}}{S^3}, \bar{S} = \sqrt{S_{ij}S_{ij}}, S_{ij} = \frac{1}{2} \left( \frac{\partial u_j}{\partial x_i} + \frac{\partial u_i}{\partial x_j} \right) \tag{14}
\]
The model constants are

\[ C_{1r} = 1.44, C_2 = 1.9, \sigma_k = 1.0, \sigma_\epsilon = 1.2 \]  \hspace{1cm} (15)

### 3.2 Boundary conditions

To solve the mathematical equations used in the present study, several boundary conditions were established, as follows:

#### 3.3 At the upstream extended region

- At the inlet

\[ u = u_{in} = const, \; v = w = 0, \; T = T_{in} = const. \]  \hspace{1cm} (16a)

- At the top and bottom boundaries

  Velocity condition: periodic condition \( U_{up} = U_{down} \) \hspace{1cm} (16b)

  Temperature condition: periodic condition \( T_{up} = T_{down} \) \hspace{1cm} (16c)

- At the side boundaries

  \[ \frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0, \; v = 0, \; \frac{\partial T}{\partial y} = 0 \]  \hspace{1cm} (16d)

#### 3.4 At the downstream extended region

- At the outlet

  \[ \frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial w}{\partial x} = \frac{\partial T}{\partial x} = 0 \]  \hspace{1cm} (17a)

- At the top and bottom boundaries

  Velocity condition: periodic condition \( U_{up} = U_{down} \) \hspace{1cm} (17b)

  Temperature condition: periodic condition \( T_{up} = T_{down} \) \hspace{1cm} (17c)

- At the side boundaries

  \[ \frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0, \; v = 0, \; \frac{\partial T}{\partial y} = 0 \]  \hspace{1cm} (17d)

#### 3.5 At the fin region

- At the top and bottom boundaries

  Velocity condition: periodic condition \( U_{up} = U_{down} \) \hspace{1cm} (18a)

  Temperature condition: periodic condition \( T_{up} = T_{down} \) \hspace{1cm} (18b)
• At the side boundaries

\[
\begin{align*}
\text{Fluid region} & : \frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0, \quad v = 0, \quad \frac{\partial T}{\partial y} = 0 \\
\text{Fin surface region} & : u = v = w = 0, \quad \frac{\partial T}{\partial y} = 0 \\
\text{Tube region} & : u = v = w = 0, \quad T = T_w = \text{const.}
\end{align*}
\]  

(18c)

(18d)

(18e)

4 | NUMERICAL METHOD

In the current work, a mesh type domain was selected to ensure the accuracy of the calculation results.\textsuperscript{24} A polyhedral mesh was used to model all domains (upstream extended region, downstream extended region, and fin region) to reduce the number of nodes and elements, as shown in Figure 4.

Equations (1)–(3) with the boundary conditions Equations (5)–(7) were solved using computational fluid dynamics. A numerical simulation was performed using the \( k-\varepsilon \) turbulent model with a standard wall function to determine the solution in the area around the fin. SIMPLE algorithm was used to solve the velocity and pressure coupling. The convergence criteria were established for the equations of continuity, momentum, and energy to meet the conditions of less than \( 10^{-5} \), \( 10^{-5} \), and \( 10^{-8} \), respectively. For accuracy, \( y^+ \) has been obtained 0.5.

4.1 | Parameter definitions

The parameters used in this study are as follows:

Reynolds number : \( Re = \frac{\rho u_m D_h}{\mu} \)  

(19)

Nusselt number : \( Nu = \frac{h D_h}{\lambda} \)  

(20)

where \( \mu, \rho, u_m, D_h, \) and \( \lambda \) are the dynamic viscosity, density, average fluid velocity, hydraulic diameter, and fluid thermal conductivity, respectively. The hydraulic diameter formula was based on the study of Abolmaali et al.,\textsuperscript{25} which is described in Equation (21).

\[
D_h = \frac{4A_c}{P}
\]

(21)

FIGURE 4 Polyhedral mesh used in the computational domain\textsuperscript{22}
where \( A_c \) and \( P \) are the area and circumference of the channel cross-section, respectively. The convection heat transfer coefficient \( h \) is obtained using Equation (22).

\[
h = \frac{q}{A \Delta T_{\text{LMTD}}} \tag{22}
\]

where \( q, A, \) and \( \Delta T_{\text{LMTD}} \) are the fluid convection heat transfer rate, heat transfer surface area, and log mean temperature difference, respectively. The log mean temperature difference is determined using Equation (23) based on Lei et al.\(^{26}\)

\[
\Delta T_{\text{LMTD}} = \frac{(\overline{T}_w - \overline{T}_{\text{in}}) - (\overline{T}_w - \overline{T}_{\text{out}})}{\ln \left[ \frac{(\overline{T}_w - \overline{T}_{\text{in}}) / (\overline{T}_w - \overline{T}_{\text{out}})}{1} \right]} \tag{23}
\]

where \( \overline{T}_w, \overline{T}_{\text{in}}, \overline{T}_{\text{out}} \) are the average temperature of the tube walls, at the inlet, and at the outlet, respectively. \( \overline{T}_{\text{in}} \) and \( \overline{T}_{\text{out}} \) were determined following the study of Luo et al.\(^{27}\) described by Equation (24).

\[
\overline{T} = \frac{1}{S} \int \int_{S} T_{\text{local}} dS \tag{24}
\]

where \( S \) is the area of the inlet and outlet.

The thermal enhancement factor (TEF) is the ratio of the improvement in heat transfer to the increase in pressure drop, which is described by Equation (25) based on the study of Maradiya et al.\(^{28}\)

\[
\text{TEF} = \left( \frac{Nu}{Nu_0} \right) \left( \frac{f}{f_0} \right)^{1/3} \tag{25}
\]

where \( f \) is the friction factor, which can be determined using Equation (26).

\[
f = \frac{2 \Delta P D_h}{\rho u_m^2 L} \tag{26}
\]

where \( L \) is the length of the rectangular channel and \( \Delta P \) is the pressure drop of the fluid flowing through the rectangular channel, which is formulated in Equation (27).

\[
\Delta P = \overline{P}_{\text{in}} - \overline{P}_{\text{out}} \tag{27}
\]

where \( \overline{P}_{\text{in}} \) is the average pressure at the inlet side and \( \overline{P}_{\text{out}} \) is the average pressure at the outlet side expressed by Equation (28).

\[
\overline{P} = \frac{1}{S} \int \int_{S} P_{\text{local}} dS \tag{28}
\]

The field synergy principle (FSP) is a method for determining the heat transfer capability of a system by determining the synergy angle between the velocity and temperature gradient vectors. A smaller synergy angle indicates an increase in the heat transfer rate.\(^{29}\) The synergy angle (\( \theta \)) can be determined using Equation (29).

\[
\theta = \cos^{-1} \left( \frac{\overrightarrow{U} \cdot \overrightarrow{VT}}{||U|| ||VT||} \right) \tag{29}
\]

where \( \overrightarrow{U} \) is the velocity vector and \( \overrightarrow{VT} \) is the temperature gradient.
### Results and Discussion

The numerical studies of the heat transfer intensification in the fin-and-tube heat exchanger with different numbers of convex strips around the tubes were conducted. The observed phenomena and their influencing parameters are discussed in detail.

#### 5.1 Validation

In the current work, validation was carried out by comparing the calculation results between the numerical simulations and experimental results conducted by Li et al.\(^23\) Figure 5 shows a comparison between the experimental results of Li et al.\(^23\) and the current simulation with four convex strips installed on a fin-and-tube heat exchanger at various Reynolds numbers. The simulation results indicate an error value of ≤10% for the Nusselt number and ≤20% for the pressure drop.

#### 5.2 Effect of the number of convex strips on the flow field and temperature

Different flow fields were noted for the fins fitted with four or eight convex strips, which affects the temperature distribution because of the different local heat transfer rates. To observe the change in the flow field due to the use of the convex
FIGURE 5  Validation of the results for the fin-and-tube heat exchanger with four convex strips: (A) Nusselt number (B) pressure drop

FIGURE 6  Flow structure analysis area on the fin-and-tube heat exchanger for the: (A) plain fin, and fins with (B) four, and (C) eight convex strips

strips, the flow field was evaluated in the flow direction at \( Y = 0.92 \) mm. The representative area analyzed is indicated by the area bounded by the dotted lines in Figure 6.

Figure 7 shows a comparison of the streamline velocities in the fin-and-tube heat exchanger with a plain fin, and fins with four and eight convex strips at Re = 3438. A secondary flow is formed in the wake region at a low velocity in the plain fin. In contrast, this secondary flow is noted observed with the installation of the convex strips. Meanwhile, the installation of the convex strips increases the fluid velocity in the gap between the tube and convex strips owing to the narrowing of the channel cross-sectional area, which increases the fluid velocity.\(^{31}\) The changes in the flow patterns due to the installation of four and eight convex strips can be analyzed in detail by expressing the flow field in the horizontal plane \( (Y = 0.92 \) mm\), as shown in Figure 8. For the plain fin, the lowest secondary flow velocity is noted in the wake region.\(^{32}\) This secondary flow is generated by the flow separation after passing through the tube owing to an adverse pressure gradient in the wake region, which cannot be overcome by the low velocity flow in the area near the tube wall. The flow stall forms a reverse flow.\(^{33}\) In contrast, increasing the number of convex strips can direct the main flow to enter the wake region and reduce the size of the wake region, which causes the mixing of the cold fluid in the mainstream and hot fluid in the wake region, thereby increasing the heat transfer rate.\(^{34}\)

The massive mixing between the cold fluid in the mainstream and hot fluid in the wake region is caused by the longitudinal vortex produced by the convex strip. The longitudinal vortexes generated by the four and eight convex strips are observed in the cross-section of the flow direction \( (YZ \) plane\) at \( X/L = 0.038, 0.073, 0.108, \) and 0.14, as shown in Figure 9. Figure 10 presents the flow field at the several locations \( (X/L) \) noted in Figure 9 to analyze the
FIGURE 7  Streamline velocity of flow in the fin-and-tube heat exchanger at Re = 3438 for the (A) plain fin, and fins with (B) four convex strips, and (C) eight convex strips

longitudinal vortex generated by the convex strip in detail. The counter-rotating vortex is not observed in the plain fin case. For the fin with eight convex strips, a larger and stronger counter-rotating vortex is observed than that in fin with the four convex strips. This is mainly attributed to an increase in the pressure difference between the high momentum flow in the mainstream and low momentum flow in the wake region owing to the increase in the number of convex strips.35

In addition to the longitudinal vortex radius, the effect of increasing the number of convex strips on the longitudinal vortex strength was also observed in this study. Figure 11 illustrates the contours of the vortex intensity at the positions (X/L. The weakest vortex intensity is observed in the plain fin case because the vortex generated in the plain fin is a secondary flow in the wake region. Meanwhile, for fin with eight convex strips, the vortex intensity is stronger than that of the four convex strips. This is attributed to the increased number of gaps between the convex strips, which accelerates the flow in the upstream tube, and between the tube and convex strips (see Figure 8), resulting in an increase in the vortex intensity. This can be explained by comparing the distribution of the local vortex intensity at the same position, as shown in Figure 11B.C.

A comparison of the local mean vortex intensity along the channel for all cases is presented in Figure 12. From the peak local mean vortex intensity is observed in the front region of the tube at X/L = 0.093 (dotted line in Figure 12). Subsequently, the vortex intensity decreases after flowing through the tube. In the plain fin, a reduction in the vortex intensity is observed in the wake region at X/L = 0.14 (dashed line in Figure 12). In contrast, for the fins with convex strips, the local mean vortex intensity in the wake region is higher than that of the plain fin owing to the induction of the longitudinal vortex when the flow passes through the gap between the tube and convex strip. Furthermore, the local mean vortex intensity of the fins with eight convex strips is 97.96% higher than that with four convex strips.

The increase in the intensity of the longitudinal vortex generated by the addition of convex strips affects the temperature distribution of the fin along the channel. The temperature distribution of the plain fin and fin with convex strips is observed in the flow direction (XZ plane) at Y = 0.92 mm from the inlet domain, as shown in Figure 13. For the plain fin, the wake region has a high fluid temperature, which is inhibited by the transfer of the recirculation flow in the wake region12 to the mainstream at a low temperature. This indicates a low rate of heat transfer in the region.
For the fins with convex strips, the installation of the convex strips narrowed the wake region because the convex strips can direct the cold fluid to the wake region around the tube, resulting in the improved mixing between the cold fluid in the mainstream and the fluid around the tube walls. The improved mixing between the cold and hot fluids increases the heat transfer rate, resulting in a more uniform temperature distribution. The fin with eight convex strips exhibits a more uniform temperature distribution than that of the fin with four convex strips. This is attributed to the stronger longitudinal vortex generated with the addition of convex strips (see Figure 10), resulting in an increase in the heat transfer rate.

The effect of increasing the number of convex strips on the temperature distribution was also observed in the cross-section along the flow direction (YZ plane) at $X/L = 0.038$ of the inlet domain, as shown in Figure 14. The addition of convex strips generates a counter-rotating vortex, which causes the mixing of the high temperature fluid around the tube wall and low temperature fluid in the mainstream. In contrast, a counter-rotating vortex is not observed for the plain fin. As shown in Figure 14B,C, the fin with eight convex strips has a more uniform temperature distribution than the fin with four convex strips.
FIGURE 9  Position of the cross-section for the velocity vector observation against the computational domain inlet of the (A) plain fin, and fins with (B) four, and (C) eight convex strips.
FIGURE 10  Velocity vector in the cross-section at Re = 3.438 for the (A) plain fin, and fins with (B) four, and (C) eight convex strips
Figure 11 Contours of the vortex intensity (Se_v) at Re = 3438 for the (A) plain fin, and fins with (B) four, and (C) eight convex strips.
FIGURE 12  Comparison of the mean local vortex intensity along the channel between the plain fin, and fins with four and eight convex strips. The dotted red lines enclosed the area of the peak local mean vortex intensity

FIGURE 13  Temperature distribution along the flow direction (XZ plane) at $Y = 0.92$ mm at $Re = 3438$ for the (A) plain fin, and fins with (B) four, and (C) eight convex strips

5.3  Comparison of the effect of the number of convex strips on the heat transfer rate

The contours of the convection heat transfer coefficient for all cases at $Re = 3438$ were observed in the cross-section of the flow direction (YZ plane) at $X/L = 0.038, 0.073, 0.108,$ and $0.14,$ as shown in Figure 15. The lowest convection heat transfer coefficient is observed in the plain fin due to weak fluid mixing. The convection heat transfer
The local convection heat transfer coefficient increases with the addition of the convex strips. A comparison of the local convection heat transfer coefficient along the channel for different fin geometries is shown in Figure 16. The local convection heat transfer coefficient increases along the channel. The highest local convection heat transfer is determined in the fins with eight convex strips, followed by the fins with four convex strips, whereas the plain fin has the lowest local convection heat transfer coefficient.

A comparison of the convection heat transfer coefficient in the fins with four and eight convex strips in the Reynolds number range of 3438–15,926 is shown in Figure 17. The convection heat transfer coefficient increases with the increase in the Reynolds number because of the increased formation of longitudinal vortices at high velocity flows. The significant difference in the convection heat transfer coefficient is also influenced by the different fin geometries, as indicated by the higher heat transfer rate for the fin with eight convex strips compared to that of the plain fin and fin with four convex strips at the same Reynolds number. In addition, the increase in the number of convex strips can direct the mainstream flow to enter the wake region, resulting in the narrowing of the wake region. In addition, the increase in the number of convex strips generates a stronger longitudinal vortex, causing the mixing of the cold fluid in the mainstream flow with the hot fluid in the wake region, resulting in an increase in the heat transfer rate. Compared to the plain fin, the convection heat transfer coefficient increased by 44.78% for the fin with eight convex strips and 8.29% for the fin with four convex strips.

### 5.4 Effect of the convex strips on the pressure distribution and flow resistance

The convex strips affect the pressure distribution and flow resistance of the fin. The pressure distribution of the plain fin is lower than that of the convex-striped fins. This is mainly ascribed inhibited flow with the absence of geometric modifications on the fin surface. In addition, the fin with eight convex strips exhibits a higher pressure distribution than that...
of the four convex strips because of the increase in the drag coefficient with the number of convex strips. Particularly, this is ascribed to the narrowing of the channel cross-sectional area, which increases the vortex intensity, thereby increasing the flow resistance. This increase in flow resistance indicates an increase in the pressure drop as the number of convex stripes was increased, as shown in Figure 18. Compared to the plain film, the pressure drop increased by 249.62% and 102.86% for the fins with eight and four convex strips, respectively.
Effect of the convex strips on the TEF

The installation of the convex strip on the fin-and-tube heat exchanger increased the convection heat transfer coefficient. However, this improved heat transfer also increased the pressure drop. Therefore, the effect of the convex strips on the hydrothermal performance was analyzed using the TEF. TEF is the ratio of the improvement in the heat transfer to the increase in the pressure drop.\textsuperscript{26,38,39} A comparison of the TEF for the fin installed with four and eight convex strips is shown in Figure 19. The mean TEF for the fins with eight convex strips is 6.27\% higher than that with four convex strips. The TEF values of both fins decrease in the Reynolds number range of 4220 to 6397 because the increase in the Nusselt
number for both cases is lower than the increase of the pressure drop. However, at a Reynolds number of 15,926, the TEF of four convex-striped fin is higher than that of the eight convex-striped fin because of the smaller increase in the heat transfer rate of the eight convex-striped fin is lower. Therefore, the thermal–hydraulic performance of the fin with eight convex strips is better than that with four convex strips.

### 5.6 Effect of the number of convex strips on the FSP

FSP is a method for determining the improvement of the heat transfer by determining the synergy angle (θ), which is between the velocity and temperature gradient vectors. Guo et al. revealed an increase in the heat transfer
indicated by a decrease in the intersection angle between the velocity and temperature gradient vectors. A comparison of the local synergy angles for the plain fin, and fins with four and eight convex strips at a Reynolds number of 15,926 is shown in Figure 20. The lowest local synergy angles for the plain fin, and fins with four and eight convex strips are 82.77°, 78.26°, and 74.56°, respectively, at $X/L = 0.27$ (yellow dotted line in Figure 20). A comparison of the average synergy angles in the Reynolds number range of 3438–15,926 is shown in Figure 21. The average synergy angles of the plain fin, and four and eight convex-striped fins are 87.46°, 82.37°, and 82.26°, respectively. Compared to the plain fin, the mean synergy angle decreased by 5.94% and 5.81% in the fins with eight and four convex strips, respectively.
6 | CONCLUSION

In the current study, numerical simulations were performed to determine the effect of the number of convex strips on the increase in heat transfer rate in a fin-and-tube heat exchanger. The conclusions obtained from the results of the current study are as follows:

1. The addition of convex strip portions on the fins of a fin-and-tube heat exchanger increased the convection heat transfer coefficient for all Reynolds numbers. Compared to the plain, the average increase in the convection heat transfer coefficient is 44.78% and 8.29% with the addition of eight and four convex strips, respectively. In addition, the increase in the number of convex strips from eight to four increased the mean flow resistance by 60.45% and 20.22%, respectively, from that of the plain fin.

2. The highest TEF was obtained in the fin with eight convex strips, which is 6.27% higher than that with four convex strips. This indicates that the thermal–hydraulic performance with the installation of eight convex strips was better than that of the four convex strips.

3. The increase of the number of convex strips reduced the synergy angle with the average synergy angles of 87.46°, 82.37°, and 82.26° for the plain fin, and four and eight convex-striped fins, respectively.

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NOMENCLATURE

| Symbol | Description |
|--------|-------------|
| A      | heat transfer total area (m$^2$) |
| $A_c$  | minimum flow area (m$^2$) |
| $c_p$  | specific heat (J/(kg·K)) |
| $D_c$  | outer tube diameter (m) |
| $D_h$  | hydraulic diameter (m) |
| $F_P$  | fin pitch (m) |
| $f$    | friction factor (−) |
| h      | convection coefficient (W/m$^2$·K) |
| H      | channel height (m) |
| L      | channel length (m) |
| Nu     | Nusselt number (−) |
| P      | pressure (Pa) |
| $\Delta P$ | pressure drop (Pa) |
| $P_c$  | perimeter of the channel (m) |
| $P_l$  | tube pitch longitudinal (m) |
| $P_t$  | tube pitch transverse (m) |
| q      | heat transfer rate (W) |
| Re     | Reynolds number (−) |
| S      | area on the inlet and outlet sides (m$^2$) |
| $S_l$  | width of slit (m) |
| $S_i$  | breadth of a slit in the direction of airflow (m) |
| Se     | vortex intensity (−) |
| T      | temperature (K) |
| $\Delta T$ | average bulk temperature (K) |
| −∇$T$ | temperature gradient (K) |
| $\Delta T_{LMTD}$ | log mean temperature difference (K) |
| U      | velocity secondary flow (m/s) |
| −$U$   | velocity vector (m/s) |
$u_{in}$  frontal velocity (m/s)
$u$  velocity in $x$-axis (m/s)
$v$  velocity in $y$-axis (m/s)
$w$  velocity in $z$-axis (m/s)
$W$  channel width (m)
$x$  position in $x$-direction (m)
$y$  position in $y$-direction (m)
$z$  position in $z$-direction (m)

**GREEK SYMBOLS**

$\alpha$  transversal angle of convex to $x$-axis (°)
$\beta$  longitudinal angle of convex to $x$-axis (°)
$\rho$  density (kg/(m$^3$))
$\mu$  dynamic viscosity (Pa s)
$\lambda$  thermal conductivity (W/(mK))
$\delta$  fin thickness (mm)
$\theta$  synergy angle (°)
$\Gamma$  diffusion coefficient

**SUBSCRIPTS**

down  bottom surface
$i, k$  unite vector component
in  inlet
$m$  average value
out  outlet
pl  plain
up  top surface
w  wall

**ABBREVIATIONS**

CRWP  concave rectangular winglet pair
FSP  field synergy principle
RWP  rectangular winglet pair
TEF  thermal enhancement factor
VG  vortex generator

**DATA AVAILABILITY STATEMENT**

Data sharing is not applicable to this article as no new data were created or analyzed in this study.

**PEER REVIEW**

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**CONFLICT OF INTEREST**

All authors declared that they have no conflict of interest relevant to this article.

**AUTHOR CONTRIBUTIONS**

SyafiuL is responsibility for the concept and design of the work. Taufan Anindhito Wicaksono contributed to modeling and processing the data. M. S. K. Tony S. U. evaluates and justifies the modeling results. Agus Suprihanto is responsible for the systematic arrangement of the writing of this article. Maria F. Soetanto provides input in analyzing the results of the modeling.
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