Numerical study on performance and fin efficiency of wavy fin-and-tube heat exchangers

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Abstract: Three-dimensional numerical studies were performed for the performance of wavy fin-and-tube heat exchangers in Body-Fitted Coordinates (BFC) system. Effects of geometric parameters on air-side heat transfer and fluid flow characteristics and fin efficiency were examined. The results showed that with the increase in Reynolds number, wavy angle, fin thickness and the decrease in fin pitch and transverse tube pitch, the heat transfer performance are enhanced; however, pressure drops are also increased. So, in practical applications, the wavy angle had better be located between 10° and 20° and the fin pitch should be located between 1.2 mm and 2.0 mm. The fin efficiency and average fin surface temperature decrease with the increase of Reynolds number, wavy angle, fin pitch and transverse tube pitch. With the increase of fin thickness, the fin efficiency and average temperature on fin surface also increase.

Keywords: wavy fin; geometric parameters; heat transfer and fluid flow performance; fin efficiency.

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1 Introduction

To save energy and reduce production cost, high-efficient and compact heat exchangers are required in many fields. For the compact heat exchangers, since the dominant thermal resistance is on air side, improving air-side fin configuration and enhancing its heat transfer performance are the most effective way to improve the performance. Wavy fins can periodically change the air flow direction and thin the flow and thermal boundaries, which are benefit to reduce the air-side thermal resistance. So, wavy fin-and-tube heat exchangers have been widely used in many engineering applications. Its air-side heat transfer and fluid flow performance have been numerically and experimentally studied by many researchers.

In experimental studies, Xin et al. (1994) studied the heat transfer and pressure drop characteristics of nine triangular wavy fin-and-tube heat exchanger surfaces with different fin pitch and tube row number. Wang et al. (1997, 2002) performed extensive experiments on the heat transfer and pressure drop characteristics of wavy fin-and-tube heat exchangers and a lot of correlations for heat transfer coefficient and friction factor were provided. Wongwises and Chokeman (2004, 2005) experimentally investigated the effects of fin thickness, fin pitch and tube row number on the air-side performance of herringbone wavy fin and tube heat exchangers. In numerical analysis, Jang and Chen (1997) numerically studied the heat transfer and fluid flow performance in a wavy fin-and-tube heat exchanger. Min and Webb (2001) investigated the performance of wavy fin coil. Manglik et al. (2005) numerically analysed the effects of fin density on the forced convection in three-dimensional (3-D) wavy-plate-fin compact channels. He et al. (2005) performed 3-D numerical study of heat transfer characteristics of plain plate fin-and-tube heat exchangers from viewpoint of field synergy principle. In recent years, several 3-D numerical studies on the air-side performance of wavy fin-and-tube heat exchangers were performed in BFC by Tao et al. (2007a, 2007b, 2007c) and He et al. (2008).

At the same time, some investigations on fin efficiency of plain plate fin-and-tube heat exchangers were performed. Arstanturk (2005) used the decomposition method to evaluate the fin efficiency and determine the temperature distribution of straight fins with temperature-dependent thermal conductivity. Rajabi (2007) adopted the Homotopy Perturbation Method (HPM) to evaluate the efficiency of straight fins and determine the temperature distribution within the fin. Sharqawy and Zubair (2007) performed the analysis on the efficiency of annular fin and the temperature distribution over the fin surface was obtained. Tao et al. (2007d) numerically studied the local heat transfer coefficient and fin efficiency of wavy fin-and-tube heat exchangers.

In the previous studies, the main emphasis was focused on studying the effects of parameters on the average air-side heat transfer and pressure drop characteristics of wavy fin-and-tube heat exchanger. And, all the previous studies on the fin efficiency were almost performed for the simple fin patterns, such as straight fins and annular fins. Due to the complicated configuration of the wavy fin-and-tube heat exchanger, little studies about the effects of geometric parameters on the fin efficiency of wavy fin-and-tube heat exchanger had been reported.

In this study, 3-D numerical studies on the air-side heat transfer and fluid flow performance of wavy fin-and-tube heat exchangers were performed by BFC system. Effects of Reynolds number, wavy angle, fin pitch, fin thickness and transverse tube pitch on the heat transfer and fluid flow characteristics and fin efficiency of wavy fin surface were investigated.

2 Model descriptions

2.1 Physical model

The schematic diagram of the wavy fin-and-tube heat exchanger is shown in Figure 1(a) with two rows of tubes in the flow direction. Figure 1(b) shows the computational domain of the wavy fin coil. The air flow direction is *x*-direction, fin spanwise direction is *y*-direction and fin thickness direction is *z*-direction, as shown in Figure 1. All the geometric parameters for the heat exchanger are presented in Table 1. The computational domain was extended 1.5 times of the original heat transfer zone at the upstream and 5 times of the original heat transfer zone at the downstream.

Figure 1 Schematic diagram and computational domain of wavy fin-and-tube heat exchanger: (a) schematic diagram and (b) computational domain (unit, mm)

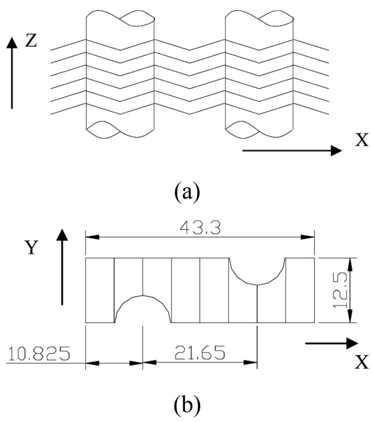


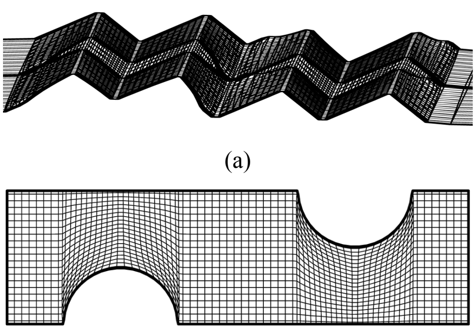
Table 1 Geometric dimension for the studied heat exchanger

Tube row number	2
Tube outside diameter (mm)	10.55
Transverse pitch (mm)	25.0
Longitudinal pitch (mm)	21.65
Fin pitch (mm)	2.0
Fin thickness (mm)	0.2
Wavy angle (°)	17.44
Wavy length (mm)	10.825
Air flow direction length (mm)	43.3

2.2 Grids generation technique

For wavy fin-and-tube heat exchangers, the configuration is very complicated. It is difficult to generate the computational grids in Cartesian coordinates, which results in the 3-D numerical studies had not been widely performed in the past decades. In recent years, the air-side heat transfer performance of wavy fin-and-tube heat exchangers are investigated by Tao et al. (2007a, 2007b, 2007c, 2007d) with BFC method. BFC method can transform the complex computational domain in physical space into a simple domain in computational space. In this paper, the BFC method is adopted to generate computational grid, and Poisson equation is used as governing equations (Thompson et al., 1974). The computational grid systems generated by BFC method is shown in Figure 2.

Figure 2 Schematic of grid systems generated by Body-Fitted Coordinates: (a) three-dimensional grid system for the whole computational domain and (b) two-dimensional grid system for fin coil region (*x*–*y* plane)



(a)

(b)

2.3 Governing equations

The governing equations for the forced steady, laminar, incompressible fluid flow and heat transfer in the physical space are presented 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

$$\frac{\partial}{\partial x_i}(\rho u_i T) = \frac{\partial}{\partial x_i}\left(\frac{k}{C_p}\frac{\partial T}{\partial x_i}\right). \tag{3}$$

The governing equations in the computational space are

$$\frac{\partial}{\partial \xi^j}(\rho U^j) = 0 \tag{4}$$

$$\frac{\partial}{\partial \xi^j}(\rho u_k U^j) = \frac{\partial}{\partial \xi^j}\left(\Gamma \frac{G^{jn}}{J}\frac{\partial u_k}{\partial \xi^n}\right) - A_k^j \frac{\partial p}{\partial \xi^j} \tag{5}$$

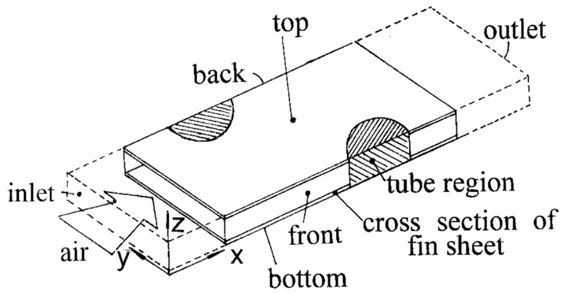
$$\frac{\partial}{\partial \xi^j}(\rho U^j T) = \frac{\partial}{\partial \xi^j}\left(\Gamma \frac{G^{jn}}{J}\frac{\partial T}{\partial \xi^n}\right) \tag{6}$$

where U^j is the velocity component in transformed space.

2.4 Boundary conditions

To study the effects of the geometrical parameters on the fin efficiency, the heat conductivity in the fins must be considered. In this study, the fin surfaces are considered as a part of the solution domain. The fin is made of pure aluminium, and the thermal conductivity is 236 W/(m·K). The air inlet temperature is set at 30°C and the tube surface temperature is fixed at 40°C. The definition of the boundary surfaces is shown in Figure 3. The boundary conditions are described as follows.

Figure 3 A pictorial view of the computational domain



1 Inlet and outlet boundary

Inlet: $u = u_{\text{in}} = \text{const}, v = w = 0, T = T_{\text{in}} = \text{const}$

Outlet: $\frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial w}{\partial x} = \frac{\partial T}{\partial x} = 0$

2 Top and bottom boundaries

Fluid region:

$$\frac{\partial u}{\partial z} = \frac{\partial v}{\partial z} = 0, w = 0, \frac{\partial T}{\partial z} = 0$$

Fin region:

Velocity condition: $u = v = w = 0$

Temperature condition: periodic conditions

3 Front and back boundaries

Fluid region: $\frac{\partial u}{\partial y} = \frac{\partial w}{\partial y} = 0, v = 0, \frac{\partial T}{\partial y} = 0$

Fin surface region: $u = v = w = 0, \frac{\partial T}{\partial y} = 0$

Tube region: $u = v = w = 0, T = T_w = \text{const.}$

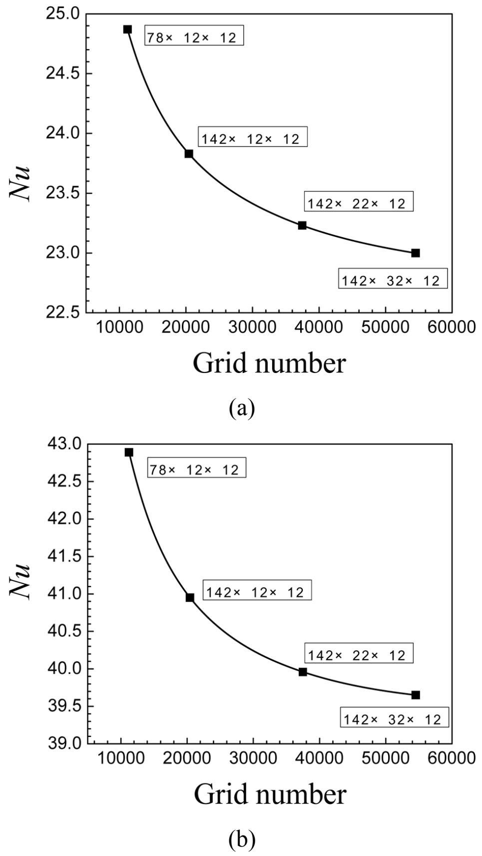
2.5 Convergence criterion

The fluid-solid conjugated heat transfer problem between fins and air is solved by the full-field computation method.

The fins in the computational domain are regarded as a special fluid of infinite viscosity. The harmonic mean method is adopted for the interface diffusion coefficient. To guarantee the continuity of the flux rate at the interface, the thermal conductivity of the fin and fluid adopts individual values, while the heat capacity of the fin takes the value of the fluid (Tao, 2001). The computational domain is discretised by nonuniform grids with the grids in the fin coil region being finer and those in the extended domains being coarser. The coupling between pressure and velocity is implemented by the SIMPLE algorithm. The governing equations are discretised by the finite volume method, and the convection term is discretised by the power-law scheme. The convergence criterion for the velocities is that the maximum relative mass residual of the cells is less than 1.0×10^{-6}, and the criterion for temperature is that the relative difference between the two heat transfer coefficients obtained from two iterations separated by 100 successive iterations is less than 1.0×10^{-6}.

To validate the solution independency of the grid, four different grid systems ($78\times12\times12$, $142\times12\times12$, $142\times22\times12$, $142\times32\times12$)) were investigated as shown in Figure 4 at $\mathrm{Re}=1000$ and $\mathrm{Re}=4000$, respectively. The validation results show that the grid $142\times22\times12$ yields a Nusselt number 1.0% higher than that of the finest grid $142\times32\times12$ at both Re conditions. So, in this paper, the grid system $142\times22\times12$ is adopted as the final computational grid.

Figure 4 Variation of the predicted Nusselt number with grid number systems: (a) Re = 1000 and (b) Re = 4000



3 Model validations

The Reynolds number, Nusselt number and friction factor used in the following sections are defined as follows:

$$\mathrm{Re}=U_{\mathrm{c}}D/\upsilon \tag{7}$$

$$\mathrm{Nu}=hD/k \tag{8}$$

$$f=\Delta pD/[(1/2)\rho U_c^2 L] \tag{9}$$

where U_c is the mean velocity in the minimum air flow cross-section, υ and k are viscosity and thermal conductivity, D is the outside tube diameter, L is the fin length along the air flow direction and Δp is the pressure drop of air flowing over L length.

The average temperature and pressure at the cross-section are defined as

$$\overline{T}=\frac{\iint_A uT\mathrm{d}A}{\iint_A u\mathrm{d}A} \tag{10}$$

$$\overline{p}=\frac{\iint_A p\mathrm{d}A}{\iint_A \mathrm{d}A}. \tag{11}$$

The heat transfer capacity, pressure drop and logarithmic mean temperature difference are defined as

$$Q=\dot{m}C_p(\overline{T}_{\mathrm{out}}-\overline{T}_{\mathrm{in}}) \tag{12}$$

$$\Delta P=\overline{p}_{\mathrm{in}}-\overline{p}_{\mathrm{out}} \tag{13}$$

$$\Delta T=\frac{(T_w-\overline{T}_{\mathrm{in}})-(T_w-\overline{T}_{\mathrm{out}})}{\ln[(T_w-\overline{T}_{\mathrm{in}})/(T_w-\overline{T}_{\mathrm{out}})]}. \tag{14}$$

Then, the heat transfer coefficient can be derived as

$$h=Q/(A\Delta T). \tag{15}$$

The fin efficiency is defined as

$$\eta_f=\frac{Q_{\mathrm{real}}}{Q_{\mathrm{ideal}}} \tag{16}$$

where, Q_{real} is the real heat transfer capacity between the fin surface and air; Q_{ideal} is the ideal heat transfer capacity, which is calculated at the assumption of fin surface temperature equal to the tube surface temperature.

To validate the reliability of the self-developed numerical simulation code, numerical simulations were carried out at the same operating conditions and fin geometrical configurations as Xin et al. (1994). The comparisons of simulated average heat transfer coefficient and friction factor with the values calculated by experimental correlations (Wang et al., 2002; Xin et al., 1994) are shown in Figures 5 and 6 under different Reynolds numbers ranging from 500 to 4000 (the corresponding frontal velocity ranges from 0.46 m/s to 3.71 m/s). As can be seen from the figures, the heat transfer coefficient increases and the friction factor decreases with the increase of Reynolds number. Compared with the Wang et al.'s experimental results (2002), the mean deviation of Nu number is 5.3%, and the maximum deviation is 8.6%. The mean deviation of f between simulation results and experimental correlation (Xin et al., 1994) is 1.9%; the max deviation is 7.5%.

Figure 5 Comparison of Nusselt number between prediction and test results

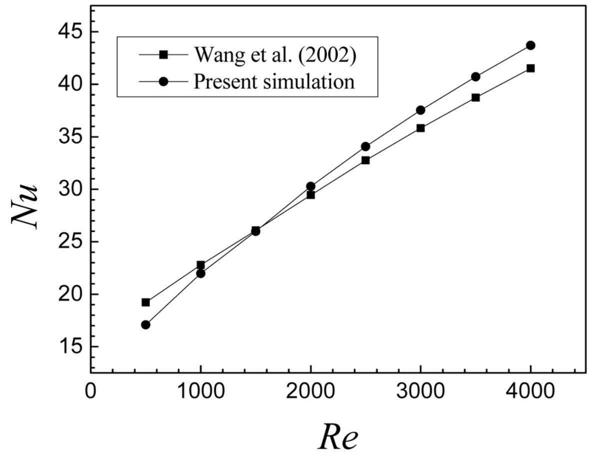
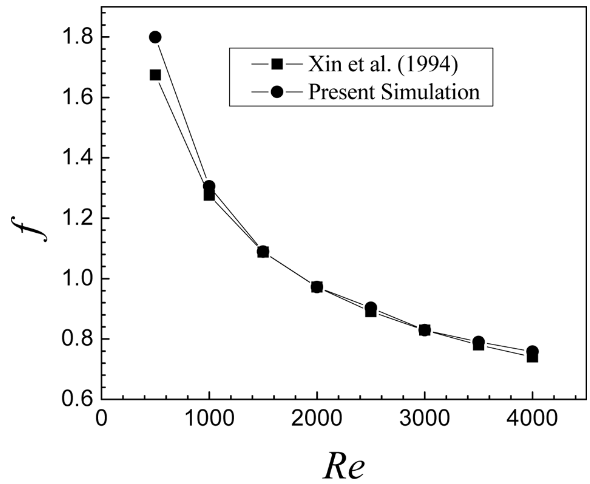


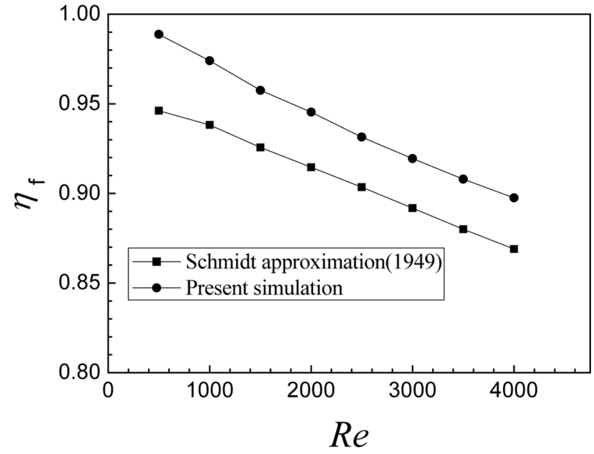
Figure 6 Comparison of friction factor between prediction and test results



The simulation results for the fin efficiency are compared with the values calculated by Schmidt approximation (Schmidt, 1949), as shown in Figure 7. The present simulation results are well accordant with the predicted results by Schmidt approximation approach. The mean deviation between them is lower than 4.0%, which demonstrates the reliability of the present simulation results.

In the following sections, the self-developed simulation code will be used to predict the effects of parameters on the heat transfer and fluid flow performance and fin efficiency of wavy fin-and-tube heat exchanger. To clearly examine the influencing effect of a certain parameter, only the parameter is variable in each section and the other parameters are set at constant values as shown in Table 1. Except investigating the effect of Reynolds number, the Reynolds number is set at 1000 in other sections.

Figure 7 Comparison result of fin efficiency



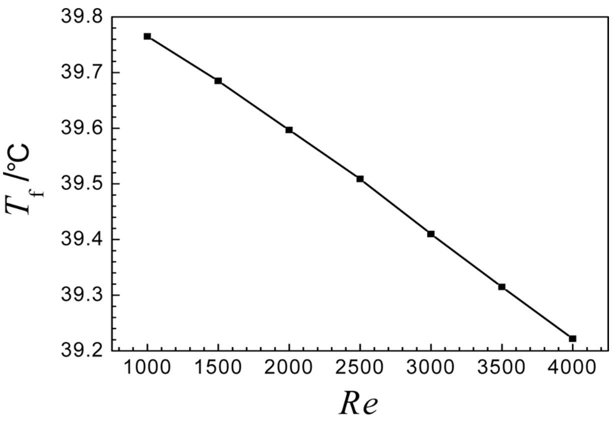
4 Simulation results and discussions

4.1 *Effects of Reynolds number*

The effects of Reynolds number on Nusselt number and friction factor are shown in Figures 5 and 6. It can be seen that with the increase of Reynolds number, heat transfer coefficient of the wavy fin-and-tube heat exchanger increase and the increasing tendency decreases gradually. The friction factor decreases with the increase of Reynolds number and the decreasing tendency becomes weaker and weaker.

The effect of Re on fin efficiency is shown in Figure 7. With the increase of Re, heat transfer performance between air and fins is enhanced. At the same time, air flow rate increases and the air temperature rise decreases. So, the real average temperature on fin surface is more and more lower than the temperature on tube surface as shown in Figure 8, which leads to the fin efficiency decreases with the increase of Re.

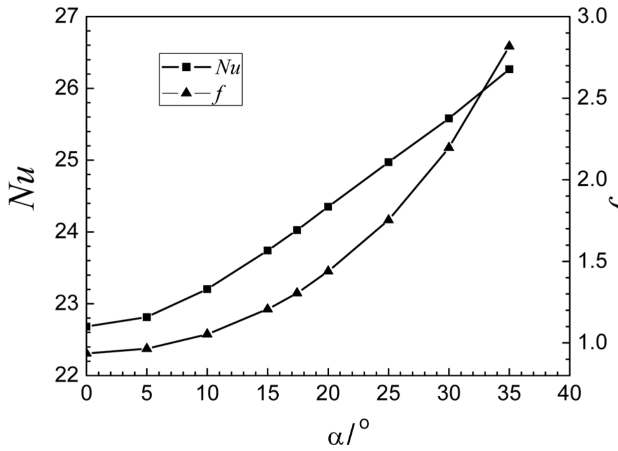
Figure 8 Effect of Re on fin surface average temperature



4.2 *Effects of wavy angle*

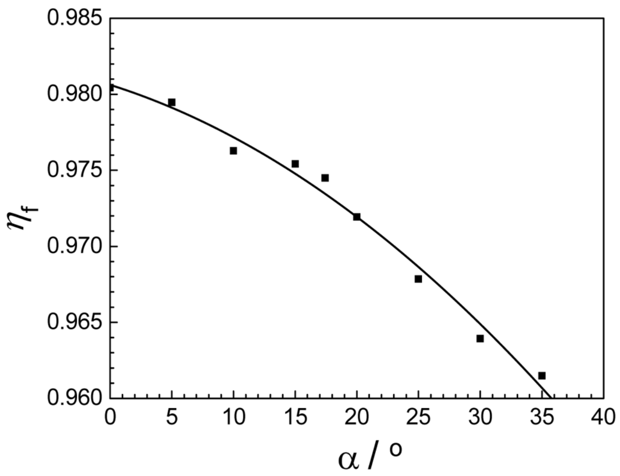
The effects of wavy angle on heat transfer and fluid flow characteristics are shown in Figure 9. With the increase of wavy angle, the disturbance for the air flow in the wavy channel is enhanced that leads to the increase of Nu, and the increasing tendency is more and more obvious. Friction factor increases with the increase of wavy angle, which is similar to the variation of Nu. Comparing the variation tends of Nu and *f*, it can be seen that the increasing percentage of *f* is higher than the increasing percentage of Nu, when wavy angle is larger than 20°. So, in practical application, the wavy angle had better be located between 10°~20°.

Figure 9 Effects of wavy angle on Nu and *f*



The effects of wavy angle on fin efficiency are presented in Figure 10. With the increase of the wavy angle, the disturbance of wave crest and wave trough to the air flow in the wavy channel increases, which leads to the heat transfer performance between air and fin surface enhanced. The real average temperature on the fin surface decreases with the enhancement of the heat transfer performance. So, the fin efficiency decreases with the increase of wavy angle.

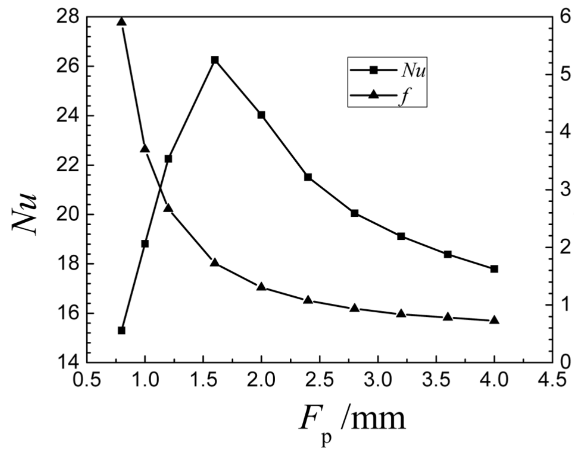
Figure 10 Effect of wavy angle on fin efficiency



4.3 *Effects of fin pitch*

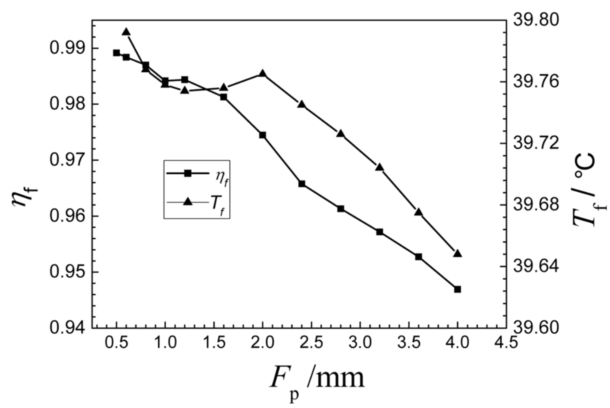
The effects of fin pitch on convective heat transfer performance are shown in Figure 11. At first, Nu increases with the increase of the fin pitch until it reaches a maximum value, and then Nu will decrease with the more increase of fin pitch. There exists an optimised fin pitch for Nu. At the smaller fin pitch, *f* decreases dramatically and the decreasing tendency slows down gradually. Comparing the variation tends of Nu and *f*, it can be seen that although the heat transfer performance can be enhanced by decreasing the fin pitch, the augmentation degree of *f* is more significant than that of Nu. So, in practical applications, the heat transfer and fluid flow performance should be considered synthetically. The fin pitch located between 1.2 mm and 2.0 mm is recommended in this paper.

Figure 11 Effects of fin pitch on Nu and *f*



The variations of fin efficiency and fin surface average temperature with fin pitch are presented in Figure 12. With the increase of fin pitch, the fin efficiency decreases, but the variation degree is very limited, especially in the small fin pitch. It can also be seen that with the increase of fin pitch, the average temperature decreases until it reaches a minimum value, the corresponding fin pitch lies between 1.2 mm and 1.6 mm, where the Nu gets the maximum value. Due to the best heat transfer performance, the fin surface average temperature is lower than that at the nearby fin pitch conditions. Then, with the further increase of fin pitch, the convective heat transfer performance deteriorates and the temperature on the fin surface increases slightly. At the same time, the air flux in the wavy channel increases quickly and the temperature rise of the air reduces. So, the average temperature on the fin surface and the fin efficiency decrease with the further increase of fin pitch.

Figure 12 Effects of fin pitch on fin efficiency and fin surface average temperature



4.4 Effects of fin thickness

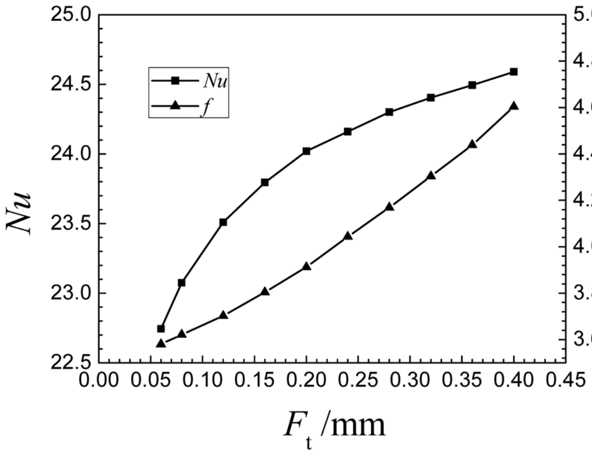
The minimum air flow cross-section decreases with the increase of fin thickness, which leads to the air velocity at the minimum air flow cross-section increasing under the same frontal velocity. So, to analyse the effect of fin thickness on heat transfer, fluid flow and fin efficiency under a certain frontal velocity, in the following section, the friction factor is defined as

$$f = \Delta p \cdot D \big/ [(1/2)\rho U^2 L] \tag{17}$$

where U is the frontal velocity.

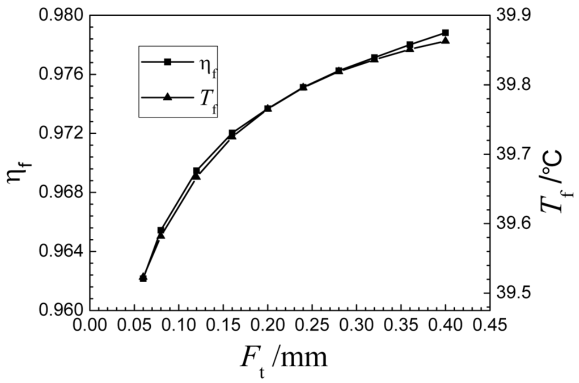
The effects of fin thickness on Nu and f are presented in Figure 13. With the increase in fin thickness, the air flow area in the wavy channel becomes narrow and the air velocity in the channel increases. So, Nu increases with the increase of fin thickness, but the increasing tendency gradually decreases. The variation of Nu with fin thickness is similar to that of Nu with Re, which indicates that the increase of Nu with the increase of fin thickness is mainly caused by the velocity augmentation. The pressure drop is the function of the velocity square. With the increase of the air velocity in the channel, the pressure drop is augmented. So, the friction factor increases quickly with the fin thickness and the augmentation tendency is more and more obvious. The fin thickness located between 0.1 mm and 0.12 mm is recommended in this paper.

Figure 13 Effects of fin thickness on Nu and f



The effects of fin thickness on fin efficiency and fin surface average temperature are shown in Figure 14. With the increase of fin thickness, the heat conduction capacity from the tube to the fin is enhanced and the heat capacity in the fins is augmented. So, although the heat transfer performance between the fins and air is enhanced with the increase of fin thickness, the average temperature on the fin surface is more and more close to the tube surface temperature, which leads to the fin efficiency increases gradually.

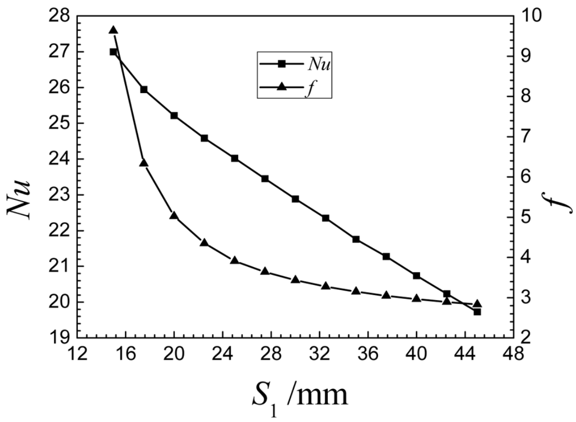
Figure 14 Effects of fin thickness on fin efficiency and fin surface average temperature



4.5 Effects of transverse tube pitch

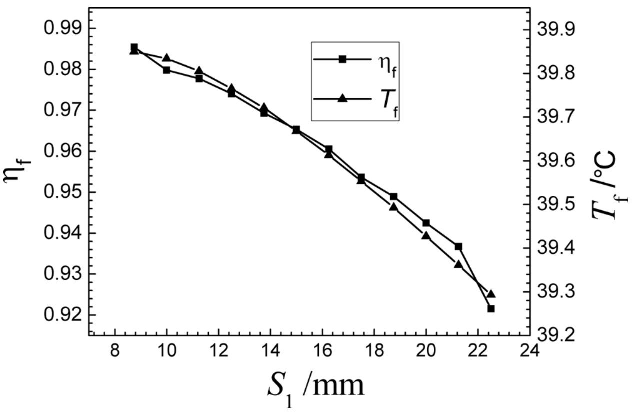
In this section, the friction factor is defined with the frontal air velocity as shown in equation (17). Figure 15 shows the variations of Nu and f with the transverse tube pitch. The disturbance of the tube to the air flow is weakened with the increase of transverse tube pitch. So, the heat transfer performance deteriorates and friction factor decreases. When the transverse tube pitch reaches a certain value, the tube has little effect on the air flow. The flow almost keeps the same flow pattern as in the wavy channel and the friction factor keeps a constant.

Figure 15 Effects of transverse tube pitch on Nu and f



The variations of the fin efficiency and average fin surface temperature with the transverse tube pitch are presented in Figure 16. The distance between the two tubes increases with the increase of the transverse tube pitch. The distance between two constant temperatures heat source increases, which leads to the heat conduction distance in the fins increases. Along the heat conduction direction, there always exists temperature difference. The temperature decreases gradually with the locations far away from the tubes at the same thermal conductivity. So, the average temperature on the fin surface decreases with the increase of the transverse tube pitch, which leads to the fin efficiency decreased quickly.

Figure 16 Effects of transverse tube pitch on fin efficiency and fin surface average temperature



5 Conclusions

In this paper, 3D numerical studies on the heat transfer and fluid flow performance of the wavy fin-and-tube heat exchangers were performed with BFC method. The fin thickness and the heat conduction in the fins are considered. The effects of Reynolds number, wavy angle, fin pitch, fin thickness and transverse tube pitch on the heat transfer and fluid flow performance and fin efficiency were investigated. The following conclusions can be derived:

- Nu increases and *f* decreases with the increase of Re. The fin efficiency and average temperature on the fin surface decrease with the increase of Re.
- With the increase of wavy angle, the disturbance to the air flow in the wavy channel is enhanced, which leads to increase of both Nu and *f*. And the increasing degree of *f* is larger than that of Nu. In practical applications, the wavy angle located between 10°~20° is recommended. The fin surface average temperature and fin efficiency decrease with the increase of wavy angle.
- Nu increases with the augmentation of fin pitch until it reaches a maximum value, and then Nu will decrease with the more increase of fin pitch. *f* decreases dramatically with the augmentation of fin pitch. In this paper, the recommendatory fin pitch is between 1.2 mm and 2.0 mm. With the augmentation of fin pitch, the air flux in the wavy channel increases and the air temperature rise decreases, which leads to the reductions of the fin surface temperature and fin efficiency.
- With increase in fin thickness, the air velocity in the channel is enhanced, which results in heat transfer and pressure drop along the increase of flow direction. The fin thickness located between 0.1 mm and 0.12 mm is recommended. With the increase of fin thickness, the average temperature on the fin surface is more and more close to the temperature of the tube, which leads to the increase of fin efficiency gradually.
- The disturbance of the tube to the air flow is weakened with the increase of transverse tube pitch. So, the heat transfer performance deteriorates and friction factor decreases. The distance between the two tubes increases with the increase of the transverse tube pitch. The temperature decreases gradually with the locations far away from the tubes. So, the average temperature on the fin surface decreases with the increase of the transverse tube pitch, which leads to the decrease of fin efficiency quickly.

Acknowledgements

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Nomenclature

C_p	Specific heat, J/kg K
D	Tube outside diameter, m
f	Friction factor
F_p	Fin pitch, m
F_t	Fin thickness, m
h	Heat transfer coefficient, W/(m^2 K)
k	Thermal conductivity, W/(m K)
L	Fin length along flow direction, m
Nu	Average Nusselt number, $(h \cdot D)/k$
Δp	Pressure drop in flow direction, Pa
Q_{real}	Real heat transfer capacity, W
Q_{ideal}	Ideal heat transfer capacity, W
Re	Reynolds number, $(U_c D/\upsilon)$
S_1	Transverse tube pitch, m
T	Temperature, K
u	Velocity components in coordinates, m/s
U	Transformed velocity, m/s
x, y, z	Cartesian coordinates
Greek symbols	
α	Wavy angle, °
ξ, η, ζ	Body-Fitted Coordinates
η_f	Fin efficiency
ρ	Density, kg/m^3
υ	Kinematic viscosity, m^2/s
Subscripts	
f	Fin surface
in	Inlet parameters
out	Outlet parameters
w	Tube surface conditions