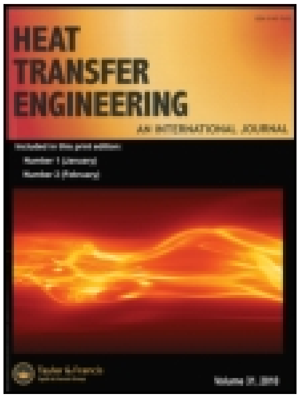

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Application of Combined Enhanced Techniques for Design of Highly Efficient Air Heat Transfer Surface

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In order to reduce the size and cost of heat exchangers, an air-side wavy fin-and-tube heat transfer surface with three-row tubes needs to be replaced by two-row tubes with some appropriate enhancing techniques. The major purpose of the present paper is to search for such new structure by numerical simulation. First, longitudinal vortex generators of Delta-winglet type are tried. The influence of number and of arrangement of the winglets on the performance of the heat transfer surface is studied in detail. The numerical results show that the fin with two winglets aligned spanwise in the front and rear of each tube (Fin W6) has higher heat transfer capability than other enhanced structures with vortex generators, but it still unable to meet the heat transfer requirement. Then a combination design of the longitudinal vortex generator with slotted protruding parallel strips is proposed and different variations of their arrangement are tried. Finally we come to such a combination (C3), which is based on Fin W6 with additional eight protruding strips situated at five positions (grouped by 1, 2, 2, 2, and 1) along the flow direction. Fin C3 can satisfy the requirements for heat transfer rate of the original wavy fin of three-row tubes with a mild increase in pressure drop, and its volume and material reduce to about 67% of the original one.

INTRODUCTION

Plate fin-and-tube heat exchangers are widely employed in various engineering fields. Many factors including the total amount of heat transferred, pressure drop, performance efficiency, manufacturing, and operating cost may influence the final design and sizing of heat exchangers [1]. In some cases the overall cost is most important, while in other applications weight and size may be the most vital factors. With the emerging of a worldwide crisis of energy shortage, the energy-saving purpose of enhancing heat transfer has become more and more crucial. In order to reduce the size and cost of heat exchangers and to save energy for their operation, an air-side wavy fin-and-tube heat transfer surface with three-row tubes needs to be replaced by two-row tubes for the same heat transfer load with an allowance of about 10 percentage points of pressure-drop increase. The major purpose of the present paper is to present the numerical design process of such a new fin-tube structure. In the following is a brief review of the fin-tube heat transfer structures and the related enhancement techniques is presented, which are the fundamental ingredients of our numerical design.

The heat transfer technologies have been divided into four "generations" by Bergles [2], and the *fourth-generation heat transfer* (or third-generation enhancement) technology refers to that based on compound enhancement techniques: "Two or more techniques may be utilized simultaneously to produce an enhancement that is larger than the individual techniques applied separately." This is termed *compound enhancement*. The objective of this research is to combine some existing air-side enhancing techniques such that the already-mentioned three-row wavy fin-and-tube heat transfer surface can be replaced by a two-row surface.

Generally, there are four types of plate-fin surface: plain plate fins, corrugated plate fin, slotted fins, and fins with punched longitudinal vortex generators. In all fin surfaces, the plain fins present the lowest heat transfer rate at the same incoming air velocity. The performance of the wavy fin lies between the plain fin and slotted fin. As a common fin form of enhanced heat transfer, the waved fin surface provides a tool to improve the thermal performance of the heat exchanger. Extensive experimental [3–5] and numerical studies [6–8] have been conducted

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on air-side characteristics and the geometric parameter influence of wavy fin-and-tube heat exchangers, and some important conclusions have been obtained. Research by Wang and Vanka [6] shows that the wavy geometries provide little or no advantage at low *Re*, and periodic shedding of transverse vortices increases the heat transfer and pressure drop at higher Reynolds numbers. Thus, the wavy fin-and-tube exchanger is usually employed at higher inlet velocity. Wang et al. [9] experimentally investigated the effects of the waffle height on the air-side heat transfer and friction characteristics of herringbone wavy fin-and-tube heat exchanger. Their results show that compared to the plain plate fin, the heat transfer rate of the herringbone wavy fin is increased by 5–10%, but the pressure drop increased by 40–60% when the wavy angle is less than 20° and the fin pitch is smaller, and the heat transfer rate and pressure drop decrease with the increasing of fin pitch.

Longitudinal vortex generators (LVGs) are one of the novel heat transfer enhancement techniques, and their mechanism for heat transfer enhancement is different from that of transverse vortex generators (TVGs). The enhancement mechanism for transverse vortices requires unsteady flow and implies reversed-flow regions, while the enhancement mechanism for the longitudinal vortices consists of strong swirling around an axis essentially aligned with the main flow direction, which causes a heavy exchange of core and wall fluid [10]. Obviously, for the same exchange rate of wall and core fluid, less energy is needed to turn the flow around an axis aligned with the main flow direction than for the generation of swirl around an axis perpendicular to the main flow direction. Therefore, the longitudinal vortices are more efficient for heat transfer enhancement than transverse vortices when both heat transfer and pressure drop are taken into account. Some investigations [10, 11] have pointed out that LVGs are preferable to TVGs for compact heat exchangers, so the LVGs are widely used as the first choice enhancing technique. In a series of literature reports [12–19] different LVG types and influence parameters have been investigated in detail. From these investigations, some useful conclusions can be drawn: Punching gives slightly better performance than mounting [10]; wing-type vortex generators (WVGs) can easily be incorporated into compact heat exchangers [10, 18]; the most effective location for the delta winglet pair relative to a circular tube is the location behind the tube, one tube apart, at 45° angle of attack [10, 20]; and the vortex generator presents higher enhanced heat transfer capability for the inline tube arrangement than for the staggered tube arrangement [12, 16]. Also, it has been verified by Wu and Tao [21] that, like other enhancement techniques, the fundamental mechanism of heat transfer enhancement made by WVGs is the improvement of the synergy between velocity and fluid temperature gradient.

Based on the results just described, in the present study plain fins of two-row tubes with punched WVGs arranged in the rear of each tube are designed as a base structure. Meanwhile, the influence of additional WVGs and of their position arrangement on heat transfer augmentation has been investigated in conjunction with a slotted technique to meet the design requirement.

The slotted fins are probably the most successful families of enhanced plate fin surfaces currently in use, including the offset strip (slit fin) and louvered fin. To obtain the fluid and heat transfer performance and better understanding of the enhancement mechanisms, a lot of research has been accomplished as reported in [22–27]. These studies have found that because the fin surface is broken into several small pieces in these fin geometries, a new boundary layer forms while a new leading edge is encountered each time. Thus the average boundary-layer thickness is smaller for slotted fin surfaces than for continuous surfaces. A thinner boundary layer corresponds to a lower heat transfer resistance but higher skin friction. Generally, the louvered fin surface has a very high heat transfer coefficient, but its pressure-drop penalty is often so great that this prevents its wide application. Therefore, in the present study the offset strip (slit fin) and WVGs have been utilized simultaneously as a combined enhancing technique to further improve the heat transfer capacity. The heat transfer rates of several such combined enhancing fin surfaces have been numerically simulated, and are compared to that of the given wavy fin-and-tube heat transfer surface with three-row tubes. In the following, physical and mathematical models are first introduced, followed by the presentation of results. Finally, an approximate optimum new structure will be provided.

PHYSICAL AND MATHEMATICAL MODELS

Physical Model and Computational Domain

Figure1 shows the schematic diagram of a herringbone wavy fin-and-tube heat exchanger with three-row tubes, where Figures 1a and b present the front view and top view of the heat exchanger, respectively. There are three tube rows along the flow direction arranged in an aligned way. As can be seen from the figure, the circular tubes and the fins are arranged periodically in both the spanwise direction and the axial direction, respectively. Thus half of the unit between the two adjacent center lines of the flow channel (top view) and the space between two adjacent fin sheets can be regarded as a representative of the heat transfer surface. As far as the direction normal to the fin sheet is concerned, depending on the specific structure of the fin studied, there are two options for selection: One is putting the fluid in the center of the computational unit, which will be called practice A, and the other is putting the fin sheet in the center, which will be called practice B. Taking the case of the plain plate fin-and-tube heat exchanger with two-row tubes in the flow direction as an example, Figure 2 shows the two practices. In the present research, practice B is adopted for the wavy fin and practice A for all the enhanced fins. For the wavy fin surface the top view of the computational domain is the shaded region in Figure 1b.

In order to shorten the production period of a new model and reduce the manufacturing cost, the manufacturer requires that the arrangement mode of tube bundles and the global geometric parameters of the new heat exchanger surface are unchanged as

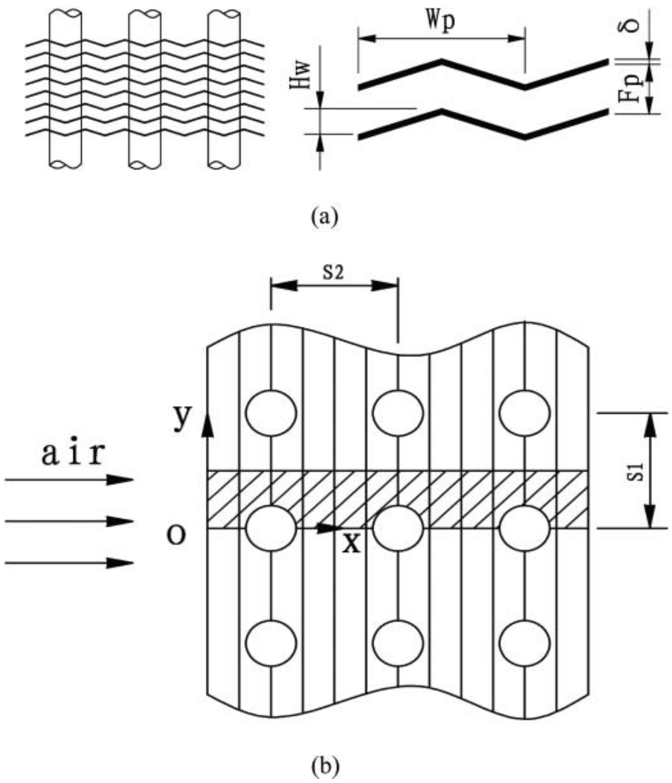


Figure 1 Schematic diagram of herringbone wavy fin-and-tube heat exchanger with three-row tubes. (a) Front view and (b) top view.

much as possible. So the new enhancing fin with two-row tubes arranged in an aligned way has the same global parameters as that of the given wavy fin with three-row tubes arranged in a line. These parameters are shown in Table 1.

Figure 3a presents a three-dimensional wavy fin unit used in numerical simulation, and its longitudinal cross-section view is shown in Figure 3b. The geometry parameters of the wavy fin are as follows: The wavy pitch (W_p, Figure 1a) is 12.5 mm; the wavy height (H_w, Figure 1a) is 1.5 mm; the processing chamfer angle is 45° (Figure 3b); and the diameter of the boss face is 15.0 mm (Figure 3b).

Figure 4 shows the three-dimensional geometry of seven types of plain fins of two-row tubes with longitudinal vortex generators. Figure 5 presents a top view of Fin W7. For all the seven fins, winglets are punched out from a fin sheet, the thickness of winglets is equal to that of the fin, and the height of the winglet (b) is equal to 0.9 times the channel height, with the angle of attack β of the winglet equal to 45°, and the winglet length (a) equal to 2 times the winglet height. The difference between Fin W1 and Fin W2 is in the punched position, and the difference between Fin W2 and Fin W3 is the location of the winglet. The main difference among the other four enhancing fins with delta winglet lies in the row number and column number of the delta winglet: Fin W4 is of two rows and two columns; Fin W5 has two rows and three columns; Fin W6 is of four rows and two columns; Fin W7 has four rows and three columns. The distance between two adjacent columns in the y direction on each row (L1) is 6.5 mm for Fin W4 and Fin W6, and the distances between two adjacent columns (L1, L2) are 1.0 mm and 6.5 mm, respectively, for Fin W5 and Fin W7.

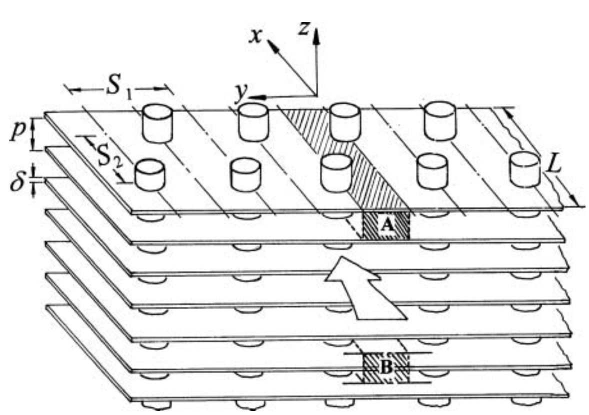


Figure 2 Computational units of plain plate fin-and-tube heat exchanger.

Table 1 Geometric parameters of wavy fin

Parameter name	Parameter value
Fin pitch f_p, mm	2.0
Fin thickness δ, mm	0.11
Tube diameter D_o, mm	9.52
Expanded tube diameter, mm	9.83
Transverse tube spacing S_1, mm	25
Longitudinal tube spacing S_2, mm	25
Fin collar outside diameter, mm	10.05

Based on our preliminary simulation results, the adoption of the vortex generators only is not able to reduce the row number from 3 to 2 with approximately the same pressure drop. Thus, a combination design of the winglet vortex generators with slotted protruding parallel strips is proposed to further improve the heat transfer performance, and different combinations of their arrangement are tried. Figure 6 presents the geometry of four combined enhanced fins. In all four fins, the geometry parameters of the winglet are the same as that of Fin W6. The difference among the four combined enhanced fins lies in location or number of the protruding strips, as can be clearly observed in Figure 6. The geometric parameters of the protruding strip are as follows (Figure 7): Strip pitch (L_s) is 1.3 mm; strip width (L_f) is 1.3 mm; strip height is 1 mm (1/2 fin pitch); strip edge position (L_a) is 0.5 mm; radius of localization circle of slit (R_s) is 7.2 mm; and the strip lengths (L_w) are 10 mm, 5.4 mm, 4.8 mm, 5.4 mm, 9 mm, 10 mm, 11 mm, 5.4 mm, 4.8 mm, 5.4 mm, and 10 mm along the flow direction, respectively.

Governing Equations and Boundary Conditions

The maximum Reynolds number studied in this paper is less than 3500; according to the analysis in the literature [7, 19, 27–29] the air flow can be assumed to be three-dimensional, incompressible, laminar, and steady flow; and the fluid

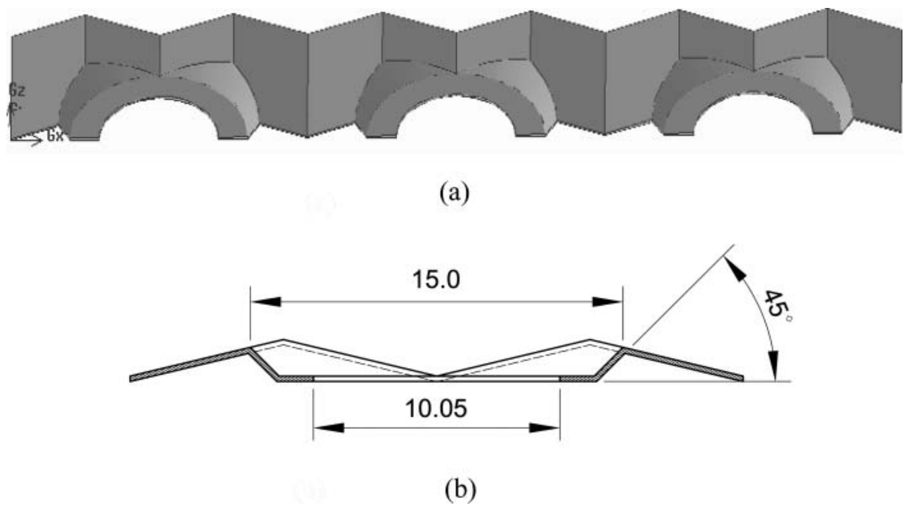


Figure 3 Wavy fin unit used in numerical simulation. (a) Three-dimensional view and (b) Longitudinal cross section.

thermophysical properties are constant within the range of computing inlet velocity and temperature difference in the numerical simulation. Thus, in the three-dimensional Cartesian coordinate system, the governing equations for mass, momentum, and energy conservation can be expressed as follows:

Continuity equation:

$$\frac{\partial}{\partial x_i}(\rho u_i) = 0 \tag{1}$$

Momentum equation:

$$\frac{\partial}{\partial x_i}(\rho u_i u_k) = \frac{\partial}{\partial x_i}\left(\mu \frac{\partial u_k}{\partial x_i}\right) - \frac{\partial p}{\partial x_k} \tag{2}$$

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 $\Gamma = \lambda / c_p$.

In order to ensure the accuracy of numerical simulation results, the fin thickness is taken into account in the present calculation. Thus, the problem becomes a conjugated one, and the fin and fluid temperatures should be solved simultaneously. Due to the fact that the fin surfaces are part of the solution domain, no conditions at the fin surfaces are required. Considering that the governing equations are elliptic, boundary conditions are required for all boundaries of the computation domain.

In order to use the uniform inlet condition and fully developed outflow condition, the computational domain is extended in both the upstream and downstream parts, with one time and six times of longitudinal tube spacing respectively, and the two extended parts are called the pre-extended and after-extended region, respectively. Take the case of practice B as an example, by neglecting the details of fin surface, the computational domain is represented in Figure 8.

Followings are the boundary conditions for numerical simulation in this paper.

Boundary conditions in the x coordinate direction are:
At the inlet:

$$u = \text{const}; \quad v = w = 0; \quad T_{\text{in}} = \text{const} \tag{4}$$

At the outlet:

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

Boundary conditions in the y coordinate direction are:
Fluid region:

$$\frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0; \quad v = 0; \quad \frac{\partial T}{\partial y} = 0 \tag{6}$$

Fin surface region:

$$u = v = w = 0; \quad \frac{\partial T}{\partial y} = 0 \tag{7}$$

Tube region:

$$u = v = w = 0, T_{\text{w}} = \text{const} \tag{8}$$

Boundary conditions in the z coordinate direction are:

In the pre-extended region: symmetry condition.
In the fin coil region and after-extended region: periodic condition.

Because the thermal resistance of the fluid side inside the tube is much less than that of the air side outside the tube and the tube wall (copper) has very high thermal conductivity, a constant tube wall temperature is assumed in the present study.

NUMERICAL METHODS

In this paper the grid system is generated by commercial software GAMBIT. The governing equations are discretized by

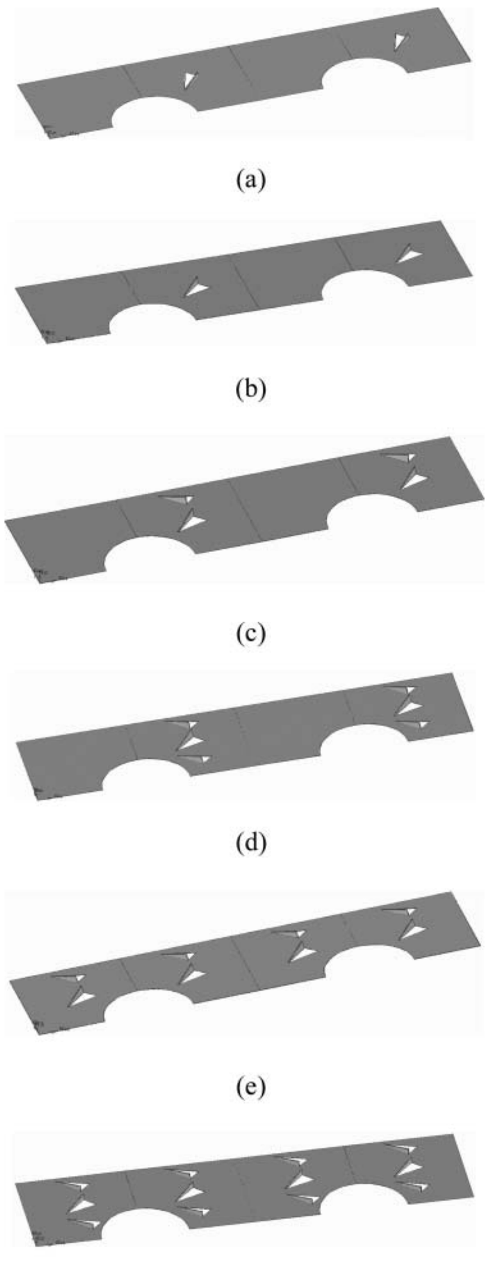


Figure 4 Geometry of seven fins with delta winglet. (a) Fin W1, (b) Fin W2, Fin W3, (c) Fin W4, (d) Fin W5, (e) Fin W6 and (f) Fin W7.

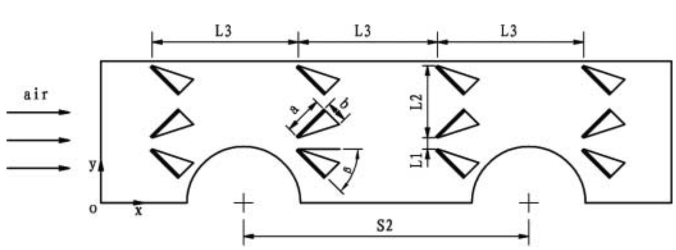


Figure 5 Top view of Fin W7.

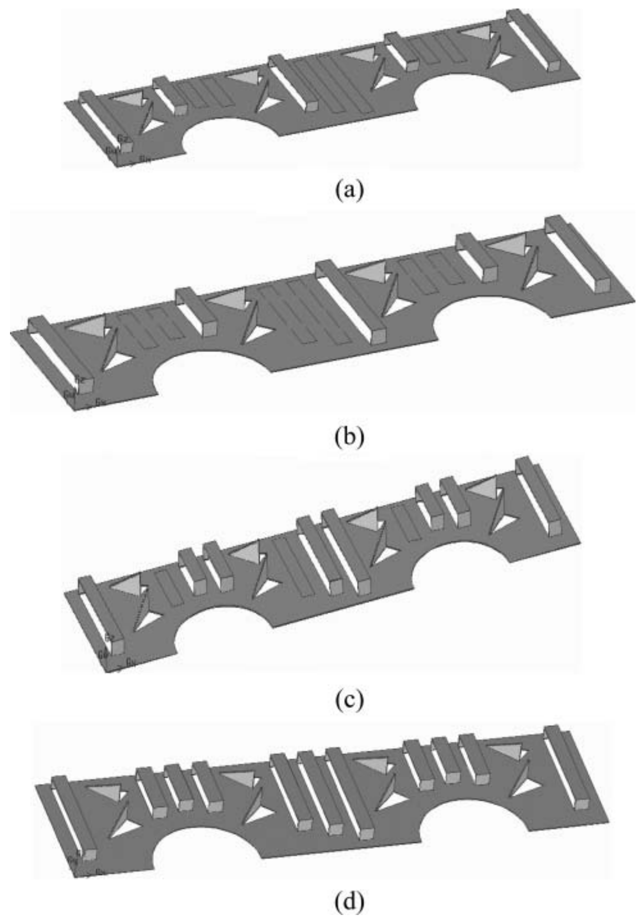


Figure 6 Geometry of four combined enhancing fins. (a) Fin C1, (b) Fin C2, (c) Fin C3 and (d) Fin C4.

the finite-volume method. The elliptic equations are solved by the full-field computational method. Because of the conjugated nature of the problem, the fin surfaces are considered as a part of the solution domain. The second-order upwind is used to discretize the convective terms in momentum and energy equations, and the diffusive term is discretized by the central difference. The SIMPLEC algorithm is adopted to deal with the linkage between velocity and pressure. The problem is solved by using FLUENT. In order to accelerate computation, parallel computing is adopted. The convergence criterion in numerical simulation is set as follows: The reduction of the residuals is below the order of 10^{-3}–10^{-4} for mass conservation equation, 10^{-6}–10^{-7} for momentum equations, and 10^{-7}–10^{-8} for the energy equation, respectively. In order to obtain an appropriate grid system, the influence of grid density on the computational results was investigated, and all the numerical results can be regarded as grid-independent. The specific grid number depends on the case, typically in the range of 1.16 and 0.92 million for wavy fin and enhanced fins, respectively. As an example, Figure 9 and Figure 10 present the local computing grid network of the wavy fin and combined enhancing fin (Fin C4), respectively. From Figure 9 it can be seen that the complicated geometry structure of the wavy-fin surface has been simulated accurately.

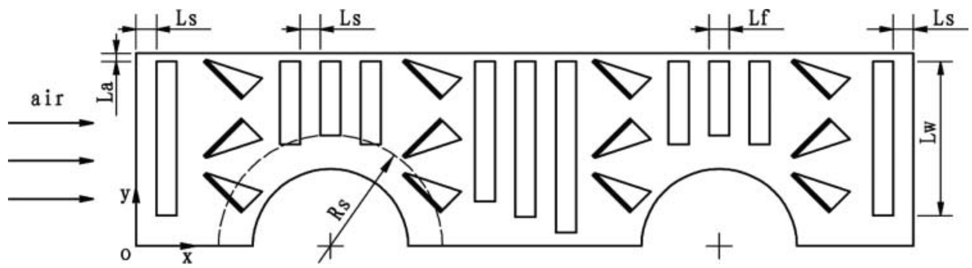


Figure 7 Top view of Fin C4.

RESULTS AND DISCUSSION

Considering the practical applications of the air heaters studied, the calculation parameters are set up as follows: The inlet velocity of air is 2.0 m/s, 2.25 m/s, 2.5 m/s, 2.75 m/s, and 3.0 m/s, respectively. The inlet temperature of air is 294 K, and the wall temperature of tubes is 333 K. The simulated tube size is the fin collar outside diameter and it is the characteristic length of the air-side Reynolds.

In order to facilitate comparisons of fluid flow and heat transfer performances, the heat transfer rate and pressure drop of the wavy fin with three-row tubes arranged in a line have been obtained in the preliminary computation by a numerical simulation method, and the results are shown in Figure 11.

The Enhanced Fins of Two-Row Tubes With Different WVGs

Three types of plain fins of two-row tubes with a longitudinal vortex generator arranged in the rear of each tube have been simulated first.

Figures 12 and 13 present comparisons of heat transfer and pressure drop of the three fins in a computing unit, respectively. It can be found from the figure that Fin W2 possesses higher heat transfer and lower pressure drop than that of Fin W1. So the punched position behind the winglet of W2 is chosen in the following simulation. The difference between Fin W2 and Fin W3 is the location for the delta winglet pair relative to the circular tube: Fin W2 is one tube apart, and 0.9 for Fin W3. We can also see from the figures that Fin W2 presents higher fluid flow and heat transfer performance than Fin W3; i.e., the heat transfer rate of W2 is a bit higher than that of W3 while its pressure drop is a bit lower than that of W3. This result is consistent with research conclusions by Fiebig [10]. As far as a plain fin of two-row tubes with a longitudinal vortex generator arranged in the rear of each tube is concerned, Fin W2 is better. But the difference is minor, and the calculated results indicate that the maximum deviation of the heat transfer rate and pressure drop between Fin W2 and Fin W3 is 2.14% and 1.60%, respectively.

Compared with the wavy fin surface of three-row tubes, the heat transfer rate and pressure drop of the enhanced plate fin surface of two-row tubes with single delta winglet (Fin W2) behind the tubes reduces to 75.4–71.7% and 57.2–58.3%, respectively, under the same frontal cross-section area in the inlet velocity range of 2.0–3.0 m/s. This implies that the longitudinal vortex generator can effectively reduce the pressure drop with a mild reduction of heat transfer.

Figure 14 shows velocity profiles for the plate plain fin and Fin W2 on the middle plane (parallel to the x–y coordinate plane) of the computational channel at an inlet velocity of 2.5 m/s. In the figure, it can be observed that vortices are generated in the wake region behind the tube where the heat transfer is deteriorated. The existence of the delta winglets arranged behind the aligned tubes reduces the wake region and hence improves the heat transfer behind the tube. Figure 15 presents the spanwise average local Nusselt number distributions for a plate plain fin and Fin W2. It can be seen from the figure that the local Nusselt numbers around the region within which the delta winglets are located are appreciably enhanced.

On the other hand, the influence area of a single longitudinal vortex generator is limited, as can be found from the Figure 14b. In order to enlarge the influence area of longitudinal vortices,

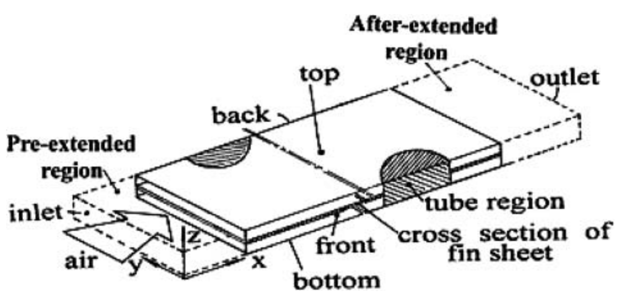


Figure 8 Computational domain.

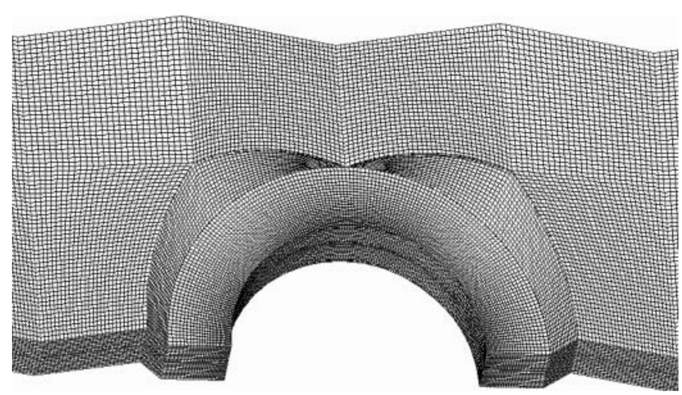
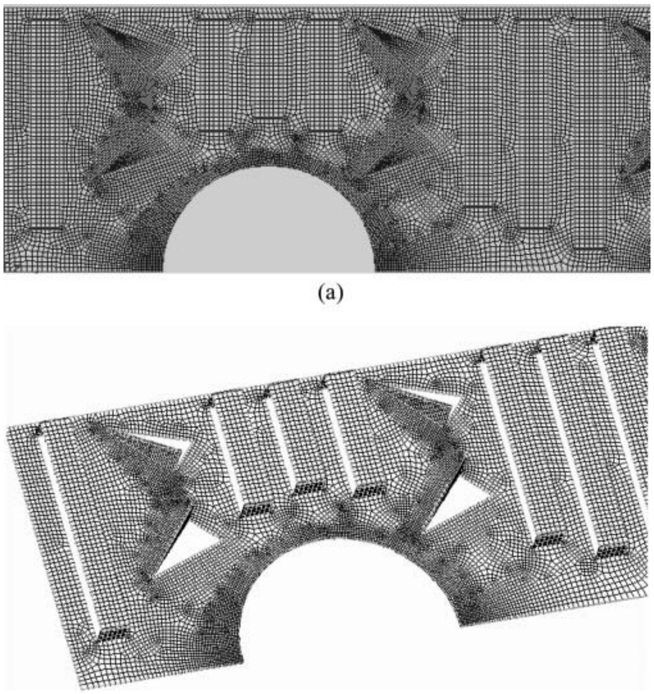


Figure 9 Local computing grid network of the wavy fin.



(a)

(b)

Figure 10 Local computing grid network of the Fin C4. (a) Top view of local computing grid network and (b) Gird network of fin surface (local).

two and three delta winglet vortex generators aligned spanwise behind each tube are tried (see W4 in Figure 4c and W5 in Figure 4d). Figures 16 and 17 present the heat-transfer and pressure-drop characteristics of the two enhanced fin surfaces, respectively. In the figures, the heat transfer rate and pressure drop of the two fin surfaces have a larger increment than that with the single winglet, but the heat transfer rate is still significantly lower than that of the wavy fin surface with three-row tubes. It should be noted that Fin W4 with two winglets presents a higher heat transfer rate and lower pressure drop than Fin W5 with three

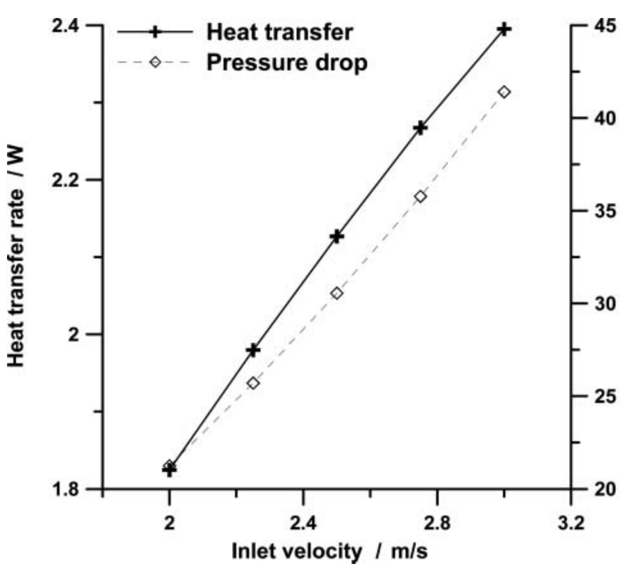


Figure 11 Heat transfer and pressure drop of wavy fin.

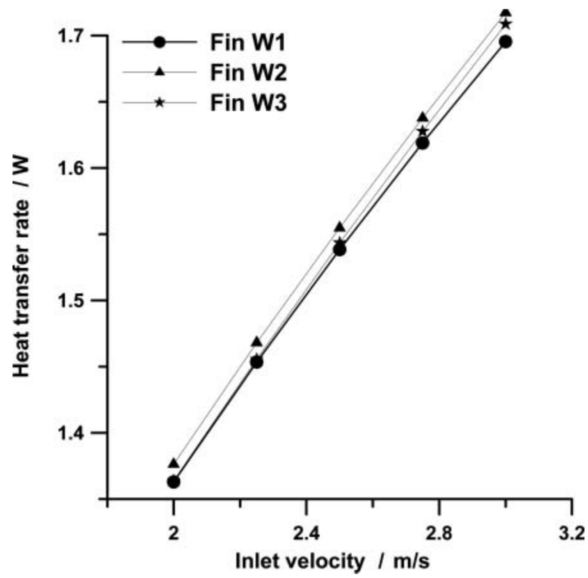


Figure 12 Heat transfer rate of fins with a winglet behind tube.

winglets. It is estimated that the winglet nearest to the tube might be situated in the wake region already. This estimation is demonstrated by Figure 18. Thus, the delta winglet nearest the tube in W5 not only fails to improve heat transfer, but also increases pressure loss. Compared with the wavy fin surface of three-row tubes, the heat transfer rate and pressure drop of the enhanced plate fin surface W4 reduce to 83.1–81.4% and 67.1–68.2%, respectively, under the same frontal cross-section area in the given inlet velocity range of 2.0–3.0 m/s.

For further enhancing heat transfer, two new plain fins of two-row tubes with two or three delta winglets aligned spanwise and arranged at four streamwise locations are designed, as shown in Figures 4e and f, respectively. The heat-transfer and pressure-drop characteristics of Fin W6 and Fin W7 are shown in Figures 16 and 17, respectively.

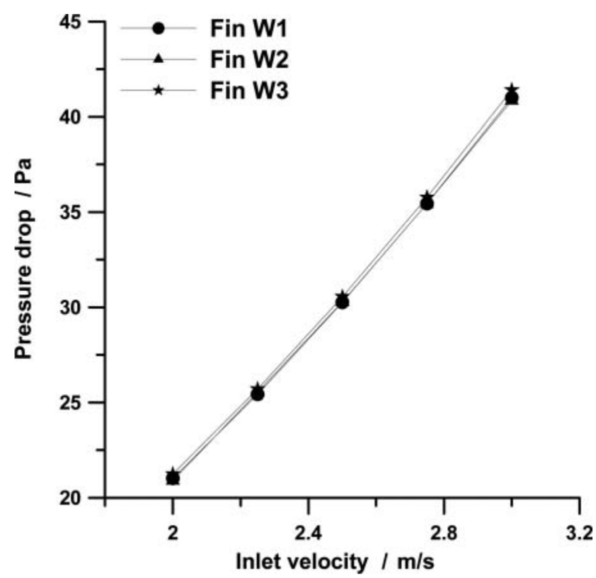


Figure 13 Pressure drops of fins with a winglet behind tube.

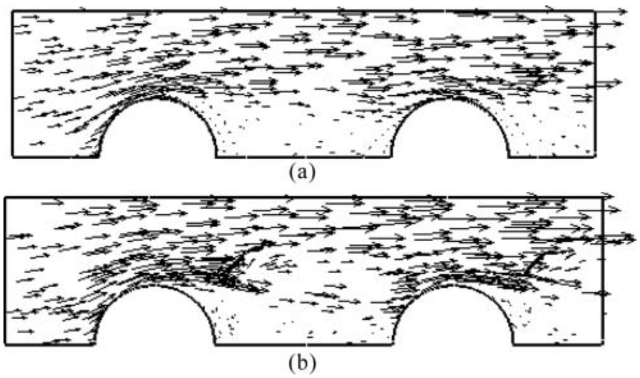


Figure 14 Velocity profiles for plate plain fin and Fin W2 on middle plane of the flow channel. (a) Plate plain fin and (b) Fin W2.

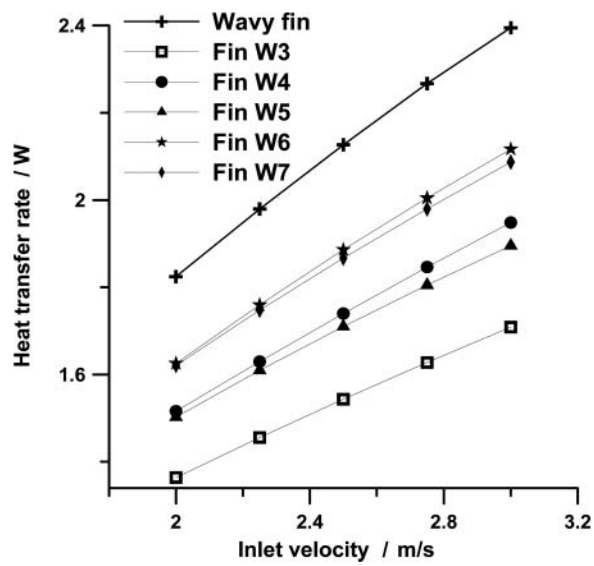


Figure 16 Heat transfer characteristics of enhancing fins.

We can see from the figures that Fin W6 presents a higher heat transfer rate and lower pressure drop that Fin W7, and this can be explained by the same reason as that for the difference between W4 and W5. Figure 19 shows the spanwise average local Nusselt numbers for Fin W2 and Fin W6.

The heat transfer rate and pressure drop of the enhanced plate fin surface W6 reduce to 89.1–88.4% and 82.7–85.5% relative to the wavy fin surface, respectively, under the same frontal cross-section area in the given inlet velocity range of 2.0–3.0 m/s.

The Combined Enhancing Fins With WVGS and Protruding Stripes

In order to increase the air-side heat transfer rate, further techniques should be used. Considering that the slotted fin with protruding strips parallel to the basic sheet can effectively enhance heat transfer with a mild pressure-drop penalty, a combination design of the longitudinal vortex generator with slotted protruding parallel strips is proposed and different variations of their arrangement are tried.

According to the already-described research on the plain fin with longitudinal vortex generator, Fin W6 is adopted as the basic fin form. In this paper we have designed four enhanced structures with combined techniques as shown in Figure 6 (Fin C1–Fin C4).

Figures 20 and 21 present the heat-transfer and pressure-drop characteristics of combined enhanced fins C1 and C2, respectively. The heat transfer rate of Fin C2 is higher than that of Fin C1, and the maximal deviation is 4.81% under the inlet velocity of 2.0–3.0 m/s. The pressure drops of Fin C1 and Fin C2 are obviously higher than that of Fin W6, and very close to that of the wavy fin with three-row tubes. The maximal deviation of pressure drop among the combined enhanced fins and wavy fin is less than 1% under an inlet velocity of 2.0–3.0 m/s. However, the heat transfer rate of Fin C2 is still lower than that of the wavy fin with three tube rows, being 97.6–96.2% for the same frontal cross-section area in an inlet velocity range of 2.0–3.0 m/s.

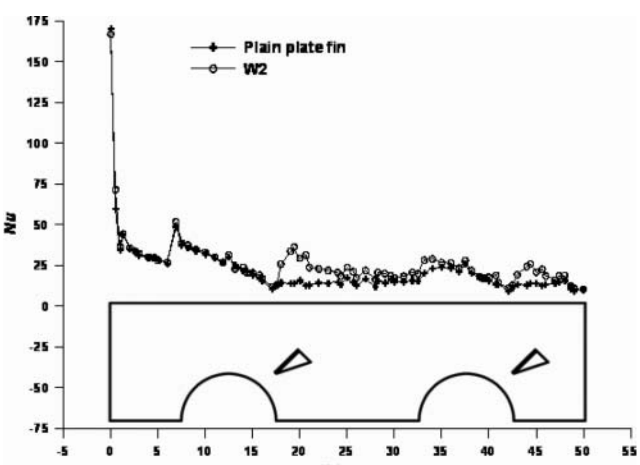


Figure 15 Local Nusselt number for plate plain fin and Fin W2.

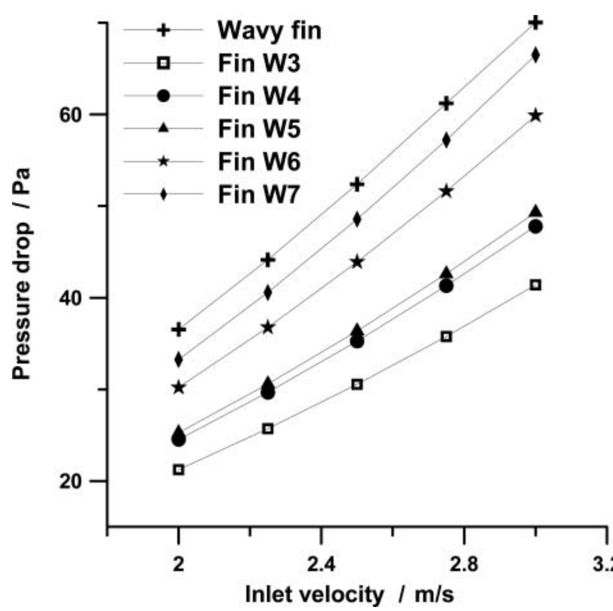


Figure 17 Pressure drop characteristics of enhancing fins.

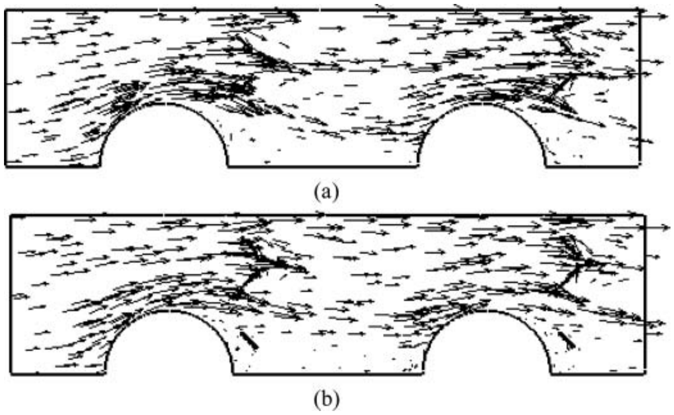


Figure 18 Velocity profiles for Fin W4 and Fin W5 on middle plane of the flow channel. (a) Fin W4 and (b) Fin W5.

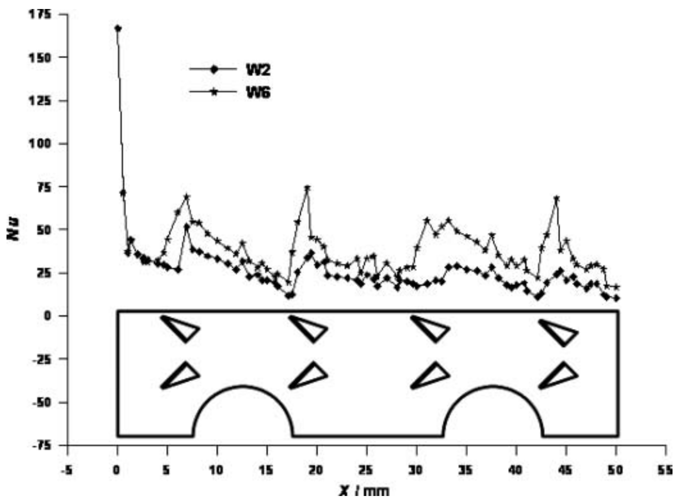


Figure 19 Local Nusselt numbers for Fin W2 and Fin W6.

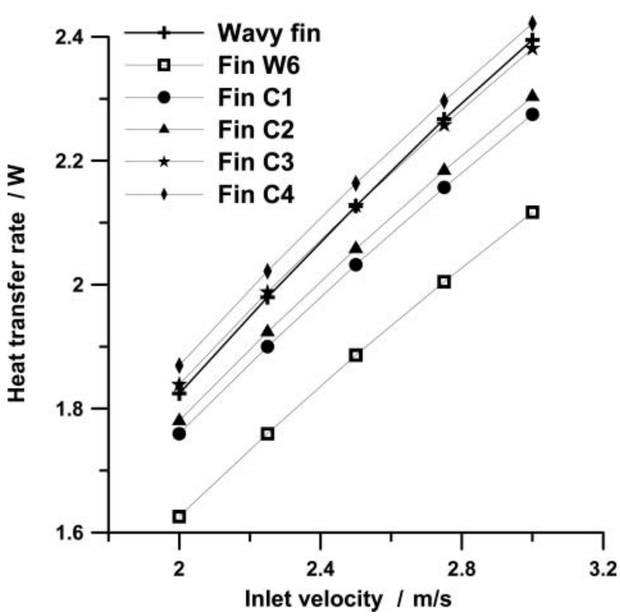


Figure 20 Heat transfer rates of combined enhancing fins.

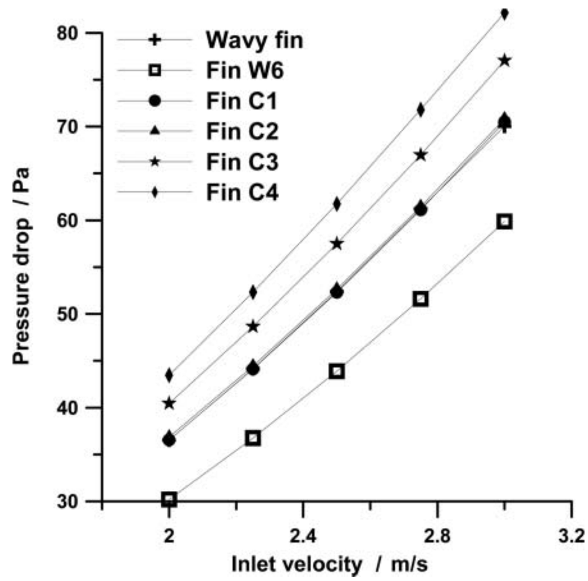


Figure 21 Pressure drops of combined enhancing fins.

In order to further enhance heat transfer, Fin C3 and Fin C4 are designed based on Fin C2, where two and three protruding strips are adopted between the spaces confined by the four spanwise aligned winglets, respectively.

The heat transfer and pressure drop characteristics of Fin C3 and Fin C4 are also shown in Figures 20 and 21, respectively. The heat transfer rate of Fin C3 is slightly higher than that of the wavy fin in lower inlet velocities and slightly lower in higher inlet velocities, and the maximum deviation is less than 0.78%. Therefore, as far as the heat transfer rate is concerned, Fin C3 can basically meet the requirement and Fin C4 can completely meet the requirement. Correspondingly, Fin C3 and Fin C4 present a higher pressure drop than that of the wavy fin. The heat transfer rate of Fin C4 is larger than that of the wavy fin by 2.46–1.11%, and the pressure drops of Fin C3 and Fin C4 are larger than that of the wavy fin by 10.7–9.4% and 18.9–17.3%, respectively, for the same frontal cross-section area in inlet velocity of 2.0–3.0 m/s. Figure 22 shows the local Nusselt numbers for the plain fin, Fin

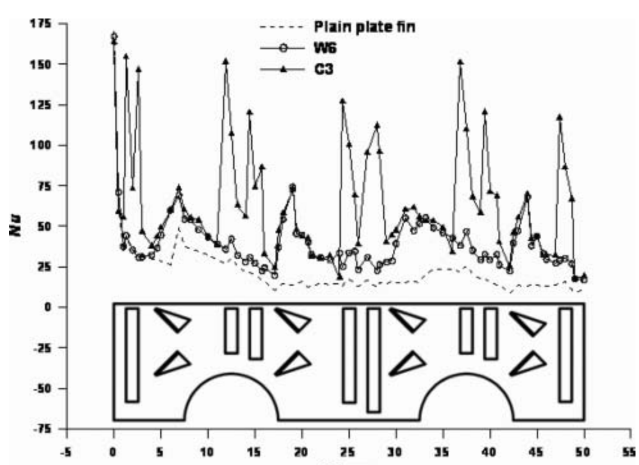


Figure 22 Local Nusselt numbers for Fin W6 and Fin C3.

W6, and Fin C3. From the figure the heat-transfer enhancement contributions from strips and winglets can be clearly observed.

Thus, the combined enhanced fin-and-tube heat exchanger surfaces of two-row tubes with two or three protruding strips behind the first three rows of winglet (Fin C3 and Fin C4) can meet the heat transfer rate requirement of the given wavy fin-and-tube exchanger with three-row tubes, and save the volume of heat exchanger, fin, and tube material by about 33%.

CONCLUSIONS

In the numerical design study of replacing a given wavy fin with three-row tubes by enhanced structures of two-row tubes, the following conclusions can be drawn:

1. The fin with winglet punched in the rear of a tube is an effective way to enhance heat transfer with a reasonable pressure-drop penalty. The winglet location of one tube diameter apart relative to the circular tube possesses better performance than for 0.9 tube diameter apart.
2. Within the variants of different locations and number of winglets tried in this paper, winglets only cannot meet the heat transfer requirement of this replacement study.
3. The combined enhanced fin by adopting delta winglet and protruding parallel strips simultaneously can effectively improve the heat transfer rate; Fin C3 and Fin C4 both can meet the heat transfer requirement of the wavy fin with three-row tube, with Fin C4 having a bit larger pressure-drop penalty.
4. Considering both heat transfer enhancement and pressure drop penalty, Fin C3 is recommended. The heat exchanger volume and metal materials of the new structure can be saved by about 33%.

NOMENCLATURE

a	winglet length, m
b	winglet height, m
c_p	specific heat, kJ kg^{-1} K^{-1}
F_p	fin pitch, m
H_w	wavy height, m
L_a	location parameter of strip, m
L_1, L_2, L_3	location parameter of winglet, m
L_f	strips width, m
L_s	strip pitch, m
L_w	strip length, m
p	pressure, Pa
R_s	localization circle radius, m
S_1	transverse tube spacing, m
S_2	longitudinal tube spacing, m
u, v, w	velocity component, m s^{-1}
W_p	wavy pitch, m

Greek Symbols

Γ	diffusion coefficient, kg m^{-1} s^{-1}
β	attack angles of winglet, degrees
δ	fin thickness, m
ρ	density, kg m^{-3}
μ	dynamic viscosity, kg m^{-1} s^{-1}
λ	thermal conductivity, W m^{-1} K^{-1}

Subscripts

i	summation indicators
in	inlet
k	free indicators
w	wall

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