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Structural optimization of flow and heat transfer characteristics of integral helical finned tubes based on genetic algorithm

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ABSTRACT

The integral spiral finned tube has significant advantages compared with the conventional finned tube, which makes it have broad application prospects in boiler economizers. In this study, numerical simulation methods were applied to investigate the effects of base tube diameter, fin height, fin pitch, fin tip width, fin root width, transverse tube pitch and longitudinal tube pitch on the heat transfer and flow characteristics of the integral spiral fin tube. The optimal structural parameters were analyzed through multi-objective optimization. The total heat transfer coefficient is not greatly affected by the fin pitch. With the increase of the longitudinal pitch, the total heat transfer coefficient first increases and then decreases, and the optimum is around 90mm. The optimal structural parameters were obtained while considering the flow and heat transfer characteristics that: the base tube diameter is 32 mm, the fin height is 13 mm, the fin pitch is 7 mm, the fin tip width is 2.4 mm, the fin root width is 4.2 mm, the transverse pitch is 80 mm, the longitudinal pitch is 90 mm, and the inlet flow velocity is 12 m/s.

Keywords: Integral spiral finned tub, flow characteristics, heat transfer characteristics, structural optimization, multi-objective genetic algorithm.

1. INTRODUCTION

Fin and tube heat exchangers are used for heat transfer in many industrial applications due to their enlarged surface, such as waste heat recovery devices, heating, ventilation, air conditioning, and refrigeration systems. [1] Meanwhile, in power plants, finned tube bundles are used in boiler economizers. [2] The application of finned tube bundles could reduce the volume of the heat exchanger, save metal consumption, and reduce flow resistance. The finned tubes commonly used in economizers include high-frequency welded spiral finned tubes, H-shaped finned tubes and integral spiral finned tubes. The integral spiral finned tube has better heat transfer and flow characteristics in practical applications.

The integral spiral fin tube uses rolling blades to roll thick-walled seamless steel pipe into shape at one time. It has significant advantages: high heat transfer efficiency, less dust accumulation, and long service life [3,4]. The integral spiral finned tube economizer could reduce the exhaust gas temperature by 15~25℃, greatly reduce the phenomenon of wear and leakage, and effectively improve the economy of the boiler compared with high-frequency welding spiral finned tube and H-shaped finned tube economizer.

It is meaningful to investigate the flow and heat transfer characteristics for the design and safe operation of finned tube heat exchangers. Many studies have been carried out on the flow and heat transfer characteristics of high-frequency welding spiral fin tube bundles [5] and H-shaped fin tube bundles [6-8] through experimental and numerical methods. However, Research on the integral spiral fin tube bundle is rarely reported. He et al. [9] studied the heat transfer and flow characteristics of the finned tube heat exchanger, and obtained the heat transfer Nusselt number relationship and the flow

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resistance Euler number relationship. Kawaguchi et al. [5] compared the heat transfer pressure drop characteristics of serrated fins and spiral fins, and concluded that serrated fins can improve the heat transfer characteristic and the friction factor considerably compared with spiral fins, in the case of large fin pitch. Moreover, the multi-objective optimization genetic algorithm was applied to optimize the structural parameters of the heat exchanger. [10-11] The optimized flow and heat transfer characteristics were obtained. It is feasible and credible to optimize the structural parameters of the integral spiral finned tube through the combination of numerical simulation and genetic algorithm.

In this study, numerical simulation methods were used to study the flow performance and heat transfer performance of the integral spiral fin tube bundle with respect to the structure parameters. Moreover, the structural parameters of the spiral finned tube are optimized through the genetic algorithm NSGA II, with Fanning friction factor (f) and total heat transfer coefficient ($\eta_0\alpha_0$) as targets.

2. METERIALS AND METHODS

2.1 Geometry and boundary conditions

The integral spiral finned tube structure and tube bundle arrangement are shown in Fig. 1. In order to study the influence of various geometric parameters on the flow and heat transfer characteristics of the integral spiral finned tube bundle, numerical simulation conditions were designed according to different geometrical parameters of the finned tube. All tube bundles were arranged in a staggered arrangement. The values of each structural parameter are as follows: the base tube diameter is 26~42 mm, the fin height is 5~16 mm, the fin pitch is 6~12 mm, the fin tip width is 0.6~3.5 mm, the width of the fin root is 3~5 mm, the horizontal tube pitch is 80~140 mm, and the longitudinal tube pitch is 70~110 mm.

The calculation unit in this study is shown in Fig. 2. The transverse width is $S_T/2$. The number of longitudinal tube rows is N_L=4. The inlet section of the unit is extended upstream by S_L to ensure uniform inlet flow velocity distribution. The outlet section is extended downstream by $5S_L$ to ensure full flow development and avoid backflow. Two fins are taken along the axial direction of the tube in the calculation area. 0.5 mm are retained at both ends of the fin in order to avoid the complex boundary, so that the height of calculation unit is (1.5 p_f + δ_{fr} +1 mm).

The inlet boundary is the velocity inlet. The flue gas temperature is 375℃. The outlet boundary is the pressure outlet, and the pressure is standard atmospheric pressure (gauge pressure is 0 Pa). The surface of the base tube and fins is non-slip wall. The wall temperature of the base tube is constant at 233℃. The fin surface temperature is obtained by coupling the fluid-solid surface convection heat transfer equation and the solid domain heat transfer equation. The trapezoidal fin section of the calculation unit is set as adiabatic non-slip wall. The rest is the adiabatic sliding wall. The convection term of the discrete equation is second-order upwind style. The diffusion term is discrete in a central difference format. The velocity and pressure terms are decoupled using the SIMPLE algorithm.

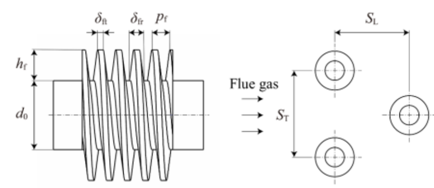


Fig. 1. The integral spiral finned tube structure and tube bundle arrangement

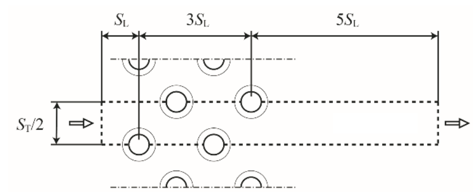


Fig. 2. Schematic diagram of the calculation unit

2.2 Governing equation

The following assumptions are adopted to appropriately simplify the calculation model:

1) The flow is incompressible steady-state flow.

2) The flow and heat transfer outside the tube are independent of the flow inside the tube.

3) The flue gas and metal physical parameters are constant.

4) The radiation heat transfer is ignored.

5) The thermal resistance of ash fouling is ignored

6) The volume force and viscous dissipation are ignored.

The general form of the governing equation is:

$$\mathrm{div}(\rho \mathbf{U}\phi) = \mathrm{div}(\Gamma_\phi \mathbf{grad}\phi) + S_\phi$$

In the formula: ϕ is a general variable, ϕ =1, u_i, T represents the continuity equation, momentum equation and energy equation respectively;

Γ_ϕ is the generalized diffusion coefficient; S_ϕ is the generalized source term.

2.3 Validation of the numerical models

The experimental correlation [5] was used to validate the numerical models, comparing the results of numerical calculations shown in Fig. 3. The difference between the *Nu* calculated by the numerical value and the calculation results of the experimental correlation is 18.24%~-3.3%. And the deviation decreases with the increase of *Re*. Numerical simulation results are similar to the experimental data, while the deviation is within the engineering allowable range. Therefore, the comparison results obtained under the same simulation conditions are credible.

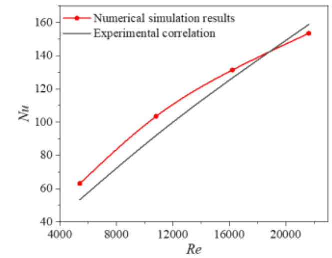
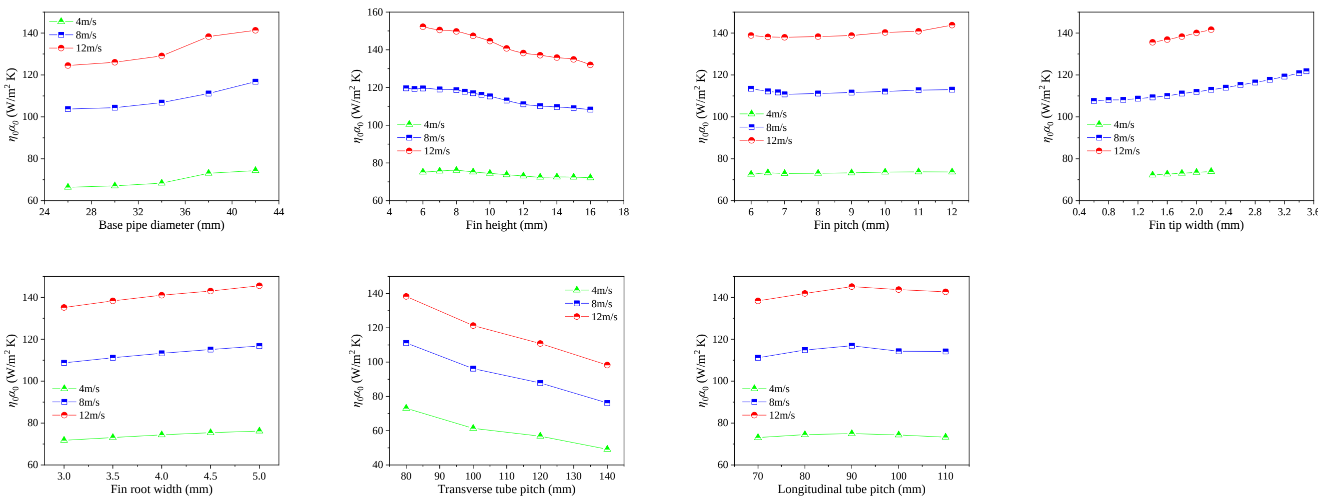


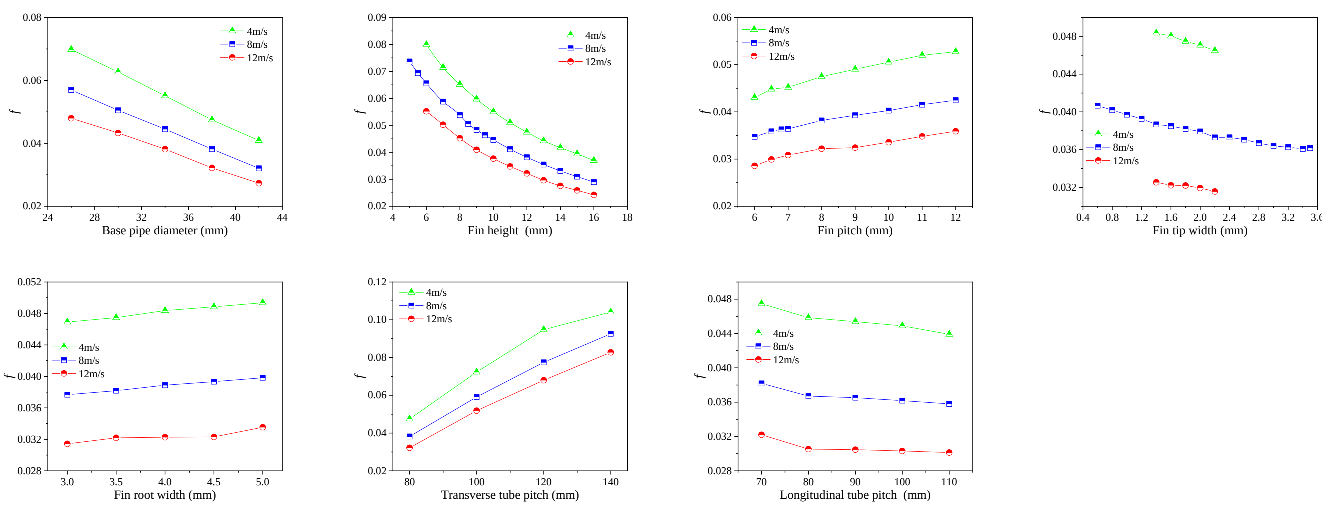
Fig. 3. Validation of the numerical models

2.4 Optimization methods

The multi-objective optimization algorithm NSGA II is applied to optimize the structural parameters of the



A. The influence of structural parameters on the total heat transfer coefficient



B. The influence of structural parameters on the friction factor

Fig. 5. The influence of structural parameters on the flow and heat transfer characteristics

spiral finned tube. The flow and heat transfer characteristics are selected as the optimization target. The total heat transfer coefficient is used to characterize the heat transfer characteristics. The Fanning friction factor f is used to characterize the flow characteristics of the finned tube.

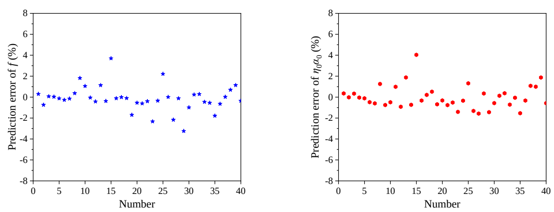


Fig. 4. The prediction error of objective function

The objective function is obtained through the RBF interpolation model. The basis function of the model is the cubic basis function ($\phi(r)=r^3$). The cubic basis function has a simple structure which is superior to Gaussian basis function in predicting the flow parameters of spiral finned tubes. Total of 209 sets of calculated data are used as sample variables. And 40 sets of calculated data are used as test variables to evaluate the objective function derived by RBF model. The results are shown in Fig. 4. The prediction error is limited to less than 5%, indicating that the obtained objective function is credible.

3. RESULTS AND DISCUSSION

3.1 Influence of structure on flow and heat transfer characteristics

Fig. 5 shows the influence of structural parameters on the flow and heat transfer characteristics of integral spiral finned tube. The results show that as the base tube diameter increases (or the fin height increases / the fin pitch decreases / the fin tip width increases / the fin root width decreases / the lateral pitch decreases / the longitudinal pitch increases / the entrance flue gas flow velocity increases), the flow resistance of the finned tube bundle increases. As the base tube diameter increases (or the fin height decreases/ the fin tip width increases/ the fin root width increases/ the lateral pitch decreases/ the inlet flue gas flow velocity increases), the total heat transfer coefficient of the fin tube bundle increases. The total heat transfer coefficient is not greatly affected by the fin pitch. With the increase of the longitudinal pitch, the total heat transfer coefficient first increases and then decreases, and the optimum is around 90 mm.

3.2 Integral spiral finned tube structure optimization

The RBF function interpolation method is adopted to determine the objective function. The total heat transfer coefficient and Fanning friction factor f are taken as the optimization targets. Then, the spiral finned tube structure and operating parameters are optimized

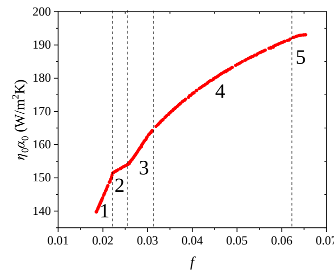


Fig. 6. Pareto front for the friction factor and Total heat transfer coefficient

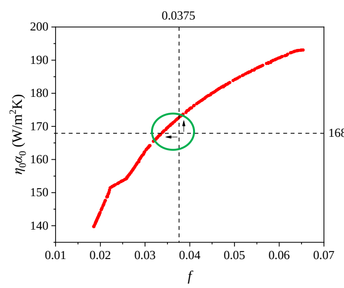


Fig. 7. Selected area on the Pareto front

through the NSGA II algorithm. The parameters considered are: base tube diameter 24~40 mm, fin height 8~14 mm, fin pitch 7~11 mm, fin tip width 1~2.4 mm, fin root width 3~4.2 mm, transverse tube pitch 80~120 mm, the longitudinal tube pitch 70~100 mm, and the inlet flow velocity 6~12 m/s. After optimization, the Pareto optimal solution of the target was obtained as shown in Fig.6. The ranges of the optimal total heat transfer coefficient and Fanning friction factor of the spiral finned tube are 139 to 153 W/m^{2}K and 0.018 to 0.066, respectively.

Table 1 Structural parameters on the Pareto front

number	d_0 (mm)	p_f (mm)	h_f (mm)	δ_{ft} (mm)	δ_{fr} (mm)	S_T (mm)	S_L (mm)	v (m/s)
1	40	11	14	2.4	3~4.2	80	100	12
2	40	7~11	14~13	2.4	4.2	80	83~100	12
3	38~33	7	14	2.4	4.2	80	89~91	12
4	33~29	7	14~8	2.4	4.2	80	90~93	12
5	29~26	7	8	2.4	4.2	80	93~94	12

Table 1 shows the optimal structure design parameters on the Pareto front. The fin tip width and transverse tube pitch were constant in the structure corresponding to the Pareto front, which means that within the parameters range, the influence of fin tip width and transverse tube pitch on the flow and heat transfer characteristics of the fin tube might be monotonous. If the heat exchangers were desired to have a larger heat exchange capability, smaller base tube diameter, lower fin pitch and fin height, larger fin tip width and fin root width are required. Supposing that both flow and heat transfer are limited, more refined scope was obtained as shown in Fig. 7. The restriction condition of the Pareto front in this area is $\eta_0\alpha_0$ >168 W/m^{2}K and f<0.0375. Under this limited situation, the optimal solution structure parameters could be taken as: base tube diameter 32 mm, fin height 13 mm, fin pitch 7 mm, fin tip width 2.4 mm, fin root width 4.2 mm, transverse tube pitch 80 mm, longitudinal tube pitch 90 mm, and inlet flow velocity 12 m/s.

4. CONCLUSION

In this study, the numerical method was applied to investigate the flow and heat transfer characteristics of the integral spiral finned tube. The influence of structural parameters was discussed in detail. The structural parameters were optimized through the multi-objective optimization genetic algorithm, with Fanning friction factor and total heat transfer coefficient as targets. The optimal structural parameters were obtained while considering the flow and heat transfer characteristics that: the base tube diameter is 32 mm, the fin height is 13 mm, the fin pitch is 7 mm, the fin tip width is 2.4 mm, the fin root width is 4.2 mm, the transverse pitch is 80 mm, the longitudinal pitch is 90 mm, and the inlet flow velocity is 12 m/s.

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