Heat Transfer Characteristics and Energy Efficiency Analysis of Finned Tube Heat Exchangers

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Abstract. Flow and heat transfer characteristics of the integral finned tube heat exchanger are studied by the numerical simulation method in the present paper. The energy efficiency levels of two types of finned tube heat exchangers (aligned finned tube and staggered finned tube) are evaluated by PEC method. The heat transfer intensity of different finned tube heat exchangers was evaluated by the field synergy principle. When the tube spacing ratio ($S_1/S_2 = 1$) is small, the field synergy of staggered finned tube is similar to the aligned finned tube, and the PEC index is less than 1; when the spacing ratio ($S_1/S_2 = 3$) is large, the field synergy of the staggered finned tube get much stronger than the aligned finned tube, and the PEC index is larger than 1 as the $Re \geq 8 \times 10^4$. Considering both the change of heat transfer intensity and the increase of resistance, it is recommended to evaluate the energy efficiency level of finned tube heat exchanger by PEC method.

1. Introduction
Heat exchangers are one of the common equipment for various industrial processes. Heat transfer performance research and energy efficiency evaluation are of great significance for the efficient use of heat exchangers. Heat exchanger research includes heat transfer enhancement, control and simulation and so on. In order to study the heat transfer enhancement of heat exchangers, scholars have proposed methods to expand heat transfer surfaces, fluid induction and surface treatment. The fin structure can effectively improve the heat exchange capacity of the pipe, and the fluid disturbance can be enhanced by the pipeline arrangement and the fin treatment, thereby further enhancing the heat transfer capability of the heat exchanger. For the heat transfer performance calculation of various fin-and-tube heat exchangers, scholars have proposed many heat transfer coefficient correlation formulas [1-4]. Zhukauskas[1] summarized a highly applicable heat transfer correlation for the heat transfer of the fluid across the tube bundle, with the Pr number ranged from 0.6 to 500.

For the study of flow heat transfer characteristics in heat exchangers, many scholars use numerical methods for analysis and discussion. Mon [5] numerical studied the effects of fin spacing on four rows of annular staggered finned tube bundles by three-dimensional numerical studies. Kahalerras [6] used numerical simulation methods to improve the heat transfer performance of the double-tube heat exchanger by using porous fins. He [7] conducted a three-dimensional numerical simulation study on the laminar heat transfer and flow characteristics of plate-fin heat exchangers. From the field synergy principle, it is analyzed that reducing the angle between velocity and temperature gradient is the basic principle of enhancing convective heat transfer. Suárez [8] proposed a novel one-dimensional model to predict the heat transfer rate of a continuous finned tube heat exchanger. Ephraim [9] carried out three-dimensional numerical simulation of four rows of elliptical finned tube heat exchangers by LES numerical simulation method, and analyzed the flow evolution in the whole heat exchanger and the...
instability of complex vortex structures. The results show that as the Reynolds number increases, flow instability occurs faster downstream of the heat exchanger. Zhang [10] conducted a CFD numerical simulation study on the performance of the finned tube heat exchanger in the air cooler, and predicted the air flow, temperature curve and heat transfer characteristics under different operating conditions.

The heat transfer research of finned tube heat exchangers is mainly focused on the study of heat transfer enhancement. Guo [11] first proposed the field synergy principle and carried out numerical verification. The field synergy number, $F_s$, is defined as the degree of synergy between the velocity field and the temperature gradient field over the entire region. The results show that the field synergy will be much less than 1 under most heat transfer conditions, and there is a greater potential for heat transfer enhancement. Webb [12] divides heat transfer enhancement into three purposes, reducing surface area, increasing heat load and reducing power consumption, and proposed PEC method to evaluate the effect of enhanced heat transfer structure on heat transfer.

Many scholars have obtained empirical correlations for convective heat transfer of fin-and-tube heat exchangers through experimental methods, numerical simulations, theoretical analysis, etc. The research on the enhanced heat transfer of fin-and-tube heat exchangers is also sufficient. While for the evaluation of the energy efficiency level of the heat exchanger is insufficient. In this paper, the field synergy theory and PEC method are used to evaluate the enhanced heat transfer effect and energy efficiency level of the finned tube heat exchanger in the staggered finned tube and the aligned finned tube. It is recommended to evaluate the energy efficiency level of the finned tube heat exchanger by PEC method. The field synergy principle could be used to evaluate the heat transfer intensity of the finned tube heat exchanger.

2. Physical model and numerical method

2.1 Physical model

In this paper, flow and heat transfer characteristics of the staggered finned tube and the aligned finned tube with different tube pitch was calculated by the numerical method. Finned tube bundle contains 16 rows of tubes. A two-dimensional assumption is made to the finned tube heat exchanger, ignoring the flow and heat transfer along the length of the tube. The working fluid is air, assuming the tube bundle is at a constant temperature of 80°C and the inlet air temperature is 25°C. The inlet boundary adopts the velocity boundary condition, the outlet adopts the pressure outlet boundary, and the other boundary adopt the periodic boundary condition. The Reynolds number is determined by the average flow velocity at the smallest section of the bundle, and the feature length is the outer diameter of the tube. The Reynolds number ranges from $10^4$-8×$10^4$. The tube spacing is divided into two types, $S_1=1\times S_2=1.5\times D$, and $S_1=3\times S_2=4.5\times D$.

![Diagram of tube bundle model](image)

Figure 1 Tube bundle model

2.2 Numerical method

The computational study was conducted using the Ansys Fluent 13.0 [13]. The flow and heat transfer was calculated by the standard k-ε turbulence model. The continuity, momentum and energy equation are as below:

Continuity:
\[ \frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} = 0 \]  

(1)

\[ \frac{\partial U^2}{\partial x} + \frac{\partial (UV)}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + 2 \frac{\partial}{\partial y}[\left( \theta + \theta_t \right) \left( \frac{\partial U}{\partial x} + \frac{\partial V}{\partial y} \right)] + \frac{\partial}{\partial x}[\left( \theta + \theta_t \right) \left(\frac{\partial V}{\partial x} + \frac{\partial U}{\partial y}\right)] \]  

(2)

\[ \frac{\partial V^2}{\partial x} + \frac{\partial (VU)}{\partial y} = -\frac{1}{\rho} \frac{\partial p}{\partial x} + 2 \frac{\partial}{\partial y}[\left( \theta + \theta_t \right) \left( \frac{\partial V}{\partial x} + \frac{\partial U}{\partial y} \right)] + \frac{\partial}{\partial x}[\left( \theta + \theta_t \right) \left(\frac{\partial U}{\partial x} + \frac{\partial V}{\partial y}\right)] \]  

(3)

\[ \frac{\partial (U h)}{\partial x} + \frac{\partial (V h)}{\partial y} = \frac{\partial}{\partial x} \left[ \left( \frac{\theta}{Pr} + \frac{\theta_t}{Pr_t} \right) \frac{\partial h}{\partial x} \right] + \frac{\partial}{\partial y} \left[ \left( \frac{\theta}{Pr} + \frac{\theta_t}{Pr_t} \right) \frac{\partial h}{\partial y} \right] \]  

(4)

\[ \frac{\partial (U k)}{\partial x} + \frac{\partial (V k)}{\partial y} = \frac{\partial}{\partial x} \left[ \left( \theta + \frac{\theta_t}{\sigma_k} \right) \frac{\partial k}{\partial x} \right] + \frac{\partial}{\partial y} \left[ \left( \theta + \frac{\theta_t}{\sigma_k} \right) \frac{\partial k}{\partial y} \right] + P_k + G_k - \rho \varepsilon \]  

(5)

\[ \frac{\partial (U \varepsilon)}{\partial x} + \frac{\partial (V \varepsilon)}{\partial y} = \frac{\partial}{\partial x} \left[ \left( \theta + \frac{\theta_t}{\sigma_k} \right) \frac{\partial \varepsilon}{\partial x} \right] + \frac{\partial}{\partial y} \left[ \left( \theta + \frac{\theta_t}{\sigma_k} \right) \frac{\partial \varepsilon}{\partial y} \right] + C_{\varepsilon 1} f_1 \frac{\varepsilon}{K} (P_k + G_k) \]  

\[ - C_{\varepsilon 2} f_2 \frac{\varepsilon^2}{K} \]  

(6)

The complete computational domain was discretized using Gambit 2.4 into unconstructed mesh. The mesh was refined around the tube where the heating commenced which significantly influences the prediction of the wall temperature. The SIMPLEC scheme was used to couple the pressure and velocity fields. The QUICK scheme was used to approximate the convection terms in the momentum equations with the UPWIND scheme used for the other transport equations. The convergence criterion for the normalized residual of each equation was set to be less than \(10^{-5}\).

### 2.3 Validation of the numerical simulation

The predicted results are compared with the Zhukauskas experimental correlation in the following figures. Figure 2 compares the Nusselt number with Re number for different tube arrangement, and the tube pitch \(S_1\) is equal to \(S_2\). The correlation is as below:

\[ N_u_f = 0.27 Re_f^{0.63} Pr_f^{0.36} \left( \frac{Pr_f}{Pr_w} \right)^{0.25} \]  

for aligned finned tube

(7)

\[ N_u_f = 0.35 \left( \frac{S_1}{S_2} \right) Re_f^{0.6} Pr_f^{0.36} \left( \frac{Pr_f}{Pr_w} \right)^{0.25}, \frac{S_1}{S_2} \leq 2 \]  

\[ N_u_f = 0.40 Re_f^{0.6} Pr_f^{0.36} \left( \frac{Pr_f}{Pr_w} \right)^{0.25}, \frac{S_1}{S_2} > 2 \]  

for staggered finned tube

(8)

![Figure 2 Nusselt number](image-url)
All the difference of the results between the numerical calculation and the Zhukauskas equation is within 15%, and most of them are less than 10%. The results show that the numerical model is accurate, and the model could be used to get the flow and heat transfer characteristics for the finned tube.

3. Results and discussion

Figure 3 shows the temperature field distribution of the two kinds of finned tube heat exchangers with the tube spacing $S_1 = S_2$ and $Re = 6 \times 10^4$. Combined with the results of the Fig. 2, it can be seen that when the pipe spacing $S_1$ is relatively small, the convective heat transfer intensity of the staggered finned tube heat exchanger is higher than that of the heat exchanger, but the change of the heat exchange intensity is not large. When the pipe spacing is small, the turbulence intensity is similar, so that the heat exchange of the two types of finned tube heat exchangers is similar.

Figure 3 temperature field for finned tube

Figure 4 shows the PEC evaluation index of the staggered finned tube heat exchanger obtained based on the evaluation of the aligned finned tube heat exchanger. The formulas of PEC and $f$ are as below:

$$PEC = \frac{Nu}{Nu_0} \left(\frac{f}{f_0}\right)^{1/3}$$

$$f = \frac{\Delta P}{2\rho u_{max}} \times \frac{D}{L}$$

As shown in the Fig. 4, the PEC index of the two kinds of heat exchanger increases with the increase of Reynolds number under two different pipe spacing forms. Except the condition of the Reynolds number equals to $8 \times 10^4$, and $S_1$ equals to three times of $S_2$, the other PEC indexes are less than 1. The results show that the energy efficiency level of the staggered finned tube heat exchanger is higher than that of the heat exchanger when the Reynolds number is large. When the Reynolds number is small or the fin tube spacing is similar, the energy efficiency level of the aligned finned tube heat exchanger is higher than that of the staggered finned tube heat exchanger.

Figure 4 PEC evaluation
Figure 5 shows the angle distribution of the velocity direction and the temperature gradient direction in the flow field region for the two kinds of heat exchanger with the Reynolds number equals to $8 \times 10^4$. The angle between the velocity field and the temperature gradient field is defined as follows:

$$\cos \theta = \frac{\vec{U} \cdot \vec{\nabla T}}{|\vec{U}| \cdot |\vec{\nabla T}|} \quad (11)$$

It can be seen from the angle distribution cloud diagram that the flow path disturbance is enhanced due to the piping arrangement of the staggered finned tube heat exchanger, and the region of the angle less than 90° is larger than that of the aligned finned tube heat exchanger. According to the field synergy principle, when the other quantities are constant, the closer the angle is to 90°, the smaller the heat transfer intensity is, and the closer to 0°, the greater the heat transfer intensity. It can be seen from the degree of synergy between the flow field and the temperature field that the heat exchange intensity of the staggered finned tube heat exchanger is enhanced relative to the aligned finned tube heat exchanger.

Figure 5 Field synergy angle

Figure 6 shows the variation of field synergy number with Reynolds number in two kinds of heat exchangers under different pipe spacing conditions. As shown in Fig.6, the field synergy number decreases with the increase of Reynolds number, which indicates that with the increase of Reynolds number, the effect of fluid impingement pipeline is reduced, and the effect of fluid flow and dissociation is enhanced.

From the perspective of field synergy, increasing the fluid velocity reduces the degree of synergy between the flow field and the temperature field. Under the two tube spacing, the field synergy number of the staggered finned tube heat exchanger is larger than that of the aligned finned tube heat exchanger, but when the tube spacing $S_1$ equals to $S_2$, the field synergy number of the two kinds of heat exchanger field is very close. When the pipe spacing is small, the resistance of the staggered finned tube is larger than that of the aligned finned tube heat exchanger. In practice application, the staggered finned tube heat exchanger is more prone to fouling and dust accumulation, so it is
recommended to prioritize use the aligned finned tube heat exchanger when the pipe spacing is small. When the pipe spacing is increased, the field synergy number of the staggered finned tube heat exchanger field is larger than that of the aligned finned tube heat exchanger, indicating that the larger the pipe spacing, the better of the field synergy and the greater heat exchange intensity. When comparing the heat transfer degree of the fin-and-tube heat exchanger, the field synergy can be used for quantitative evaluation. Under the same Reynolds number and Prandtl number, the larger the field synergy, the higher the heat transfer intensity.

4. Conclusion
In this paper, the flow and heat transfer characteristics of the aligned finned tube heat exchanger and staggered finned tube heat exchanger are numerical simulated with different tube pitch. The enhanced heat transfer level and energy efficiency of the two heat exchangers were evaluated by PEC method and field synergy principle. When considering heat transfer strength and resistance changes, it is recommended to use the PEC method for the energy efficiency on the finned tube heat exchangers. When evaluating the enhanced heat transfer level of fin-and-tube heat exchangers, it is recommended to use the field synergy number for quantitative evaluation.

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