THREE-DIMENSIONAL ASSESSMENT OF THERMAL-HYDRAULIC BEHAVIOUR IN HEAT EXCHANGERS FITTED BY WAVY ANNULAR FINS

by

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In this study, numerical studies to clarify the influence of wave number and amplitude on thermal-flow behavior of wavy annular finned-and-tube heat exchangers are described. For a range of Reynolds number from 4400 to 14300, the influence of wave amplitude, $1.5 \leq A \leq 4.5$ mm, and wave numbers, $2 \leq N_w \leq 6$, on forced convection heat transfer was examined. It was revealed that the wave amplitude and number have an impact on the heat flux, Colburn factor, and friction factor. The wavy annular-fins with a 3 mm amplitude and $N_w = 4$ waves obtained the highest values at all Reynolds numbers in terms of the overall performance criterion ($j/f^{1/3}$).

Key words: *heat exchanger, fin, Colburn factor, heat transfer, assessment*

Introduction

Annular finned-tube heat exchangers (FTHE) are widely used in heat generation systems because of their straightforward design, reduced cost, lighter weight, and high efficiency. However, in this fin shape, at least seven characteristic regions in flow physics and heat transfer intensity take place over the fin surface. Poor heat exchange can be found in some of these areas (the wake region behind the tubes) [1, 2]. So, reducing the effect of these dead regions by modifying the fin profiles is the most effective way in searching for more efficient heat exchangers (HE).

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Researchers have suggested a number of passive methods to significantly increase the energy efficiency of annular finned-tube HE. The circular tube with serrated and star-shaped fins HE is preferred in various industrial applications due to its light weight, low cost, compactness and high energy performance. Mansour *et al.* [3] conducted the 3-D numerical simulations to study the effect of the number of fin segments, $4 \leq N_s \leq 14$, and twisting angles, $0° \leq \beta \leq 25°$, on the efficiency of serrated shaped fins placed in the centre of circular tube. Recently, Keawkamrop *et al.* [4] studied the effect of serrated welded spiral fin dimensions on energy efficiency of HE. Another technique is the periodic interruption of the growth of both thermal and dynamic boundary-layers over the annular fin surface by the use of star-shaped type fins. Only Bošnjaković *et al.* [5-7] performed experimental and numerical analysis to determine the air-side performance of this new fin with staggered tube layouts.

Perforated fins have been proposed by some researchers as a way to lessen the wake zone behind tubes, where the secondary flow is produced. Due to this, weight of HE is reduced, thermal exchange rate is enhanced, pumping power is decreased and hence greater energy efficiency is achieved. In these studies, Liu *et al.* [8]; and Rauber *et al.* [9], the researchers show the importance of perforations in improving the performance of plate finned-tube HE for frosting and no-frosting conditions. An experimental study by Lee *et al.* [10] investigated the effect of perforated circular holes on thermal-flow characteristics of circular finned tube air-cooled HE. They determined the heat transfer coefficient and the pressure losses for smooth circular fin, 2-hole and 4-hole perforated fin. Banerjee *et al.* [11] compared the heat transfer and flow physics having four row annular finned-tubes of air-cooled steam condensers with different perforation location. For spiral-type fins, Lee *et al.* [12] confirmed the ability of circular perforations to improve the hydrothermal performance of HE with less material consumption. Zaidan *et al.* [13] examined and compared the heat transfer and flow properties of a one-row annular fin-and flat-tube with three different types of perforations. Recently, Bošnjaković and Muhič [14] combined the passive techniques of star-shaped fins and perforated fins to further improve the efficiency of annular fins. Some researchers show that the eccentric finned-tube technique significantly improves the HE performance under the same total thermal exchange surface area. In Benmachiche *et al.* [15]; and Tahrour *et al.* [16], the authors examined the effect of this new idea on heat transfer and fluid flows for both 1-row and 4-row staggered bank of circular fin. For laminar flow natural convection, Senapati *et al.* [17] carried out a 3-D numerical investigation to study the impact of eccentric annular finned horizontal cylinder on temperature plume and flow field of HE.

A large number of researchers indicate that the use of wavy-shaped fins instead of straight-fins enhances the compactness of HE, convective heat transfer intensity and the flow mixing generated by the spanwise and streamwise vortices. The first research work that applied this fin shape to annular fin geometry was accomplished by Rath and Dash [18]. Kumar *et al.* [19] performed 3-D CFD simulations to compare the thermal-flow characteristics of HE with various plate and annular fin patterns. Morales-Fuentes and Loredo-Saenz [20] performed a series of simulations using the realizable turbulence model of ANSYS Fluent to ascertain the energy efficiency of circular tubes with square fin, circular fin, and pin fin. Recently, Tahrour *et al.* [21] examined forced convection heat transfer for 3-row staggered bank equipped with five types of independent fins (concentric fins, eccentric fins, perforated fins, serrated fins, and star-shaped fins) and the fin spacing varies between 2 mm and 7 mm.

According to the overview of the aforementioned literature, the change in annular fin design has a significant effect on HE performance. To the best of our knowledge, only Rath and Dash [18] have analyzed the natural convection heat transfer in HE with wavy annular fins. There are no researching works regarding the impact of wavy fin pattern on forced convection heat transfer in HE with annular fins. Hence, the purpose here is to study the influence of wavelength and/or amplitude of sinusoidal wavy annular-fins on air-side performance of 1-row bundles using a 3-D numerical simulation. Three factors were examined in order to assess the air-side performance: the heat flux, the Colburn factor, and the friction factor. For the first time, we examined how amplitude and wave number factors of sinusoidal wavy annular-fins affected the performance evaluation criterion P_{EC} of HE. The numerical results have been obtained for a considerable interval of wavelength and amplitude for a range of $4400 \leq \mathrm{Re} \leq 14300$.

Model descriptions

Figure 1 depicts the planned wavy annular fin-and-circular tube HE arrangement. Hot fluid circulates inside the tubes as cold air travels between the undulating annular fins. Aluminum was used to make the solid body, including the annular fins, and it had a thermal conductivity of k = 202.4 W/mK. Table 1 provides a summary of the values for the fundamental characteristics of the current wavy annular fins. Various fin amplitudes and wave numbers (wavy length) were tested in this experiment. The intake region is stretched 1.2 times the fin diameter D_f to achieve a uniform velocity at the entrance. The 3.6 times the diameter of the fins are added to the outflow zone, however, to prevent the backflow effect.

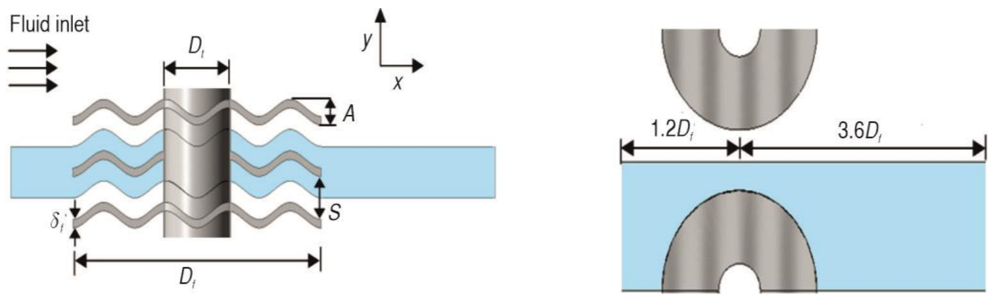

Figure 1. The researched wavy annular finned-tube HE geometric properties

Table 1. Geometrical details

Name	Range Studied
Wave amplitude, A	1.5-4.5 mm
Wave number, N_w	2, 3, 4, 5, 6
Fin diameter, D_f	99 mm
Fin spacing, S	5 mm
Fin thickness, δ_f	1 mm
Tube diameter, D_t	27 mm

Mathematical modelling

The Reynolds number used in this study, which is dependent on the outside diameter of the tube collar, $D_c = D_t + 2\delta_f$, and the maximum speed at the smallest free flow cross-section $u_{\max}$ ranges from 4400 to 14300. This results in the airflow being incompressible, turbulent, steady-state, and 3-D [19-22]. Based on previous research for annular fins [15, 16, 19-22], the RNG k-ε model of ANSYS 18.2 are adopted in this study. The following equations were used to explain the behavior of fluid flow:

Continuity equation:

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

Momentum equation:

$$\frac{\partial}{\partial x_j}\left[\rho u_i u_j + p\delta_{ij} - \mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3}\delta_{ij}\frac{\partial u_l}{\partial x_l}\right) + \rho_a \overline{u_i^{'} u_j^{'}}\right] = 0 \tag{2}$$

where

$$-\rho_a \overline{u_i^{'} u_j^{'}} = \mu_t\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \frac{2}{3}\left(\rho_a k + \mu_t \frac{\partial u_i}{\partial x_i}\right)\delta_{ij} \tag{3}$$

Energy equation:

$$\frac{\partial\left[u_i(\rho_a E + p)\right]}{\partial x_i} = \frac{\partial}{\partial x_i}\left[(k_a + k_t)\frac{\partial T}{\partial x_i}\right] \tag{4}$$

where k_t denotes the thermal conductivity of the turbulent flow and E denotes the total energy.

Numerical modelling

Adequate boundary conditions must be used to solve the equations. As shown in fig. 2, a constant temperature of 353 K is assigned to the tube surface and all of the velocity components are set to zero. Moreover, dry air at a temperature of 288 K enters the computational domain with a uniform velocity, $2 \le V_{\text{in}} \le 6.5$ m/s. The symmetrical boundary is set on the side walls whereas the periodical boundary is defined on the lower and upper wavy annular surfaces to minimize the computing load, and to speed up the computation time.

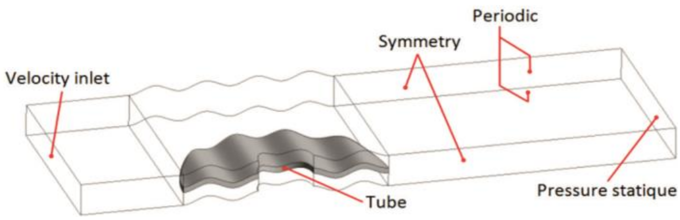

Figure 2. The 3-D Computational domain with its prescribed boundary conditions

In the present work, we have used the ANSYS WORKBENCH 18.2 meshing tool to divide the computational domain into a finite number of cells characterized by finer size edges in regions where the gradients are high such as surrounding tubes as well as wavy annular-fins, as reported in fig. 3. To determine adequate grid density, we have calculated the Colburn factor, j, and the friction factor, f, for A = 4.5 mm and N_w = 6. The N_w waves with various grid densities for the case of Re =

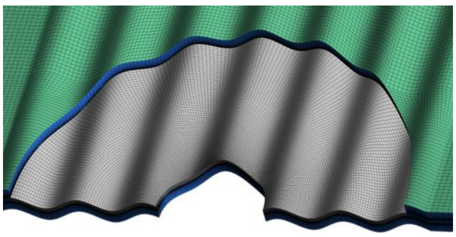

Figure 3. Generated mesh in the present simulations

14300. The findings indicate that, the mesh of 1.6 million hexahedral cells is the best to keep a balanced trade-off between convergence time and solution accuracy.

Factors used to assess air-side performance are: the heat flux, φ, the Colburn factor, j, the friction factor, f, and the performance evaluation criterion, P_{EC}, [21].

Numerical results and analysis

The present study focuses on the impact of wavelength and/or amplitude of wavy annular-fins on air-side performance of one-row HE. For a fixed fin spacing of $S = 5$ mm, the range of wave amplitude, $1.5 \leq A \leq 4.5$ mm, and wave number of, $2 \leq N_w \leq 6$, are chosen to be studied herein. In order to evaluate the accuracy of our numerical model, we compared in our previous papers [15, 16] the results obtained for the case of circular plate fin with the experimental data of Chen *et al.* [2].

At the higher Reynolds number value, fig. 4 exhibits the temperature distribution of the computational domain for the cases of various situations, *i.e.*, $A = 1.5$ with $N_w = 2$ and $A = 4.5$ with $N_w = 6$. As shown, a great temperature gradient along with the longitudinal flow direction of wavy annular-fin with $A = 4.5$ mm and $N_w = 6$ waves compared with the case of $A = 1.5$ mm and $N_w = 2$ waves. It means that a useful heat exchange between the finned-tube surfaces and air-flow required high values of amplitude and number of waves. This result can be explained by the fact that increasing the waviness and the amplitude of wavy surfaces enhances the fluid mixing, decreasing the wake region behind the tubes, and increases the total surface area of HE (wavy annular finned-tube with $A = 4.5$ mm can have up to 7% higher surface area compared to the case of fin with $A = 1.5$ mm.

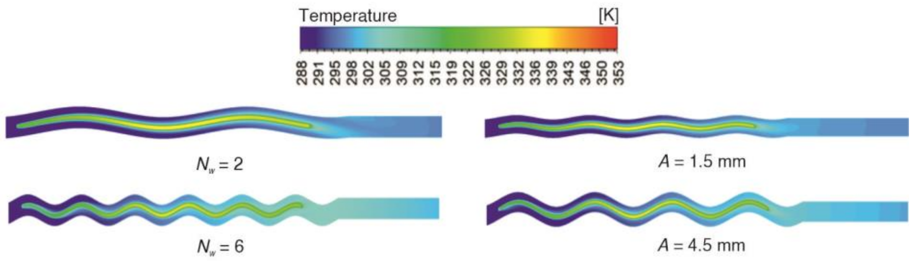

Figure 4. Temperature contours in wavy annular fin for two situations

This second part of the results focuses on the effect of wave amplitude, $1.5 \leq A \leq 4.5$ mm, on thermal-flow characteristics of HE for $N_w = 4$ mm waves, $S = 5$ mm, and a range of $4400 \leq \mathrm{Re} \leq 14300$. The heat flux values of all wave amplitudes are compared and presented in fig. 5. Obviously, the amplitude, A, significantly affects the convective heat transfer. The heat flux, φ, was enhanced with Reynolds number due to the higher heat transfer intensity at high velocity. Due to the useful increase in heat exchange surface area with the amplitude of wavy annular-fin, the figure shows that the heat flux increase by 13.6% (at Re = 4400), 12.8% (at Re = 7700), and 13.4% (at Re = 14300) when A changed from 1.5 to 4.5 mm. Similarly, Wen *et al*. [23] demonstrated that, for sinusoidal wavy channel of HE, the heat flux enhanced by 33.74% when compared to the straight surface.

Figure 6 shows the comparison of the Colburn and friction factors for various amplitude values and Reynolds number. The values of j fell by 74-76.3% as the Reynolds number-value increased from 4400 to 14300, however the f factor decreased by 15.2-32.6%. Regardless of Reynolds number, wavy annular-fin with $A = 4$ mm generates the greatest amount of j, which is calculated to be 8.15% more for Re = 4400 and 7.3% higher for Re = 14300 than for

A = 1.5 mm. Additionally, these same plots show that f increases by 31.2-36.9% between A = 1.5 and 4.5 mm. The increased drag force and the size of the recirculating cells with wavy annular-fins are used to justify the increased f.

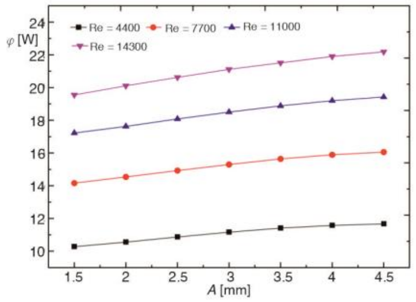

Figure 5. Wave amplitude's effect on thermal flux

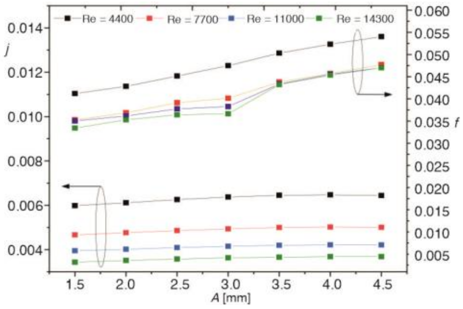

Figure 6. Impact of wave amplitude on Colburn and friction factors

The aim of each thermal device design is to give higher thermal transfer intensity while minimizing pressure losses. Therefore, the performance evaluation criterion P_{EC} is used here to evaluate the overall performance of wavy annular finned-and-tube HE. For N_w = 4 waves, fig. 7 provides the variation in P_{EC} *vs*. wave amplitude at all values of Reynolds number. Whatever the value of A, P_{EC} decreases along with increasing of Reynolds number due to the rapid decrease in the Colburn factor with Reynolds number, as previously shown in fig. 6. Over the whole range of Reynolds number, wavy annular-fins with A = 3 mm yields the most significant amount of $j/f^{1/3}$, which is estimated to be 3.5% and 6.4% higher than that for A = 1.5 mm and 4.5 mm, respectively, at Re = 11000. Finally, when considering the thermal exchange surface and pressure losses, annular fins with A = 3 mm provides the maximum thermal-flow efficiency.

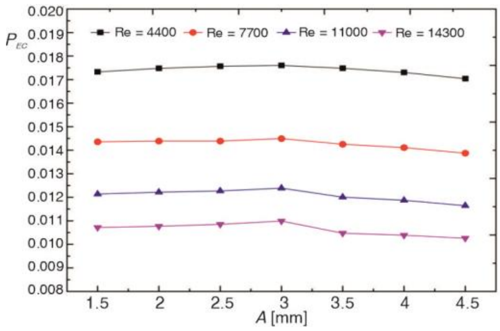

Figure 7. Wave amplitude's effect on performance evaluation criteria

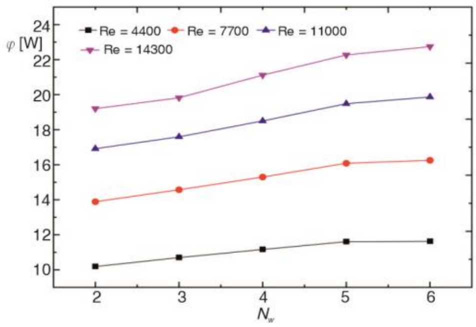

Figure 8. Wave number's effect on thermal flux

The optimization of HE with sinusoidal wavy annular-fins also requires knowledge of the optimum wavelength to obtain the maximum heat transfer capacity with minimum pressure losses. A series of simulations have been carried out on the fin with the best wave amplitude, A = 3 mm, and for the number of waves ranging from two to six waves. Figure 8 presents the distribution of thermal flux, φ, for different, N_w, values. The value of φ for wavy annular-fins with N_w = 6 is more significant than those of the other cases whatever the Reynolds number. The heat flux increases by 14.1% (at Re = 4400) and 18.4% (at Re = 14300) with N_w increasing from 2 to 6 waves. This is because the waviness significantly lengthens the heat exchange surface area and improves the turbulent intensity. Similarly, Cheng *et al.* [24] observed that wavy fins with 4-waves own twice Nusselt number of that of 1-wave, while the heat exchanger with 2-waves only owns about 20% higher than single-wave.

Figure 9 presents the comparison of j and f factors for different N_w values and for a range of $4400 \leq \mathrm{Re} \leq 14300$. As expected, j increases with increasing of N_w from 2 to 6 waves regardless the value of Reynolds number. Precisely, the j factor provided by a fin with $N_w = 6$ is the most significant, which is higher by about 2.5-3.8% and 10-11.9% than those of fins with $N_w = 4$ and $N_w = 2$, respectively. At higher value of N_w, the thermal boundary-layer is broken at each wave crest, producing more considerable flow mixing and thermal enhancement. Also, fig. 9 shows the increase in friction factor with N_w for the whole range of Reynolds number. For example, a wavy fin with $N_w = 6$ provides 22.1-58.2% and 41.6-86.4% higher friction factor than those for a fin with $N_w = 4$ and $N_w = 2$, respectively.

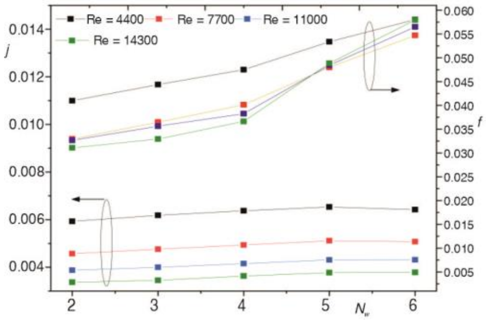

Figure 9. Impact of wave number on Colburn and friction factors

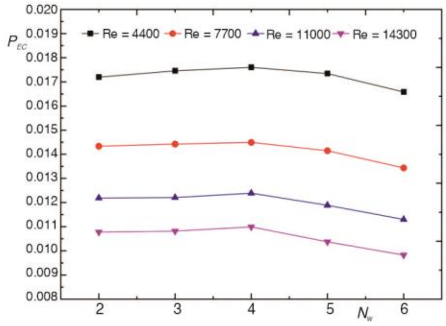

Figure 10. Wave number's effect on performance evaluation criteria

To choose the best waviness of annular-fins, the performance evaluation criterion P_{EC} in HE with different wave number and at $A = 3$ mm is shown in fig. 10. As can be seen in the figure, P_{EC} first increases with increasing of N_w from 2 to 4 waves, and then decreases when N_w continues increasing from 4 to 6 waves. The reason for this phenomenon is the dominance of helical flow patterns over cross-flow. In addition, annular fins with four waves provide 1.1-2.9% and 6.2-11.9% higher P_{EC} than those for the fin with two and six waves, respectively. The same results are reported in the research work of Wen *et al*. [23]. In conclusion, the best performance evaluation criterion of HE heat exchangers is obtained in case of sinusoidal annular fin-and-tube with $A = 3$ mm and $N_w = 4$ waves. Future research could examine the performance of the heat exchanger with fins of different shapes, such as V-barriers [25], and compare theme to the performance of the existing fins.

Conclusion

This numerical study aimed to investigate the influence of wave amplitude and number on air-flow and heat transfer behavior in wavy annular finned-and-tube HE. For the first time, the impact of wavy annular-fin dimensions with circular tubes on turbulent thermal-flow characteristics of HE is presented and discussed in details. The main findings are listed as follows.

- The heat flux and friction factor, f, increase according to the wave amplitude. However, for all Reynolds number, wavy annular-fin with $A = 4$ mm yields the most significant amount of j.

- Increases in heat flux (14.1-18.4%), in friction factor (41.6-86.4%), and in the Colburn factor (10-11.9%) were observed with an increase in N_w from two to six waves.
- The results show that with augmentation Reynolds number, both friction, f, and Colburn, j, factors decrease whereas the heat flux increases.

In terms of the performance evaluation criterion P_{EC}, wavy annular-fins with wave amplitude of A = 3 mm and N_w = 4 reach the greatest P_{EC} values. Thus, when the thermal exchange surface and the pressure losses are relevant, a wavy annular-fin with A = 3 mm and N_w = 4 is recommended in the design of wavy annular finned-and-tube HE.

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