EXPERIMENTAL STUDY ON THE HEAT TRANSFER AND RESISTANCE CHARACTERISTICS OF PIN-FIN TUBE

by

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In order to obtain the influence of the structural parameters of the pin-fin tube on the heat transfer and resistance characteristics, ten different structural parameters of the pin-fin tube bundle were studied in this paper. The influence of the transverse pitch and the longitudinal pitch of the tube bundles, the transverse fin spacing and the height of the fin on the heat transfer and resistance characteristics of the pin-fin tube bundle were obtained, and the corresponding correlations were proposed. The results showed that the convection heat transfer coefficient and the flow resistance increased with the longitudinal pitch of the tube. The transverse fin spacing decreased, the heat transfer coefficient and the flow resistance increased. The change of the transverse pitch and the height of the pin-fin had little effect on the heat transfer and resistance performance of the tube bundles.

Key words: *pin-fin tube, heat exchanger, heat transfer characteristics, resistance characteristics*

Introduction

Currently, more and more attention has been paid to the utilization of waste heat resources in China [1]. Heat exchanger is the core component of waste heat recovery device and widely used in high energy-consuming industries, such as chemical industry, petroleum, electric power, and metallurgy. Heat exchanger accounts for 40% of the total equipment in chemical and oil refining units, accounting for 30-45% of the total investment [2, 3]. In order to improve the efficiency of heat exchanger, heat transfer enhancement is an effective measure. Up to now, many passive methodologies have been proposed. The extended surface has been proved to be an effective measure to improve the thermal performance of heat exchangers by enhancing gas-side heat transfer. Pin-fin tube bundles is a typical equipment with extended surface and widely used.

Pin-fin tube as a kind of heat transfer element with extended surface can not only enlarge the heat transfer area, but also significantly enhance the convective heat transfer coefficient of the fin. Therefore, many scholars have studied its characteristics. Metzger *et al.* [4] experimentally studied the heat transfer characteristics of the heat exchangers with arrays of short pin-fins in rectangular ducts. The results showed that the transverse intercept of pin-fins

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had a great influence on the heat transfer of pin-fins, and the smaller the transverse intercept, the larger the convective heat transfer coefficient. Wirtz *et al.* [5] found that the heat transfer and resistance coefficient of the heat transfer surface with staggered pin-fins were higher than that of the heat transfer surface with in-line pin-fins. Al-Jamal and Khashashneh [6] reported the heat transfer and resistance performance of triangular and circular pin-fin arrays under the same heat flux and the same substrate. The results showed that the Nusselt number of circular pin-fin array was higher than that of triangular pin-fin array [6]. Li *et al.* [7] reported the heat transfer and flow resistance characteristics of short elliptical pin-fin arrays arranged staggered in rectangular ducts. The results showed that the heat transfer coefficient of the elliptical pin-fin was higher than that of the circular pin-fin. Tahat *et al.* [8] investigated the heat transfer characteristics of staggered and in-line pin-fins, and proposed the heat transfer correlation of staggered and in-line pin-fins, respectively. Jonsson and Moshfegh [9] experimentally and numerically studied the heat transfer and resistance characteristics of seven types of heat dissipation elements. The results showed that pin-fin heat transfer elements were not suitable for the conditions with high Reynolds number. Soodphakdee *et al.* [10] reported the heat transfer performance of various commonly used pin-fin heat exchanger elements, including elliptical pin-fin, circular pin-fin, and square pin-fin in staggered arrangement and in-line arrangement. It was found that the elliptical pin-fin had the best heat transfer performance under low resistance and low pump power conditions, and the circular pin-fin had the best heat transfer performance under high resistance and high pump power conditions. Ail *et al.* [11] examined the angle effect of pin-fin heat sink channel using water based graphene nanoplatelets nanofluids, and found heat sink with 22.5° channel angle shows better thermal performance as compared to other tested heat sinks. Sara [12] analysed the convective heat transfer performance of pin-fins under the same pump power and founded that the pin-fin group under low Reynolds number condition performed better. Ail *et al.* [13-16] studied the effect of thermal conductivity, pin height, pin thickness, pin spacing, vapour velocity, and pin geometry on pin-fan tubes. Obviously, the researchers aforementioned mainly focused on pin-fins in flat or rectangular channels, which were used in small heat exchangers such as gas turbine blades, integrated circuit chip cooling equipment, oil or gas fired marine boilers and lubricating oil coolers. However, there are few studies on the flow and heat transfer characteristics of pin-fin tube bundles used as waste recovery device in boiler.

Therefore, it is necessary to study the heat transfer and resistance characteristics of pin-fin tube bundles under the background of large-scale thermal energy utilization equipment, such as low-temperature economizer, waste heat boiler. In this paper, the heat transfer and resistance characteristics of pin-fin tubes are studied by building a hot wind tunnel test bench, and the formulas with high precision for calculating the heat transfer and resistance characteristics of pin-fin tubes are obtained to guide the engineering application of pin-fin tubes.

Experimental system and data processing

Experimental system

The experiment was carried out on a hot wind tunnel platform including a hot wind system and a circulating water system, as shown in fig. 1. Air was blown by a frequency converter fan, and the air-flow rate was adjusted by the frequency converter through the control of the fan speed and measured by a vortex flowmeter. The air was heated to 150 °C by an electric heater, and then equalized the flow through the inlet equalizer and entered the test section. The water in the water tank entered the test section after passing through the pump, electromagnetic flowmeter and the water temperature measuring section of inlet. Then entered the temperature measuring section of outlet water after heat exchanging with the hot air outside the tube. Finally, the water flowed back to the water tank, and formed a closed water circulation system.

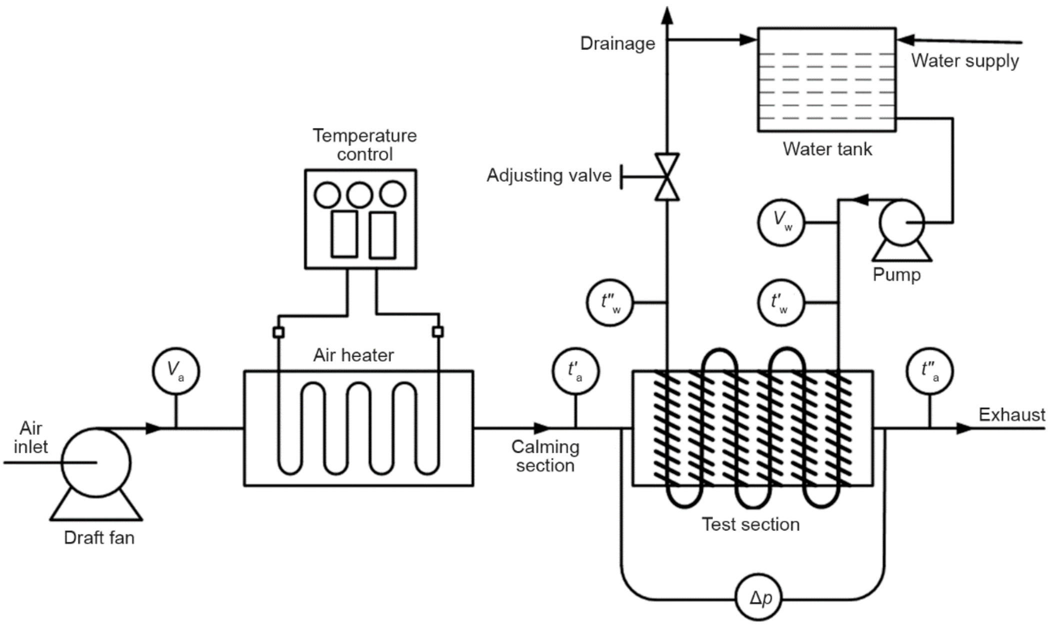

Figure 1. Schematic diagram of the experimental system; t_w – inlet water temperature, t''_w – outlet water temperature, t'_a – inlet air temperature, t''_a – outlet air temperature, V_w – water flow, V_a – air-flow, Δp – pressure drop of tube bank

During measurements, the inlet water temperature was 30 °C, and the inlet air temperature of the experimental section was 150 °C. The inlet and outlet water temperature are measured by Pt 100 thermal resistance with a calibrated accuracy of A grade. The air heater is composed of air heating module and temperature control measurement. The accuracy of temperature control is ±1 °C.

The Reynolds number of the working fluid was more than 12000, which belonged to complete turbulence state. Air velocity at minimum sectional area was 6-16 m/s. The corresponding Reynolds number of the air outside the pin-fin tube was 8000-30000, which covered the range of Reynolds number about flue gas on each heating surface of waste heat utilization equipment, such as waste-heat boiler and low temperature economizer.

The air-flow is measured by a vortex flowmeter model LUGB-2320. The accuracy level of the vortex flowmeter is 1.5 (Chinese standard, ±1% of display value). Before measurement, the vortex flowmeter was checked by standard flowmeter. The water flow is measured by electromagnetic flowmeter with accuracy of 0.5 grade.

Test section

In this paper, ten different tube bundles with outside diameter of Ø45 mm were experimentally studied. The effects of four structural parameters on heat transfer and resistance characteristics of pin-fin tube bundles were investigated, including transverse spacing, S_1, lon gitudinal spacing, S_2, transverse fin spacing, P_h, and variable fin height, H. The main structure parameter for testing pin-tube were shown in tab. 1, and the geometric variables are illustrated in fig. 2. The sketch of the test section was shown in fig. 3. The width and the height of the test section was 320 mm and 9 times the transverse spacing. Four rows and ten columns arrangement were adopted for pin-fin tube bundles, which could be considered sufficient to obtain representative heat transfer and pressure drop data according to previous experiments.

Table 1. Main structure parameter for testing pin-fin tube

Tube bundle	H [m]	P_z [m]	P_h [m]	S_1 [m]	S_2 [m]	N
E1	0.108	0.0157	0.028	0.113	0.095	6
E2	0.108	0.0157	0.028	0.118	0.095	6
E3	0.108	0.0157	0.028	0.123	0.095	6
E4	0.108	0.0157	0.028	0.113	0.09	6
E5	0.108	0.0157	0.028	0.113	0.100	6
E6	0.108	0.0157	0.016	0.113	0.095	6
E7	0.108	0.0157	0.020	0.113	0.095	6
E8	0.108	0.0157	0.024	0.113	0.095	6
E9	0.096	0.0157	0.028	0.113	0.095	6
E10	0.102	0.0157	0.028	0.113	0.095	6

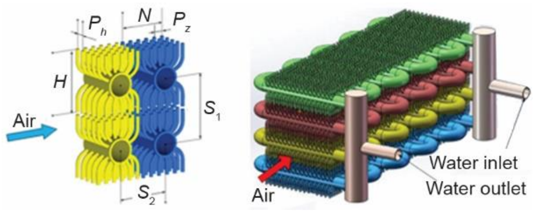

Figure 2. Geometric schematic diagram of the pin--fin tube and tube bank with in-line arrangement

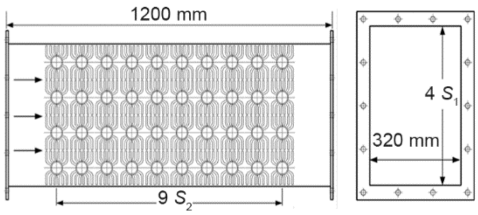

Figure 3. Dimensions of the test section

Data processing

The overall heat transfer coefficient was calculated using eqs. (1) and (2):

$$Q = K A_o \Delta t_m \tag{1}$$

where Q [kW] is the calculated heat transfer, K [Wm^{-2}K^{-1}] – the total heat transfer coefficient, A_o [m^2] – the fin side total heat transfer area, and Δt_m [°C] – the logarithmic mean temperature in counter-current arrangement.

Logarithmic mean temperature in counter-current arrangement Δt_m is calculated by:

$$\Delta t_m = \frac{(t'_a - t''_w) - (t''_a - t'_w)}{\ln \dfrac{t'_a - t''_w}{t''_a - t'_w}} \tag{2}$$

$$\frac{1}{K}=\frac{A_o}{\alpha_o(A_t+\eta_f A_f)}+\frac{\delta_t}{\lambda_t}\frac{A_o}{A_m}+\frac{1}{\alpha_i}\frac{A_o}{A_i} \tag{3}$$

The fin side convection heat transfer coefficient, α_o, was calculated using eq. (4):

$$\alpha_o=\left(\frac{1}{K}-\frac{\delta_t}{\lambda_t}\frac{A_o}{A_m}-\frac{1}{\alpha_i}\frac{A_o}{A_i}\right)^{-1}\frac{A_o}{A_t+\eta_f A_f} \tag{4}$$

where K [Wm^{-2}°C^{-1}] is the overall heat transfer coefficient, δ_t [m] – the thickness of base tube, λ_t [Wm^{-2}°C^{-1}] – the thermal conductivity of base tube, A_m [m^2] – the base tube heat transfer area, A_i [m^2] – the heat exchange area of inwall of base tube, and α_i [Wm^{-2}K^{-1}] – the convection heat transfer coefficient of the inwall of base tube based on A_i.

The fin efficiency was determined using eq. (5). Firstly, the fin efficiency is treated by assuming an initial value and the convective heat transfer coefficient, α_o, on the air side was calculated according to eq. (3). Then the fin efficiency was calculated by eqs. (6) and (7). Compared with the assumed initial value, if the difference was less than 0.1%, the calculated value was available. The air side convective exchange coefficient and the fin efficiency were obtained. Otherwise, the iterative calculation continued:

$$\eta_f=\frac{Q_o}{Q_a} \tag{5}$$

where Q_o [kW] is the actual heat transfer quantity and Q_a [kW] – the heat transfer quantity when the fin bed temperature was on behalf of the entire fin temperature:

$$m=\sqrt{\frac{4\alpha_o}{\lambda_a d}} \tag{6}$$

where d [m] is the pin fin diameter and λ_a [Wm^{-1}K^{-1}] – the thermal conductivity of air.

$$\eta_f=\frac{\tanh(mh)}{mh} \tag{7}$$

where h [m] is the average height of pin-fin.

In this paper, the dimensionless criterion was used to characterize the heat transfer and resistance characteristics of pin-fin side. Specifically, the heat transfer characteristics of the tube bundle were characterized by the change of Nusselt number with Reynolds number, and the resistance characteristics were characterized by the change of Euler number with Reynolds number. Finally, Kline and McClintock method is adopted to estimate the uncertainty, which has been widely used to calculate the uncertainty of Nusselt number and Euler number [17-20].

Experimental results and analysis

Effect of transverse pitch

In this paper, pin-finned tube bundles E1, E2, and E3 were investigated to discover the effects of transverse pitch, S_1, on heat transfer and resistance characteristics. The transverse pitch, S_1, was 113 mm, 118 mm, and 123 mm, respectively. The corresponding transverse relative pitch (S_1/d_o) range was 2.51-2.73. When the transverse pitch of each tube bundle changed, the remaining structural parameters of the pin-fin tube kept invariant, namely d_o = 45 mm, S_2 = 95 mm, P_h = 28 mm, P_z = 15.7 mm, H = 108 mm, N = 6. Among them, the outer diameter d_o is the characteristic length.

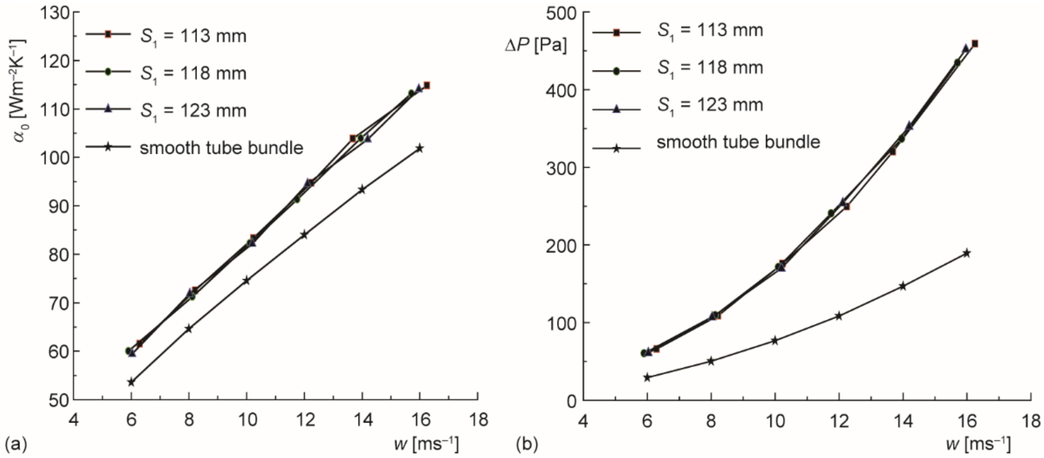

Figure 4. The effect of transverse pitch on heat transfer (a) and resistance performance (b)

As shown in fig. 4, over heat transfer is improved because of the higher α_o than bare tube. When the transverse pitch was given fixed value, the convective heat transfer coefficient and the flow resistance of the pin-fin tube bundle are increased with the air velocity. Because the air velocity increased, the turbulence intensity. Therefore, the heat transfer performance and flow resistance of the pin-fin tube bundles increased with the transverse pitch of the pin-fin tube bundles. When the minimum cross-section air velocity was constant, the convective heat transfer coefficient and flow resistance of the pin-fin tube bundles changed little.

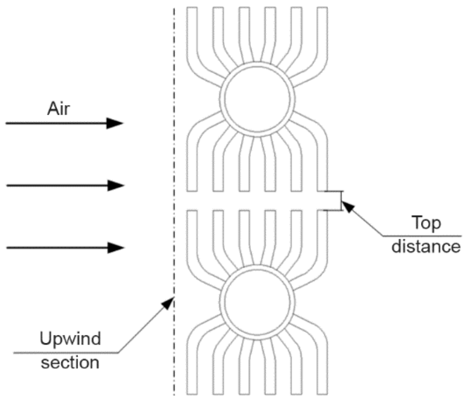

Figure 5. Tube bundles section and air-flow direction

The results showed that when the minimum cross-section air velocity is constant, the increase of the transverse pitch of the tube bundle leads to the increase of the cross-section velocity, as shown in fig. 4, and the width of the flow area between the fin tops of the two rows, that is, the fin top distance in fig. 5 changed from 5 mm to 15 mm, and the air-flow resistance between the fin tops of the two rows was smaller. Therefore, the increase of the lateral pitch of the tube did not enhance the turbulence of the air washed tube bundle.

Figure 7 showed the relationship between the heat transfer performance of three kinds of pin-fin tube bundles with different transverse pitch and the cross-section velocity. The smaller the transverse pitch, the smaller the fin-top pitch, the stronger the turbulence and the better the heat transfer performance of air-flowing across the pin-fin tube bundles.

Effect of longitudinal pitch

In this paper, pin-finned tube bundles E1, E2, E3, E4, and E5 were studied to find the effects of longitudinal pitch, S_2, on heat transfer and resistance characteristics. The longitudinal pitch, S_2, was 90 mm, 95 mm, and 100 mm, respectively. The corresponding longitudinal relative

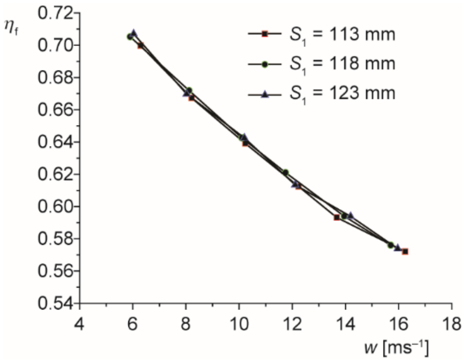

Figure 6. The effect of transverse pitch on tube bundles efficiency

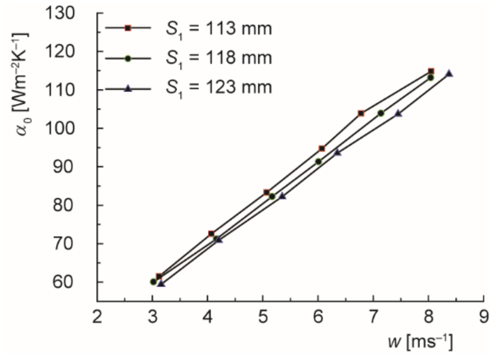

Figure 7. Relationship between tube bundle heat transfer performance and cross-flow velocity in the upwind direction and transverse pitch

pitch (S_2/d_o) range was 2.00-2.22. When the longitudinal pitch of each tube bundle changed, the remaining structural parameters of the pin-fin tube kept invariant, namely d_o = 45 mm, S_1 = 113 mm, P_h = 28 mm, P_z = 15.7 mm, H = 108 mm, N = 6.

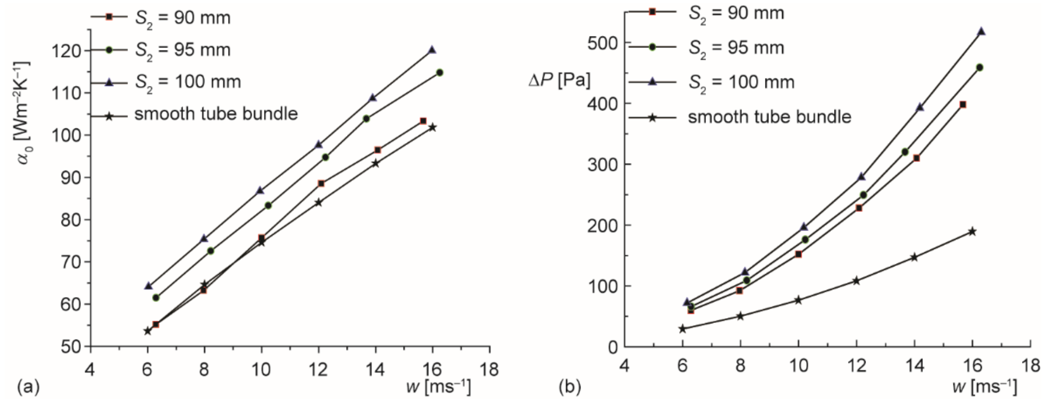

Figure 8. The effect of longitudinal pitch on tube heat transfer and resistance performance

Figure 8 showed that over heat transfer is improved because of the higher α_o than bare tube. When the longitudinal pitch of the tube bundle was constant, the convective heat transfer coefficient and the flow resistance of the tube bundle increased with the air velocity and also increased with the longitudinal pitch. As the longitudinal pitch increased, the convection heat transfer between the back row tube bundle and the air was strengthened, the turbulence density increased. At the same minimum cross-section velocity, when S_2 = 95 mm and S_2 = 100 mm, the convective heat transfer coefficients increased by 10.3% and 14.8%, respectively, and the flow resistance increased by 11.0% and 26.7%, respectively.

Figure 9 showed the relationship between the efficiency, the longitudinal pitch of the tube bundle and the air velocity at the minimum cross-section of the tube bundle. It can be seen that the fin efficiency decreased with the increase of the minimum cross-section air velocity, when the longitudinal pitch was constant. When the minimum cross-section velocity of the tube

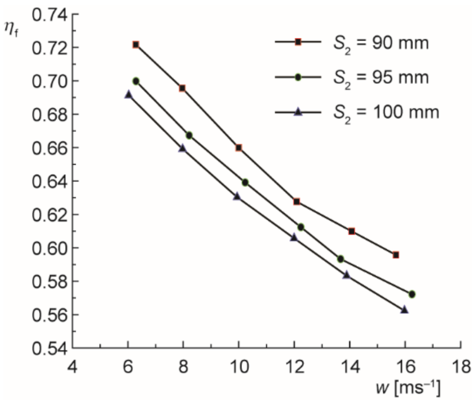

Figure 9. The effect of longitudinal pitch on tube fin efficiency

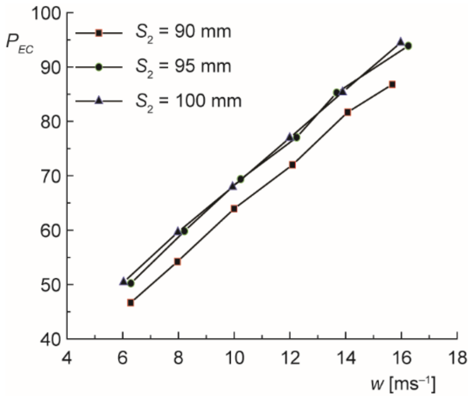

Figure 10. Comparison of heat transfer performance of three kinds of long-pitch pin-fin tube bundles

bundle was constant, the fin efficiency decreased with the increase of the longitudinal pitch, because the longitudinal pitch increased, the greater the turbulence density when air swept the pin-fin tube bundle, the higher the average surface temperature of the pin-fin, so the fin efficiency decreased.

To comprehensively compare the heat transfer and resistance performance of the pin-fin tube bundle, performance evaluation indicator P_{EC} was adopted according to literature [21-25], as shown in eq. (8):

$$P_{EC} = \frac{\mathrm{Nu}}{f^{1/3}} \tag{8}$$

where f is the friction factor, defined:

$$f = \frac{2\Delta p}{\rho w^2} \tag{9}$$

The P_{EC} reflected the convective heat transfer per unit power consumption when the gas flow through the surface of the heat exchanger. The larger the P_{EC}, the better the comprehensive heat transfer performance of the heat exchanger. The overall performance of the three different longitudinal pitch pin-fin bundles were shown in fig. 10. As we can see that longitudinal pitch increased, the value of P_{EC} increased, in other words, the comprehensive heat transfer performance was promoted. Considering the actual engineering application, the increase of the longitudinal pitch increased the longitudinal length of the heat exchanger. Consequently, when the longitudinal pitch was 95 mm, the integrated performance of the pin-fin tube bundle was optimal.

Effect of transverse fin spacing

As can be seen from fig. 11 that over heat transfer is improved because of the higher α_o than bare tube. When the transverse fin spacing was constant, the convective heat transfer coefficient and the flow resistance of the pin-fin tube bundle were increased with the air-flow rate. When the air-flow rate was constant, the transverse fin spacing reduced, then the convective heat transfer coefficient and the flow resistance of the pin-fin tube bundle increased. Because the air corridor between the fins became narrow as transverse fin spacing decreased, which made the air turbulence intensity more strenuous. At the same minimum cross-section flow rate, compared with P_h = 28 mm, corresponding to P_h = 16 mm, 20 mm, and 24 mm, the convective heat transfer coefficient increased by up to 15.0%, 11.4%, and 6.4%, respectively. The flow resistance increased by up to 85.5%, 43.4%, and 19.3%. Comparatively speaking, P_h = 20 mm had better performance.

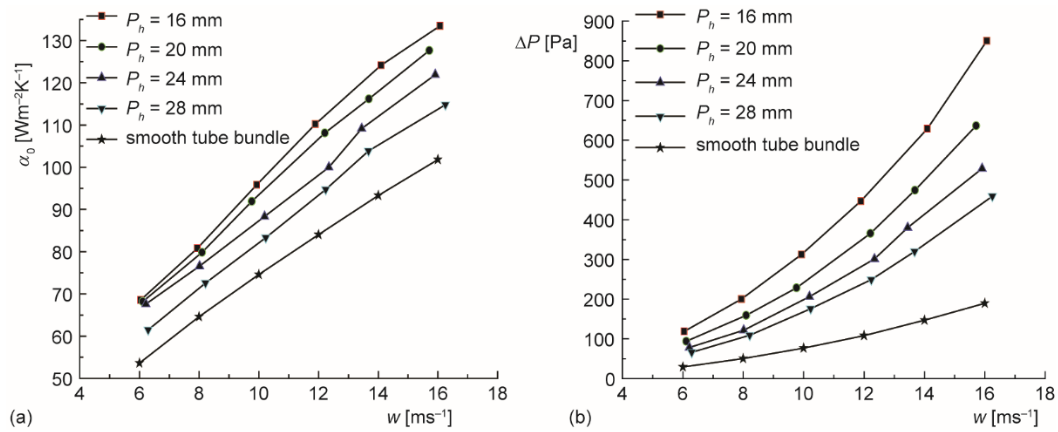

Figure 11. The effect of transverse pitch on heat transfer (a) and resistance performance (b)

Figure 12 showed the relationship between the efficiency of the tube fin and the transverse fin spacing and the minimum cross-section air velocity. The results indicated that the fin efficiency decreased with the increase of the minimum cross-section air velocity when the transverse fin spacing was given fixed value. When the minimum cross-section velocity of the tube bundle was constant, the fin efficiency increased with the transverse fin spacing. As the transverse fin spacing increased, the turbulence intensity of air across the fin surface decreased, the fin surface temperature decreased and the heat transfer temperature difference increased, the heat transfer capacity increased, so the fin efficiency increased.

Figure 13 showed the comparison of the comprehensive heat transfer performance about four kinds of pin-fin tube bundles with different transverse fin spacing. The comparison indicated that the comprehensive heat transfer performance per unit power consumption of pin-fin tube bundles was the best when the transverse fin spacing was 20 mm.

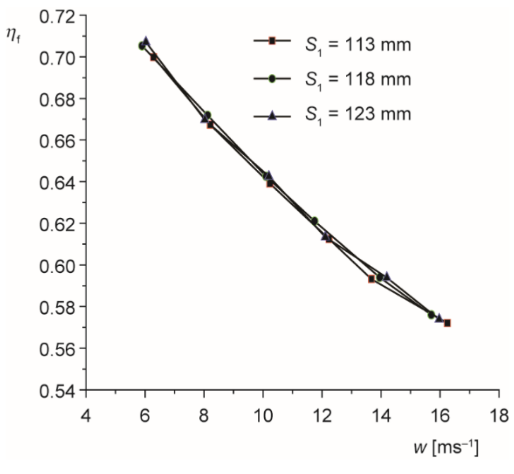

Figure 12. The effect of transversal wing distance variation on tube fin efficiency

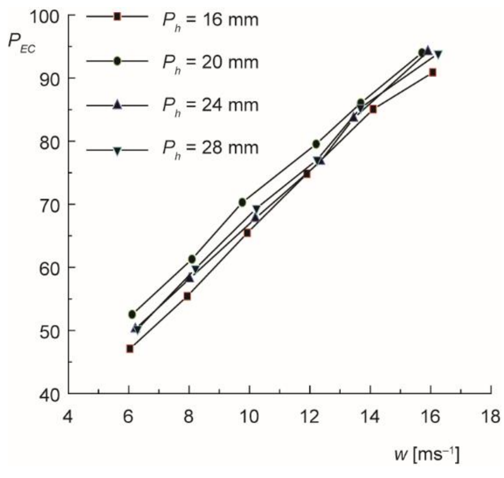

Figure 13. Comparison of heat transfer performance of four kinds of transverse fin span between pin-fin tube bundles

Effect of fin wing height

In this paper, pin-finned tube bundles E1, E9, and E10 were studied about the influence of fin height, H, on heat transfer and resistance characteristics. The height of pin-fin, H, were 96 mm, 102 mm, and 108 mm, respectively. When the pin-fin height changed, the other structural parameters of the pin-fin tube kept invariant, namely d_o = 45 mm, S_1 = 113 mm, S_2 = 95 mm, P_z = 15.7 mm, P_h = 28 mm, N = 6.

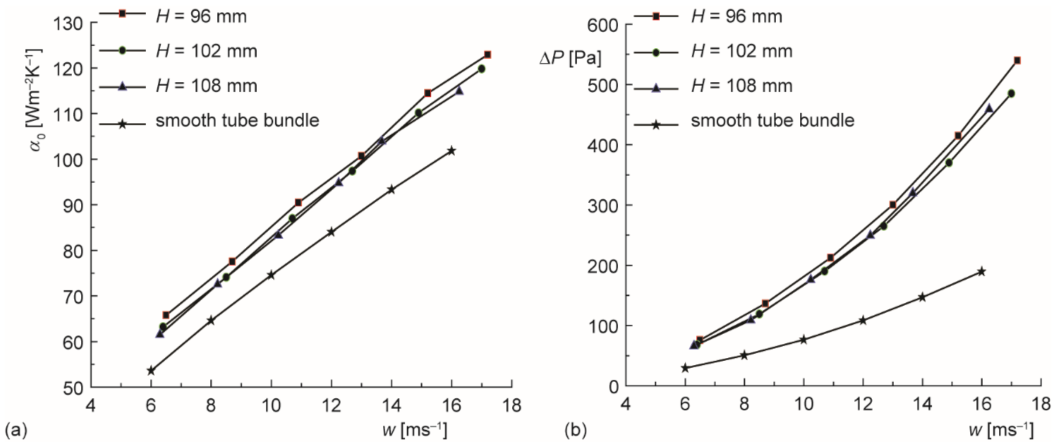

Figure 14. The effect of pin-fin height on heat transfer (a) and resistance performance (b)

Figure 14 showed that over heat transfer is improved because of the higher α_o than bare tube. When the fin height was constant, the convective heat transfer coefficient and flow resistance of the finned tube bundle increased with the air velocity. When the minimum cross-section air velocity was constant, the convection heat transfer coefficient and the flow resistance of tube bundle changed little with the decrease of pin height. The results indicated that, with the decrease of the fin height, the air-flow area between the fin tip increased when the transverse pitch remained unchanged. When the fin height changed from 108 mm to 96 mm, the tip distance shown in fig. 6 changed from 5 mm to 17 mm. Although the cross-section velocity increased, the flow area between the tip of the fin formed a certain width of low flow resistance. The convective scour of air to the pin-fin was weakened.

Figure 15 showed the relationship between the tube fin efficiency and the air velocity of the pin-fin height and the minimum cross-section of tube bundle. As we can see that the fin efficiency decreased with the increase of the minimum cross-section air velocity of the tube bundle. When the minimum cross-section air velocity of was constant, the fin efficiency decreased with the increase of the pin-fin height. Because the average surface temperature of pin-fin increased and fin efficiency decreased, when the height increased.

When the transverse pitch of the tube bundle kept invariant, the higher the pin-fin height, the larger the heat transfer surface area, and the smaller the minimum cross-section. At the same air-flow rate, the flow velocity of the minimum cross-section was larger and the heat transfer performance was better. As shown in fig. 16, the convective heat transfer coefficients outside the pin-fin tube bundles varied with the pin-fin height and the cross-section velocity. The higher the pin-fin height, the better the heat transfer performance.

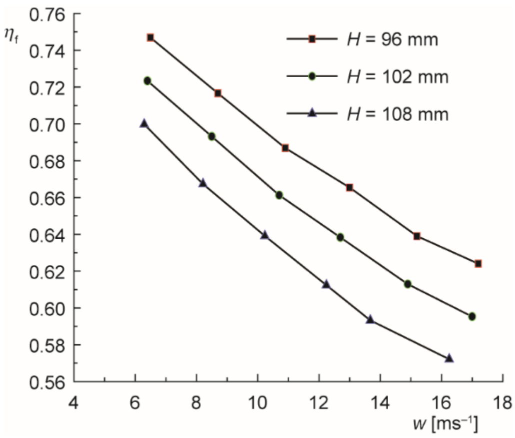

Figure 15. The effect of height variation of pin-fin on the efficiency

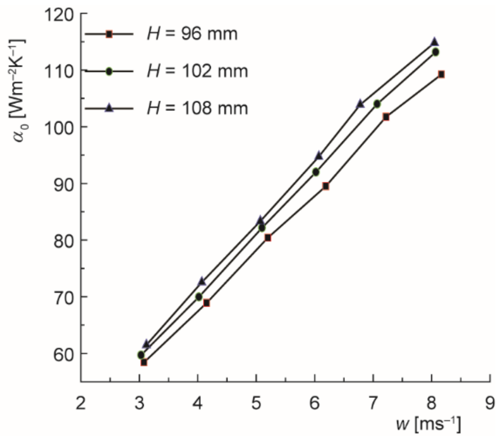

Figure 16. Relationship between heat transfer performance and upwind cross-section velocity

Correlation formula of heat transfer and resistance characteristics

After finished experiments, by multiple regression analysis, the correlations of heat transfer, resistance characteristics, and the formulas of fin efficiency were obtained, including transverse pitch, longitudinal pitch, transverse pitch and pin-fin height.http://fanyi.baidu.com/ - zh/en/javascript:void(0);

Correlation formula of heat-transfer characteristics:

$$\mathrm{Nu} = 0.082\,\mathrm{Re}^{0.663}\,\mathrm{Pr}^{0.33}\left(\frac{P_h}{d_o}\right)^{-0.293}\left(\frac{S_1}{d_o}\right)^{0.033}\left(\frac{S_2}{d_o}\right)^{1.250}\left(\frac{H}{d_o}\right)^{-0.270} \tag{10}$$

The maximum relative deviation between Nusselt number and the experimental results is 4.8%. The error analysis of the correlation of heat transfer characteristics is shown in fig. 17.

Correlation formula of resistance characteristics:

$$\mathrm{Eu} = 0.044\,\mathrm{Re}^{0.027}\left(\frac{P_h}{d_o}\right)^{-1.056}\left(\frac{S_1}{d_o}\right)^{0.112}\left(\frac{S_2}{d_o}\right)^{2.024}\left(\frac{H}{d_o}\right)^{-0.205} \tag{11}$$

The maximum relative deviation between the Euler number and the experimental results is 4.2%. The error analysis of the resistance characteristic correlation is shown in fig. 18.

Fin efficiency calculation correlation:

$$\eta = 11.80\,\mathrm{Re}^{-0.210}\left(\frac{P_h}{d_o}\right)^{0.085}\left(\frac{S_1}{d_o}\right)^{0.033}\left(\frac{S_2}{d_o}\right)^{-0.43}\left(\frac{H}{d_o}\right)^{-0.623} \tag{12}$$

The maximum relative deviation between η and the experimental results is 4.8%. The error analysis of the correlation of heat transfer characteristics is shown in fig. 19.

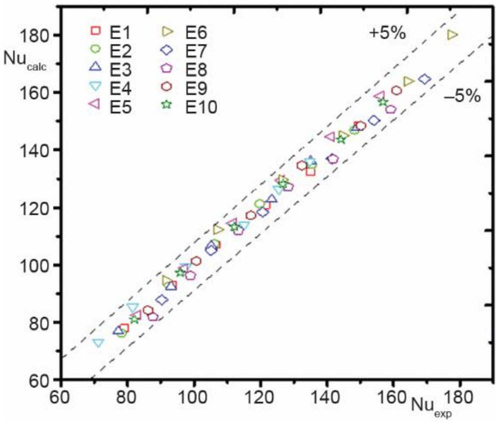

Figure 17. Error analysis of correlation formula of heat transfer characteristics

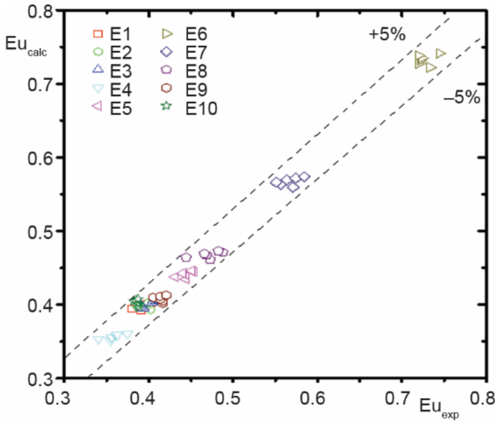

Figure 18. Error analysis of relative formula of resistance characteristics

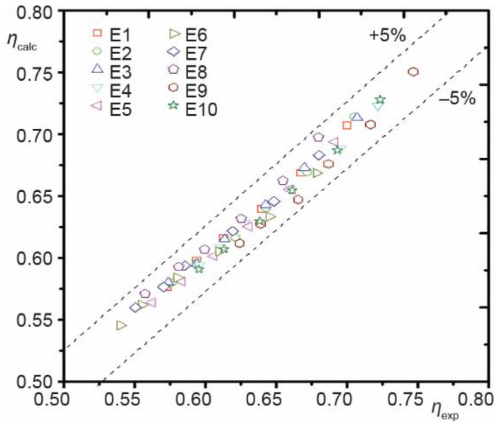

Figure 19. Error analysis of relative formula of fin efficiency

Conclusions

A set of experiments was performed to evaluate the influence of the structural parameters of the pin-fin tube on the heat transfer and resistance characteristics. Based on the experimental results, the following conclusions were drawn as follows.

- The longitudinal pitch and transverse fin spacing had great influence on the heat transfer and resistance performance of pin-fin tubes. With the increase of longitudinal pitch, the convective heat transfer coefficient outside the tube bundle and the flow resistance increased. When the longitudinal pitch was 95 mm, the comprehensive heat transfer performance of the tube bundle was the best. When the transverse fin spacing decreased, the convective heat transfer coefficient outside the tube bundle increased, and the flow resistance increased. When the transverse fin spacing was 20 mm, the comprehensive heat transfer performance of the tube bundle was the best.
- Considering the compactness of the heat exchanger and increasing the heat transfer area, the experimental tube bundle E7 had the best comprehensive heat transfer performance when the transverse pitch of the tube bundle was 113 mm and the fin height was 108 mm (pin-fin distance 5 mm).
- The correlations of heat transfer and resistance calculation and fin efficiency calculation for pin-fin tube bundles were presented, including the effects of transverse pitch, longitudinal pitch, transverse pitch and pin-fin height.

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Nomenclature

A_b – base tube area, [m^2]
A_f – fin area, [m^2]
A_i – heat exchange area of in-wall of base tube, [m^2]
A_m – base tube heat transfer area, [m^2]
A_o – fin side total heat transfer area, [m^2]
A_t – base tube area
d – pin-fin diameter, [m]
d_o – the outer diameter of pin-fin tube, [mm]
Eu – Euler number (= $\Delta p/\rho w^2$)
H – variable fin height, [mm]
h – average height of pin-fin, [m]
K – overall heat transfer coefficient, [Wm^{-2}°C^{-1}]
Nu – Nusselt number (= $\alpha_o d_o/\lambda_a$)
P_h – transverse fin spacing, [mm]
P_z – performance evaluation indicator, [mm]
Q – calculated heat transfer, [kW]
Q_a – heat transfer quantity when the fin bed temperature was on behalf of the entire fin, [kW]
Q_o – actual heat transfer quantity, [kW]
Δp – pressure drop
Re – Reynolds number (= $\rho v d_o/\mu$)
S_1 – transversal pitch, [mm]
S_2 – longitudinal pitch, [mm]
Δt_m – logarithmic mean temperature in counter-current arrangement, [°C]
w – air velocity, [ms^{-1}]

Greek symbols

α_i – convection heat transfer coefficient of the in-wall of base tube based on A_i, [Wm^{-2}K^{-1}]
α_o – convection heat transfer coefficient of air side, [Wm^{-2}K^{-1}]
δ_t – thickness of base tube, [m]
λ_a – thermal conductivity of air, [Wm^{-1}K^{-1}]
λ_t – thermal conductivity of base tube, [Wm^{-2}°C^{-1}]
ρ – density
μ – dynamic viscosity

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