Numerical simulation on flow and heat transfer of fin-and-tube heat exchanger with longitudinal vortex generators

Li Li, Xiaoze Du, Yuwen Zhang, Lijun Yang, Yongping Yang

Abstract

Fin-and-tube heat exchangers with plain fins are widely used in direct air-cooled condenser system in the power plant, because of its relatively simple property compared to some other fins with variable cross-sectional area channel. In the present study, heat transfer performance and pressure drop for fin-and-tube heat exchanger with longitudinal vortex generators (LVGs) on the fin surface were numerically investigated. Rectangular and delta winglet pairs were punched/mounted on the fin surfaces to enhance the heat transfer of the air-side of the fin-and-tube heat exchangers. The results showed that the Nusselt numbers increased up to 20% for LVGs on plain fins comparing with plain fins channel without LVGs. The heat transfer enhancement by rectangular winglets was more significant than that of the delta winglets. The rectangular winglet with angle of attack of 25° showed the best overall performance than any other angles of attack in rectangular winglets configurations. Additionally, the delta winglet with angle of attack of 45° showed the best overall performance than the other angles of attack in delta winglets configurations.

1. Introduction

Since the high-performance air-cooled condenser has significant advantage on water saving, it is widely applied in power plants in the regions where water source is limited. To significantly decrease thermal resistance on the air side, the fin-and-tube exchanger with vapor flow inside the tube and air flow outside between fins is a prevalent kind of the air-cooled condenser. Optimized geometry of fin-and-tube bundles are often seen in industrial applications in order to effectively improve the thermal performance of air-cooled heat exchanger. Through several generations of development, the plain fins with ellipse flat base tube are the advanced application of fin-and-tube in power plant condenser field. There have been many experimental and numerical investigations [1–7] on the heat transfer characteristics of wavy passage. Since the plain fins are the simplest and easy to manufacture, they still pay important roles in industry. Plain fins with longitudinal vortex generators (LVG) punched/mounted out of fin surface as a passive heat transfer enhancement method are drawing increasing attentions from the researchers.

Jacobi and Shah [8] presented a review of progress of longitudinal vortices up to the mid-nineties. Lau [9] experimentally studied the turbulent flow in a channel with LVGs by using X-wire and a quadruple hot-wire probe. Gentry and Jacobi [10] experimentally investigated the heat and mass transfer on a flat plate with delta–wing vortex generators at low Reynolds numbers. The mechanism responsible for the enhancement was based on the interactions between vortex and boundary layer. Lau et al. [11] continued to study the momentum and heat transport in turbulent flow channel with mounted vortex generators. Wang et al. [12,13] performed a series of flow visualization experiments to investigate the flow pattern of vortex generators in fin-tube channel. Their results showed that the frictional penalty of the proposed vortex generators was about 25–55% higher than that of the plain fin geometry. The potential of winglet vortex generator arrays for air-side heat transfer enhancement was experimentally evaluated by Joardar and Jacobi [14] through the wind-tunnel test. The heat transfer coefficient and pressure drop versus Reynolds number were obtained in the study. Zhang et al. [15] compared the heat transfer performances between configurations with punched/mounted vortex generators by using the naphthalene sublimation.
method and the results showed that the difference was insignificant.

Besides the aforementioned experimental investigations, many numerical simulations were conducted to assess the enhancement of LVGs on heat transfer in different configurations. Deb et al. [16] analyzed heat transfer characteristics and flow structure in laminar and turbulent flows through a rectangular channel containing vortex generators by solving the full Navier–Stokes and energy equations. Numerical investigation of heat transfer in channel with LVGs was carried out by Zhu et al. [17] who showed that average heat transfer enhancement of 450% was obtained by combining the effect of rib-roughness and vortex generators. Conjugate heat transfer in finned oval tubes with staggered punched LVGs was calculated by Chen et al. [18,19]. Detailed velocity field and vortex formation were presented to demonstrate the mechanism of enhancement of heat transfer. Wu and Tao [20–22] carried out continuous simulation to investigate the laminar convection heat transfer in fin-and-tube with LVGs combined with field synergy principle. Tian et al. [23] simulated heat transfer and flow characteristics of wavy fin-and-tube with delta winglets with constant flow cross-sectional area. The numerical investigation of fin-and-oval-tube with LVGs was reported by Chu et al. [24]. He et al. [25] analyzed the heat transfer and pressure drop for fin-and-tube heat exchanger with continuous and discontinuous winglets and compared their results with the conventional winglet configuration. They also conducted another study [26] to show the significant enhancement on heat transfer of rectangular winglet pairs.

Most of the existing simulation works about the longitudinal vortex generators did not take the effects of thickness of the winglet into consideration. Furthermore, few reports focused the application of LVGs in configuration of fin-and-tube with ellipse flat tube in direct air-cooled condensers of power plant except Du et al. [27]. In the present study, rectangular and delta winglet pairs were punched/mounted on the fin surfaces to enhance the heat transfer of air-side in fin-and-tube. The results were compared with those of fin-and-tube with wavy fins [28] to show the advantage of LVGs on the overall performance. The research provides useful information on the optimization of the air-side structure in air-cooled heat exchanger.

2. Numerical approach

2.1. Physical model

The schematic diagram of typical fin-and-tube heat exchanger with plain fin and ellipse flat tube used in direct air-cooled condenser system is shown in Fig. 1(a), which is named as baseline model and there is no longitudinal vortex generator on the fin. The diagram of fin-and-tube with rectangular winglets punched out from the fin surfaces is shown in Fig. 1(b), in which the angle of attack \(\beta\) with respect to the incoming flow direction is equal to zero. Several other fin-and-tubes with LVGs punched/mounted on the fin surfaces with \(\beta\) greater than zero are investigated in this study. The coordinate system is presented in Fig. 1(b), where \(x\) is the streamwise direction, \(y\) is the span-wise direction and \(z\) is in the fin pitch direction. The geometric parameters of the typical fin-and-tube are shown in Table 1. The outside diameter of the tubes is 20 mm and the width of the fin is 200 mm. The fin pitch is 2.8 mm and the fin thickness \(h\) is 0.3 mm. Ten winglet pairs are punched/mounted on the fin surface and placed symmetrically on both sides of the ellipse flat tube under common flow down in an inline arrangement. For the rectangular winglet, the length of winglet is 2 mm and the height of the winglet is half of the length. For the delta winglet, the length is 2 mm. The shape is not rigidly triangle but a trapezoid considering the actual manufacture precision of punch. The front height of the winglet is 0.3 mm and the back height is 1 mm. The thickness of winglet is equal to that of the fin. The angle of attack \(\beta\) of the rectangular winglet can have eight different values (\(\beta = 0^\circ, 10^\circ, 15^\circ, 20^\circ, 25^\circ, 30^\circ, 45^\circ, 60^\circ\)). The angle of attack \(\beta\) of delta winglet can have seven different values (\(\beta = 10^\circ, 15^\circ, 20^\circ, 25^\circ, 30^\circ, 45^\circ, 60^\circ\)). Due to the periodic and symmetric characteristic of fin-and-tube, two pieces of fins in the \(z\)-coordinate and the half domain in the \(y\)-coordinate of the structure is selected as the computational domain, as shown in Fig. 2. The domain is extended
by 20 mm for the entrance section to ensure the inlet uniformity and is extended by 160 mm for the exit section to avoid the circulation effect.

2.2. Governing equations and boundary conditions

The flow in the computational domain is considered to be incompressible with constant thermophysical properties. It was also assumed that the flow is three-dimensional, turbulent and no viscous dissipation. The governing equations include continuity equation, momentum equation and energy equation.

Continuity equation:

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

Momentum equation:

$$\frac{\partial (\rho u_i u_j)}{\partial x_i} = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_i} \left[ \mu \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \rho u_i u_j \right]$$ \hspace{1cm} (2)

Energy equation:

$$\frac{\partial u_i T}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ (\Gamma + \Gamma_t) \frac{\partial T}{\partial x_i} \right]$$ \hspace{1cm} (3)

The Reynolds stress in Eq. (2) is:

$$-\rho u_i u_j = \mu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \rho k \delta_{ij}$$ \hspace{1cm} (4)

where the turbulent viscosity $\mu_t$ and the turbulent dynamic thermal diffusivity $\Gamma_t$ are given by $\mu_t = \rho \beta k$ and $\Gamma_t = \mu_t / \rho T$, respectively. The turbulent kinetic energy $k$ and the turbulent frequency $\omega$ are computed from $k-\omega$ model [29,30]:

$$\frac{\partial (\rho u_i k)}{\partial x_i} = \mu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \rho \beta^2 k \omega + \frac{\partial}{\partial x_i} \left[ \left( \mu + \mu_t \right) \frac{\partial k}{\partial x_i} \right]$$ \hspace{1cm} (5)

$$\frac{\partial (\rho u_i \omega)}{\partial x_i} = \frac{\omega}{k} \mu_t \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) \frac{\partial u_i}{\partial x_j} - \rho \beta \omega^2 + \frac{\partial}{\partial x_i} \left[ \left( \mu + \mu_t \right) \frac{\partial \omega}{\partial x_i} \right]$$ \hspace{1cm} (6)

where the coefficients are $\alpha = 5/9$, $\beta = 0.75$, $\beta^* = 0.09$, $\sigma_k^* = 2$, $\sigma_\omega = 2$.

The experimental verification results were discussed in Ref. [31], where the $k-\omega$ model was proved as the most reasonable turbulent model in this research through the comparison between different turbulent models.

The schematic diagram of the computational domain is shown in Fig. 3. The solid domain includes base tube, fins and LVGs, as shown in Fig. 3(a). The boundary conditions are shown in Figs. 2(a) and 2(b). For the inlet boundary condition, the velocity and temperature of the air are assumed to be uniform, $u = u_{in}$, $v = w = 0$, $T = T_{in}$. The periodic boundary conditions are set at the upper and lower boundaries of the domain $\phi_{upper} = \phi_{lower}$. The constant wall temperature boundary condition is set at the outer surface of the base tube. The symmetric boundary conditions are set at the sidewall of the fluid region, $v = 0$, $\partial u_i / \partial y = \partial w / \partial y = 0$, $\partial T / \partial y = 0$. The outlet boundary condition is set at the outlet surface, $\partial u_i / \partial x = 0$, $\partial w / \partial x = 0$, $\partial T / \partial x = 0$.

2.3. Solution method

The commercial software ANSYS CFX was employed to carry out the simulation. The governing equations with corresponding boundary conditions were solved by using a finite volume method. The high resolution scheme was adopted as the advection scheme, which could present more accurate result than that calculated by adopting upwind scheme. For solving the coupling between the fluid and solid, a control surface approach was used to perform the connection across the general grid interface attachment. The solution was believed to be converged once the maximum residual of all variables, such as pressure, velocity components and temperature were less than $10^{-5}$.

The Reynolds numbers are based on the average velocities and the hydraulic diameter at the average flow cross-sectional area. Heat transfer performances are evaluated at the average bulk

<table>
<thead>
<tr>
<th>Table 1</th>
<th>Dimensions of baseline fin-and-tube.</th>
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<tbody>
<tr>
<td>Items (coordinate)</td>
<td>$W_i (x)$</td>
</tr>
<tr>
<td>Unit (mm)</td>
<td>200</td>
</tr>
</tbody>
</table>
temperature of the fluid. The Reynolds numbers, Nusselt numbers and Colburn factor \( j \) are defined as follows:

\[
Re = \frac{\rho u_{av} D_{av}}{\mu}
\]

(7)

\[
Nu_{av} = \frac{h_{av} D_{av}}{\lambda}
\]

(8)

\[
j = \frac{Nu_{av}}{Re Pr^{1/3}}
\]

(9)

where \( u_{av} \) is the mean velocity of the minimum flow cross-section area and \( D_{av} \) is hydraulic diameter that is computed by the ratio of flowing area and wetted perimeter of the flow section \( D_{av} = 4 \left[ (2s_t - 2\delta_t)H + 2s_t \times 0.3 \right] / 4H + 2s_t + 2 \times \delta_t \), where 0.3 is the half of gap between two fin-and-tubes in y-direction, where \( h \) is the heat transfer coefficient that can be calculated with the following relation:

\[
h_{av} = \frac{Q}{\eta_0 \Delta T}
\]

(10)

where \( Q \) is the heat transfer rate expressed as:

\[
Q = c_p m (T_{out} - T_{in})
\]

(11)

and \( \eta_0 \) is the overall fin surface efficiency:

\[
\eta_0 = 1 - \frac{A_f}{A_0} \left( 1 - \eta_f \right)
\]

(12)
where \( \eta_f \) is the fin efficiency calculated with the following relation:

\[
\eta_f = \frac{T_{f_{av}} - T_{w}}{T_{a_{av}} - T_{w}}
\]

where \( T_{f_{av}} \) is the average temperature of fin surfaces, and \( T_{a_{av}} \) is the average temperature of the tube surface, to be calculated from the values of the simulated results. \( T_{av} \) is the average temperature of flow based on inlet and outlet, that is

\[
T_{av} = \frac{T_{in} + T_{out}}{2}
\]

\( A_f \) is the total area of fin surfaces, and \( A_0 \) is the total area of heat transfer, that is the sum of areas of fin surfaces and tube surface.

The logarithm temperature difference is defined as:

\[
\Delta T = \frac{(T_{w} - T_{in}) - (T_{w} - T_{out})}{\ln \frac{T_{w} - T_{in}}{T_{w} - T_{out}}}
\]

The pressure drop coefficient is defined as:

\[
C_f = \frac{\Delta P^*}{\frac{1}{2} \rho u^2 \Delta A}
\]

\[
f = \frac{\Delta P^* A}{\frac{1}{2} \rho u^2 A_0}
\]

where \( \Delta P^* \) is the difference of the total pressure between inlet and outlet; \( A \) is the minimum cross-section area of flow along the streamwise.

2.4. Grid independence

Accuracy of the CFD solution strongly depend on the grid system, which must be constructed to minimize the grid-induced errors and to solve the relevant flow physics. In order to check the amount of mesh schemes under fixed computer memory, single fin with symmetric arrangement was used. The multi-block method was used and the structured grids were generated. Meanwhile, the total equal block-divided strategy was adopted in comparing the heat transfer and pressure drop results of models with or without vertex generators. For example, the winglet blocks divided in the simulation models were not only in the configuration with winglet but also in the configuration without winglet. In Fig. 3(c), the winglet block lined out is rectangular vertex generator and it belongs to a solid domain. The hole block belongs to the fluid domain. In Fig. 3(d), the winglet block belongs to the fluid domain and the hole block belongs to the fin solid domain. Several mesh schemes have been conducted to examine the dependence of mesh size on the accuracy of the analysis results. The total number of mesh were 1,160,000, 1,950,000, 2,580,000, 2,810,000, and 4,970,000 respectively. The results are seen in Fig. 4. The third grid scheme was chosen as the final mesh for all the simulations in this work.
3. Results and discussion

3.1. Influences of the angle of attack

To study the performance dependence of the winglet on the angle of attack, several configurations with different angles of the attack were investigated. The angles of attack of rectangular winglet pairs were 0°, 10°, 15°, 20°, 25°, 30°, 45°, 60°, and those of delta winglet pairs were 10°, 15°, 20°, 25°, 30°, 45°, 60°. For convenience, the configurations were named as rec-0, rec-10, rec-15, rec-20, rec-25, rec-30, rec-45, rec-60, del-10, del-15, del-20, del-25, del-30, del-45, and del-60, respectively. Fig. 5 shows comparison of the performances of fin-and-tube with or without rectangular winglets. Considering the average velocity of actual wind flowing through fan in the power plant is approximately 3 m/s, the inlet velocities were chosen as 1–6 m/s, and the corresponding Reynolds number were 450–3000.

Fig. 5(a) shows the comparisons between the normalized Nusselt numbers of LVGs with different angles of attack (Nu0 refers to the Nu of baseline case). The baseline case was the fin-and-tube without LVG. It can be seen that the adding of LVGs can
effectively enhance the Nusselt number. In addition, the Nusselt numbers increase with increasing angle of attack and the best enhancement is up to 20%. The enhanced heat transfer may come from the facts that the breaking off of the boundary layer and the secondary flow that is produced by the longitudinal vortex generators; the secondary flow allows the fluid to impinge to the fin surface directly [32]. The larger the angle of attack is, the more enhanced heat transfer is. Under a given angle of attack, there are variable cross-sectional areas along the flowing channels around the winglet. Therefore, when the angle of attack increases, the heat transfer is enhanced. Fig. 5(b) shows the normalized flow resistance coefficient of configurations with different angles of attack (Cf₀ refers to the Cf of baseline case). It demonstrates that the flow resistance coefficient decreases with increasing Reynolds number, and the result for the case with the vortex generator exhibits stronger drag than that for the plain fin. The values increase with the enlargement of angles of attack. The flow separation is induced by the rectangular winglet. The form drag of the rectangular winglet causes the increase of drag.

It is essential to compare the overall performance and the performance evaluation factor (PEF). Fig. 5(c) compares the performance evaluation factor \( \frac{\left(j/f^2\right)}{\left(j_0/j_0^2\right)} \) (\( j_0/j_0^2 \) refers to the \( j/f^2 \) of baseline case) versus the Reynolds number. It is interesting to note that the PEF reaches a peak in the range of the Reynolds number in this study. When the Reynolds number is low, the extent of heat transfer enhancement is small, even negative in some configurations. With the increasing Reynolds number, the effect of the longitudinal vortex generators on PEF increases firstly and then decreases at high Reynolds number, which means that the heat transfer enhancement is insufficient to make up for the increase of drag coefficient. The probable reason may be related to the structure of the fin-and-tube. Two fins without vortex generators form a nearly rectangular channel. The initial pressure drop of plain fins configuration is small. Furthermore, the increase of pressure drop induced by the vortex generators is relatively large. When the Reynolds number is close to 450, the pressure drop induced by the vortex generators plays a primary role in the drag of the channel. The heat transfer is firstly enhanced with Reynolds number from

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**Fig. 8.** Local temperature distribution of fluid at different x locations of del-45 case (\( u_{in} = 3 \) m/s). (a) Temperature; (b) Total Pressure.
450 to 1000. Subsequently, the heat transfer decreases due to gradually weakening influences from the vortex generators. By comparing of the effects of angles of attack, it can be seen that the rectangular winglet with angle of attack of 25° shows the best overall performance.

Fig. 6 shows the performances of the fin-and-tube heat exchanger with delta winglet pairs punched out of the fin surfaces. The maximum heat transfer enhancement is near 15% after adding the delta winglet, which distinguishes from that rectangular winglet with angle of attack of 45° showing the best heat transfer (see Fig. 6(a)). At the same time, the pressure drop coefficient of delta winglet does not increase with the increase of angle of attack. In other words, the delta winglet with angle of attack of 45° is nearly same to that of 60° (see Fig. 6(b)). As for the performance evaluation factor versus the Reynolds number, it has the similar tendency with the rectangular winglet. There is an optimized angle of attack with the increasing of Reynolds number. It can be seen that the delta winglet with angle of attack of 45° shows the best overall performance.

Fig. 7 shows the local heat transfer coefficient along the streamwise (x-direction) from inlet to outlet at the inlet velocity of 3 m/s. The heat transfer coefficient gets the highest value at the inlet, and then decreases with the increasing of x. The local heat transfer gets enhanced due to the break of boundary layer when the flow meets the LVGs. The total tendency of the heat transfer coefficient presents ten peaks because of the ten pairs of LVGs. The peaks occur at the locations of the LVGs and the values of peaks gradually decrease as the fluid flows through the fins.

The distributions of temperature and total pressure of the fluid at different cross sections along the streamwise are shown in Fig. 8(a) and (b), respectively. Several typical locations (the beginning cross-sections and the ending cross-sections of vortex generators along the channel and several y-z sections at the intervals of 1 mm) were chosen. Here, x = 0.01 m is the location of the first pair of winglets (x = 0 is the location at the beginning of fins); x = 0.012 m is the location of the ending of the first pair of winglets; x = 0.0195 m is the location of the center distance of the first two pairs of winglets; and x = 0.031 m is the location of the second pair of winglets. It can be seen that the temperatures near the tube and fins are higher than those located far away the fins and tube. Due to the strong mixing of fluids, the higher fluid temperatures at the location of LVGs can be noticed. Because of the variation of cross-section area, the pressure near the LVGs is lower than that at other locations. The local total pressure between the two pairs of LVGs at x = 0.0195 m is higher than the local total pressure at the locations of LVGs, such as those at x = 0.011 m, x = 0.012 m, x = 0.031 m, etc.

3.2. Influences of shape

It was reported in our previous work that the wavy fin with the periodical variable cross-sectional area channel showed the great enhancement of heat transfer [28], but the highest pressure drop led to the lowest overall performance. The two approaches of heat transfer enhancements (the channel with periodical variable cross-sectional area and the channel with the LVG) are compared. It should be pointed out that the data processing in Ref. [28] did not considering the fin-efficiency and the velocity in calculation of the pressure drop coefficient, in which the velocity used is the inlet value but not the average section area as in this study. Consequently, there are discrepancies between the results in the present work and those reported in Ref. [28].

Fig. 9(a) shows the normalized Nusselt number versus the Reynolds number. It can be seen that the normalized Nusselt numbers for the cases with wavy fin are much greater than that of the plain fin configurations with LVGs. The flow cross-section area varies with the air flow along the channel downstream in the wavy fin configuration. For the fin-and-tube with wavy fins, the heat transfer enhancement effect takes place in the entire channel, since the variable cross-sectional areas causes flow oscillation. On the other hand, the LVG only impacts the region near the vortex
generators. Another conclusion that can be made is that for vortex generators with the same height-to-length ratio at the same position in the channel, the heat transfer enhancement by rectangular winglet is more significant than that of the delta winglet. The change of pressure in wavy fin configuration occurs everywhere in the channel, whereas it occurs merely in the local region for the case of plain fin with vortex generators configuration. This is the reason that the pressure drop coefficient in wavy fin is so larger than other configurations as shown in Fig. 9(b). At the same time, the rectangular winglet shows higher pressure drop than the delta winglet configuration. In Fig. 9(c), it is seen that the PEF of wavy fin is less than that of configuration with LVGs. The advantage of using of vortex generator can be verified in terms of pressure drop, although the heat transfer enhancement is relatively lower than the periodical variable cross-sectional channel such as the wavy fins.

3.3. Influences of punched holes

Vortex generators can be either mounted on the surface of fins, or punched out from the surface of the fins. The total heat transfer area in the fin surface with vortex generators mounted on is slightly larger than that of the fins with punched holes. In this study, configurations with del-45 winglets punched or mounted on the fin surface are simulated to compare the influences of punched hole. Fig. 10(a) shows the comparisons between the normalized Nusselt numbers for the cases with and without punched holes. The heat transfer enhancement induced by the punched holes can be noticed with the increasing of Reynolds number. Fig. 10(b) presents the normalized pressure drop coefficient versus Reynolds number. It demonstrates that air flowing through the punched holes leads to the higher pressure drop for configurations with punched hole than that without punched hole with the increasing of Reynolds number. The overall performances are relatively increased by the punched holes as shown in Fig. 10(c).

3.4. Flow patterns

Strong secondary swirling flow can be generated when fluid passes through the LVG. To gain a further insight into the generation of primary and secondary vortices, streamlines starting from different points of the leading edge and trailing edge of the vortex generators were plotted. Fig. 11(a) shows the streamlines from points of winglet leading and trailing edge for the first pair of mounted (without punched holes) del-45 LVG near Re = 1430.
There are four vortex generators in the series. In the delta vortex generators, the leading edge includes top edges and front edges, which are lined out in Fig. 11(a). Lines on the winglets were drawn to show the starting position of the streamlines from different edges. The streamlines starting from front edges were shown on the upper winglet near the base tube. Those starting from the top edges were drawn on the lower winglet near the base tube. On the winglets far away from the base tube figured out both streamlines from front edges and top edges. When air flows through the leading edges, it rotates heavily due to the pressure difference coming from the variable cross-sectional area with or without vortex generators. These streamlines braid together can form the core of the primary vortex. Through the streamlines starting from the winglet trailing edge, it can be seen that the rotation of the fluid becomes weaker at the trailing edge.

To get a better understanding of the longitudinal vortex which has the axes parallel to the main flow direction, the same typical locations introduced in Fig. 8 were chosen to show the structure of longitudinal vortex produced by the LVGs or punched holes. Fig. 11(b) includes the vector graphs of the several sections along the x-direction in mounted del-45 configuration. There are two main vortices in the cross-section at \( x = 0.012 \) m: the left one after the left winglet rotates in clockwise direction, and the right one after the right winglet rotates in counterclockwise direction. The cross-sectional area of flow between winglet and base tube wall decrease along the streamwise. At the same time, the cross-sectional area between two winglets increases from the beginning of the winglet to the ending of the winglet form a divergent channel. The gradually increasing channel cross-sectional areas leads to the fact that the static pressure in the region between the first pair of winglets rises slowly from the beginning to the ending cross-section, where the air pressure achieves maximum, then air flow into constant cross-section channel until the second pair of winglets. In divergent section, the momentum of the fluid layer near the surface is not high enough to overcome the increase of pressure, separation and reverse flow phenomena may occur to result in the secondary vortex and swirling of the fluid. Additionally, the winglets construct a convergent channel between winglet and upper fin and between winglet and tube wall, which accelerates air flow to enhance the heat transfer. From the locations at \( x = 0.012 \) m–0.0195 m, the strength of swirling become weaker along the flow direction until the second pair winglet, the second pair of vortex produced.

Fig. 12(a) shows the streamlines starting from the edges for punched del-45 LVG with punched holes at the same Reynolds number with that in Fig. 11. The streamlines starting from the front edges and top edges were shown on the upper and lower winglet nearby the base tube. The streamlines starting from the trailing edges were drawn on the upper winglet that is far away from the base tube. Both streamlines from the leading edges and trailing

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**Fig. 12.** Flow patterns at different locations of the punched del-45 case \((u_m = 3 \text{ m/s})\). (a) Streamlines start from winglets. (b) Flow patterns at different x locations.
edges were shown on the lower winglet far away the base tube. It can be seen that the punched holes seem to weaken the rotation of flow on the front edges and eject the flow on the top edges. The streamlines starting from edges of the punched holes corresponding with leading edges were shown on the upper winglet nearby the base tube. The streamlines starting from the edges corresponding with trailing edges and connection edge were shown on the lower winglet nearby the base tube. All streamlines from edges of punched hole were shown on the winglets far away the base tube. The fluid between fins channel flowing through the holes and twist together forms the core of the corner vortex. Fig. 12(b) shows the vector plots at the same cross-section of the configuration with punched del-45 winglets. It can be seen that, after the punched holes there are vortices that have opposite rotational directions to the main vortex produced by winglets. The secondary vortexes may be produced by the fluid flow through punched holes. The intensity of the vortex is larger than the main vortex, then decreases along the flow direction smaller than the main vortex.

Fig. 13(a) represents streamlines of punched rec-45 LVG at the same Reynolds number. The arrangement of starting position of streamlines is same as Fig. 12(a). The streamlines under the lower winglets come from the neighbor fin because of the periodic boundary condition on the upper and lower surfaces in the fluid domain. It can be seen that the extent of twist and rotates is heavier than that of the delta vortex generators. Fig. 13(b) is the vector plot at the same cross-section of the configuration with punched rec-45 winglets. The style of the vortex is similar to those of the del-45 winglets. Obviously, the location of main vortex is different from the del-45 winglets. The main vortex location (at x = 0.012 m and x = 0.013 m) is relatively higher than the vortex produced by the punched holes near the upper fins in the z-direction. As seen in Fig. 13(b), the leading edge of rectangular winglet are higher than the leading edge of delta winglet. The contribution of heat transfer enhancement is the longitudinal vortex along the channel. Whereas, the longitudinal vortex increase the pressure drop. The flow pattern shows detailed streamlines and longitudinal vortex in the channel produced by vortex generators and holes. Further understanding of heat transfer enhancement due to separated and swirling of fluid was elucidated.

4. Conclusions

The heat transfer performance and pressure drop of fin-and-tube heat exchanger with rectangular and delta shaped LVGs in react thickness punched/mounted on plain fin configurations were numerically studied. The main conclusions are drawn as follows:

![Flow patterns at different locations of the punched rec-45 case (u_in = 3 m/s). Flow patterns at different locations of the punched rec-45 case (u_in = 3 m/s). (a) Streamlines start from winglets. (b) Flow patterns at different x locations.](image-url)
(1) For Reynolds numbers between 450 and 3000 (corresponding to face inlet velocities between 1 and 6 m/s), the Nusselt numbers increase up to 20% for LVGs on plain fins comparing to plain fins channel without LVGs. Under the same angle of attack, the heat transfer enhancement by rectangular winglet is more significant than that of the delta winglet. The rectangular winglet with angle of attack of 20°–30° shows the best overall performance, while the delta winglet with angle of attack of 45° shows the best overall performance.

(2) Comparing heat transfer and drag between different shapes of fin channels, the result of wavy fin has significantly better performance than that of the plain fin configurations with LVGs. As a result, the overall performance of wavy fin is far less than the plain fin configuration.

(3) The existence of punched holes enhance heat transfer of rectangular vortex generator more significantly than that of delta one. In general, the heat transfer of configuration with punched holes is slightly better than that of configuration without punched holes.

(4) Several vortices can be produced not only by the leading edge and trailing edge of winglets but also by the holes. The strength of swirling and the intensity of the vortice after the prior pair winglets become weaker along the flow direction until the next pair winglets.

Acknowledgments

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References