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Thermal Design of a Shell and Tube Heat Exchanger with Internal Fins

Lei Hu, Ting Ma, Pan Zhang, Qiuwang Wang*

Key Laboratory of Thermo-Fluid Science and Engineering, MOE, Xi'an Jiaotong University, Xi'an, Shaanxi, 710049, China
wangqw@mail.xjtu.edu.cn

To alleviate the severe challenge of energy depletion, it's extremely urgent to improve the available heat recovery technology. Heat exchangers play a key role in waste heat recovery and have attracted much attention. To recover waste heat, a shell and tube heat exchanger with sinusoidal wavy fins and blocked-core tube inside the tube, and with bare surfaces outside was designed by segmented log-mean temperature difference (LMTD) method. In the shell and tube heat exchanger, liquid water outside the tube was heated by compressed air inside. The effect of air-side inlet pressure on the geometry of heat exchanger was investigated under identical temperature conditions ($T_{a,i}$, $T_{a,o}$, $T_{w,i}$, $T_{w,o}$) and water-side inlet pressure. The results show that air-side inlet pressure has little effect on the structure parameters of heat exchanger, but greatly affect the pressure drop of air side.

1. Introduction

With the rapid growth in demand for heat recovery across the world and the increasing likelihood of energy depletion in coming decades, improving the efficiency of heat exchangers has taken on heightened urgency. As a typical compact heat exchanger, heat exchanger with internal fins has been attracting great attention and significantly benefiting modern industry.

In the past decades, a variety of internal fins have been proposed. Furthermore, many researchers have investigated the performance of internally finned tube both experimentally and numerically. Wang et al. (2008) found that the sinusoidal wavy fin had the best comprehensive performance. Lin et al. (2011) conducted numerical simulation and concluded that the local Nusselt numbers distributed periodically. Additionally, Zeitoun et al. (2004) analysed the effect of fin heights. Halıcı et al. (2001) experimentally investigated the effect of tube row number. Peng et al. (2016) analysed the thermo-hydraulic performances of an internally finned tube. The numerical and experimental results indicated that the wave angle significantly affected the thermal performance. Duan et al. (2018) investigated the turbulent flow and heat transfer characteristics of double-tube structure internal finned tube and found that the increase of fin numbers led to more uniform distribution of temperature and velocity field.

The performance of the heat exchanger with internal fins have been widely investigated. However, for industrial production, the inlet pressure of working fluid will change significantly in some cases. To meet the needs of rapid production in industry, it's urgent to analyse the relation between inlet pressure and the geometry of heat exchanger. In this paper, to make sure whether the geometry of heat exchanger need to be changed under different inlet pressure, the thermal design of a shell and tube heat exchanger with sinusoidal wavy fins and blocked-core tube inside and bare surfaces outside was conducted. In the shell and tube heat exchanger, the heat source is compressed air inside the tube, and the cold source is water outside. The effect of air-side inlet pressure on the geometry was investigated.

2. Thermal design algorithm

The simplified structure of the shell and tube heat exchanger is shown in Figure 1. The sinusoidal wavy fins and blocked-core tube are shown in Figure 1(c). The heat exchanger mainly consists of two air boxes, several baffles and a certain number of tubes. The baffles and tubes are arranged between the two air boxes. The tube pass

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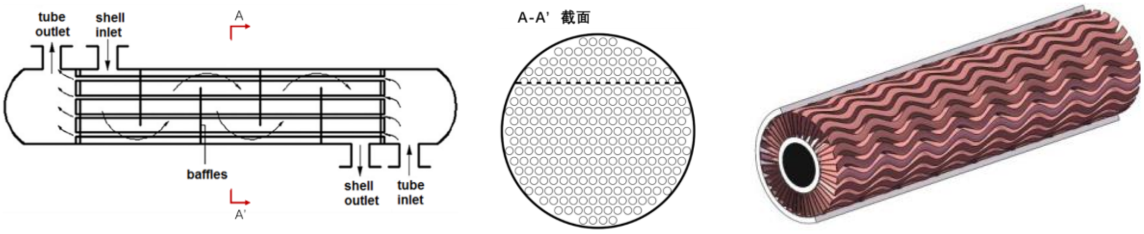
number, the baffle number, and the tube number are determined by the performance parameters of the heat exchanger.

2.1 Segmented LMTD method

LMTD method is widely applied for the design of heat exchanger. The log-mean temperature difference between the cold water and the compressed air is calculated as:

$$\Delta T_{\mathrm{m}} = \frac{(T_{\mathrm{a,i}} - T_{\mathrm{w,o}}) - (T_{\mathrm{a,o}} - T_{\mathrm{w,i}})}{\ln \dfrac{(T_{\mathrm{a,i}} - T_{\mathrm{w,o}})}{(T_{\mathrm{a,o}} - T_{\mathrm{w,i}})}} \tag{1}$$

where T_a, T_w are the temperature of air and water, respectively. Subscript *i* represents inlet of the fluid, and *o* represents outlet.



(a) Heat exchanger (b) Baffle (c) Sinusoidal wavy fins

Figure 1: Schematic of heat exchanger

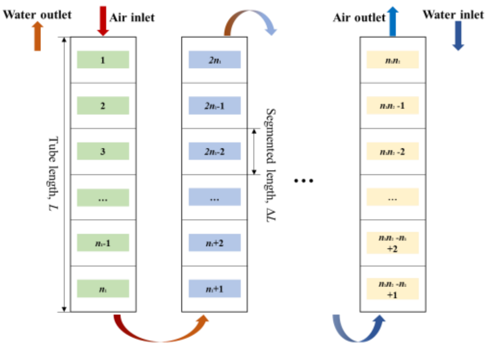


Figure 2: Illustration of segmented LMTD method

While using LMTD method, the specific heat capacity at certain pressure is always assumed to be constant. However, in fact, it will change with temperature inevitably. Hence, it’s not accurate to take specific heat capacity as a constant, especially when the temperature change is relatively high. Therefore, segmented LMTD method was employed in this paper as a modified LMTD method to increase accuracy. For the traditional LMTD method, the entire heat exchanger is taken as a whole. However, in segmented LMTD method, the heat exchanger is discretized to many small heat transfer elements and each small element is taken as an independent heat exchanger, as shown in Figure 2. In every small element, the temperature difference is much smaller, and the change of specific heat capacity can be ignored. Therefore, it’s reasonable to take the specific heat capacity as constant in each heat transfer element.

2.2 Heat balance

For the segmented LMTD method, the total number of heat transfer elements is:

$$N = n_1 n_2 \tag{2}$$

where n_1 is the number of heat transfer elements in one tube pass, and n_2 is the number of tube passes. The total heat transfer rate is calculated as:

$$Q = \sum_{j=1}^{N} Q_{\mathrm{j}} \tag{3}$$

where Q_j represents the heat transfer rate of the j^{th} ($1 \le j \le N$) heat transfer element, and Q_j is defined as:

$$Q_j = q_{\mathrm{m,w,j}} c_{\mathrm{p,w,j}} (T_{\mathrm{w,o,j}} - T_{\mathrm{w,i,j}}) = q_{\mathrm{m,a,j}} c_{\mathrm{p,a,j}} (T_{\mathrm{a,i,j}} - T_{\mathrm{a,o,j}}) \tag{4}$$

$$Q_{\mathrm{j}} = k_{\mathrm{j}} A_{\mathrm{j}} \Delta T_{\mathrm{m,j}} \tag{5}$$

where k_j is the overall heat transfer coefficient, and A_j is the total heat transfer area, and $\Delta T_{m,j}$ is the log-mean temperature difference. $q_{m,j}$, $c_{p,j}$, $T_{o,j}$, and $T_{i,j}$ are the mass flow rate, the specific heat capacity at certain pressure, the outlet temperature, the inlet temperature of fluid, respectively. Subscript *w* represents water, and *a* represents compressed air.
The overall heat transfer coefficient *k* is calculated by:

$$\frac{1}{kA} = \frac{1}{\eta_{\mathrm{i}} h_{\mathrm{i}} A_{\mathrm{i}}} + R_{\mathrm{fi}} + \frac{\ln(D_{\mathrm{o}}/D_{\mathrm{i}})}{2\pi \lambda_{\mathrm{wa}} L} + R_{\mathrm{fo}} + \frac{1}{\eta_{\mathrm{o}} h_{\mathrm{o}} A_{\mathrm{o}}} \tag{6}$$

where L is the tube length. D_o and D_i are the outer diameter and inner diameter of the outside tube. λ_{wa} is the thermal conductivity. η_i and η_o are the surface efficiency. h_i and h_o are the convective heat transfer coefficient. A_i and A_o are the heat transfer area. R_{fi} and R_{fo} are the surface thermal resistance. Subscript *i* represents inside the tube, and *o* represents outside the tube.
The effectiveness of heat exchanger is defined as:

$$\varepsilon = \frac{T_{\mathrm{w,o}} - T_{\mathrm{w,i}}}{T_{\mathrm{a,i}} - T_{\mathrm{w,i}}} \tag{7}$$

2.3 Water side

For water side, the heat transfer mode is convection between the water and the outer wall of the tube. The water is heated and due to its relatively low temperature (from 290 K to 330 K), the radiation between water and the outer tube is negligible. There is an arc-form gap in every baffle, as shown in Figure 3.
The water-side convective heat transfer coefficient is estimated as:

$$h_{\mathrm{w}} = 1.72 \frac{\lambda_{\mathrm{w}}}{D_{\mathrm{o}}^{0.4}} Re_{\mathrm{w}}^{0.6} Pr_{\mathrm{w}}^{1/3} (\frac{\mu_{\mathrm{w}}}{\mu_{\mathrm{wa}}})^{0.14} \tag{8}$$

where μ_w is the average viscosity, and μ_{wa} is the local viscosity near tube wall.
The Reynolds number of water is defined as:

$$Re_{\mathrm{w}} = \frac{\rho_{\mathrm{w}} u_{\mathrm{w}} d_e}{\mu_{\mathrm{w}}} \quad (200 \le Re_{\mathrm{w}} \le 60{,}000) \tag{9}$$

where u_w is the average velocity in the cross section, and d_e is the hydraulic diameter of shell side.

$$d_e = \frac{D^2 - N_{\mathrm{t}} D_{\mathrm{o}}^2}{D + N_{\mathrm{t}} D_{\mathrm{o}}} \tag{10}$$

$$A_{\mathrm{m}} = \sqrt{A_{\mathrm{s}} A_{\mathrm{w}}} \tag{11}$$

$$A_{\mathrm{w}} = A_{\mathrm{g}} (1 - \beta) \tag{12}$$

$$A_{\mathrm{S}} = BD(1 - \frac{D_o}{l}) \tag{13}$$

$$\beta = 0.907 (\frac{D_o}{l})^2 \tag{14}$$

where N_t is the total number of tubes. D is the shell inner diameter. A_m is the area of cross section. A_g is the area of the arc form. B is the distance between two baffles. l is the central distance between two neighboring tubes.
The area of the arc form A_g is calculated as:

$$A_{\mathrm{g}} = \frac{R^2 \theta}{2} - \frac{b(R - h_{\mathrm{d}})}{2} \tag{15}$$

$$h_{\mathrm{d}} = 2Rsin^2\frac{\theta}{4} \quad (16)$$

$$b = 2R\sin\frac{\theta}{2} \quad (17)$$

where θ is the arc central angle, and R is the inner radius of the shell. h_d and b are the height and chord length of the arc form, respectively.
The water-side pressure drop is calculated as:

$$\Delta p_{\mathrm{w}} = (\Delta p_{\mathrm{f1}} + \Delta p_{\mathrm{f2}})f_{\mathrm{s}}n_2 \quad (18)$$

$$\Delta p_{\mathrm{f1}} = Ff_{\mathrm{k}}N_{\mathrm{TC}}(N_{\mathrm{b}} + 1)\frac{\rho u_{\mathrm{k}}^2}{2} \quad (19)$$

$$\Delta p_{\mathrm{f2}} = N_{\mathrm{b}}(3.5 - \frac{2B}{D})\frac{\rho u_{\mathrm{k}}^2}{2} \quad (20)$$

$$A_{\mathrm{k}} = B(D - N_{\mathrm{TC}}D_{\mathrm{o}}) \quad (21)$$

$$f_{\mathrm{k}} = 5.0Re_{\mathrm{k}}^{-0.228} \quad (22)$$

$$Re_{\mathrm{k}} = \frac{\rho u_{\mathrm{k}}d_{\mathrm{e}}}{\mu} \quad (23)$$

$$N_{\mathrm{TC}} = 1.1(N_{\mathrm{t}})^{0.5} \quad (24)$$

where A_k is the area of section parallel to the axis and through the central line of baffle or the line nearest to the central. u_k is the average velocity of water. Δp_{f1} and Δp_{f2} are the pressure drop through tubes and that through arc-form gap. f_s is 1.15 for water. F is the correction factor of tube arrangement to pressure drop and is equal to 0.5 in this paper. f_k is the friction factor of water. N_{TC} is the number of tubes in the central line or in the line nearest to the central. N_b is the number of baffles.
The heat transfer area in the water-side is calculated as:

$$A_{\mathrm{i}} = \pi D_{\mathrm{o}}(L - m_b\sigma)N_{\mathrm{t}} \quad (25)$$

where m_b is the number of baffles in one tube pass, and σ is the baffle thickness.

2.4 Air side

For air side, the heat transfer process between the compressed air and the inner wall of the tube is by convection. For the blocked-core tube with internal wavy fins, as shown in Figure 4, the Nusselt number and the Darcy friction factor are defined as follows:

$$Nu_a = 0.058Re_a^{0.825}\left(\frac{s}{d_{\mathrm{h}}}\right)^{0.256}\left(\frac{z}{d_{\mathrm{h}}}\right)^{-0.452} \quad (26)$$

$$f_a = 40Re_a^{-0.481}\left(\frac{s}{d_{\mathrm{h}}}\right)^{0.748}\left(\frac{z}{d_{\mathrm{h}}}\right)^{-1.166} \quad (27)$$

where s is the wave height; z is the wave length; d_h is the hydraulic diameter of the inner tube. These expressions are valid for the following ranges: $904 \le Re \le 4{,}520$, $0.61 \le s/d_h \le 2.45$, $6.12 \le z/d_h \le 11.02$.
The Reynolds number of air is defined as:

$$Re_{\mathrm{a}} = \frac{\rho_{\mathrm{a}}u_{\mathrm{a}}d_h}{\mu_{\mathrm{a}}} \quad (28)$$

where ρ_a, u_a and μ_a are the density, the average velocity, and the viscosity of air, respectively.

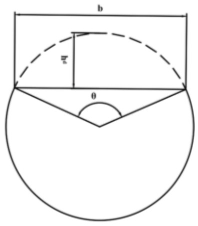


Figure 3: Schematic of baffle

The hydraulic diameter is defined as:

$$d_h = \frac{4A_c}{P} \tag{29}$$

$$A_c = \frac{\pi(D_\mathrm{i}^2 - d_\mathrm{o}^2)}{4} - l_f\delta_f \tag{30}$$

$$P = \pi(D_\mathrm{i} + d_\mathrm{o}) + 2l_f \tag{31}$$

$$l_f = \frac{\pi}{2}D_\mathrm{i} + m(D_\mathrm{i} - d_\mathrm{o}) - 2m\delta_\mathrm{f} + \pi(\frac{d_\mathrm{o}}{2} + \delta_\mathrm{f}) \tag{32}$$

where l_f and δ_f are the unfolded length and thickness of the fins, and *m* is number of internal fins in the cross section.
The air-side convective heat transfer coefficient, h_a is calculated as:

$$h_a = \frac{Nu_a d_h}{\lambda_a} \tag{33}$$

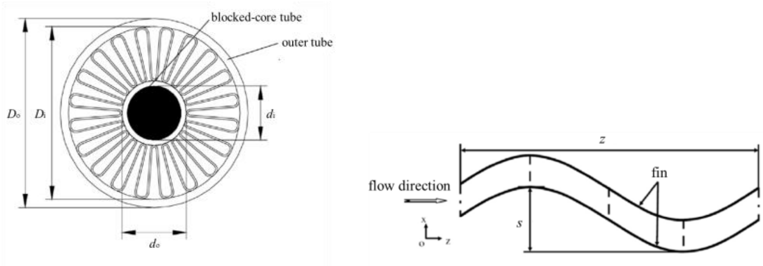
where λ_a is the thermal conductivity of air.
The pressure drop of air side is defined as:

$$\Delta P_a = \frac{fL}{2d_h}\rho_\mathrm{a}u_\mathrm{a}^2 \tag{34}$$

The heat transfer area of air side is calculated as:

$$A_\mathrm{o} = 2l_f L\frac{l_s}{z} \tag{35}$$

where l_s is the unfolded length of one period of wavy channel.



(a) Cross-sectional view (b) One period of wavy channel

Figure 4: Schematic of longitudinally internal wavy finned tube

2.5 Material properties

The physical properties of water and air are determined by their temperature and pressure. As the change of pressure was relatively small in this paper, the temperature played a key role in the physical properties. Therefore, each physical property is expressed as a polynomial function of temperature based on the properties data from the NIST chemistry webbook.

3. Results and discussion

In order to investigate the effect of heat source pressure on the thermal design, two heat exchangers were designed under 0.1 MPa and 1.0 MPa of air-side inlet pressure, respectively. $T_{a,i}$, $T_{a,o}$, $T_{w,i}$, $T_{w,o}$ and cold source pressure were constant.

As shown in Table 1, the heat exchangers designed under 0.1 MPa and 1.0 MPa are exactly the same. The results indicate that the pressure of fluid has little effect on the final design. Additionally, the pressure drop of water side of the two heat exchangers is also the same. However, the pressure drop of air side is quite different. The pressure drop under 0.1 MPa is 0.076 MPa, while that under 1.0 MPa is 0.0071 MPa. The pressure drop under 1.0 MPa just occupied 9.3 % of that under 0.1 MPa. It's understandable that density of air increases with the increase of pressure. Then the volume will get smaller when operating pressure is increased, which results in the lower pressure drop.

Table 1: Geometry and performance results under different heat source pressures

Parameter	Units	0.1 MPa	1.0 MPa
Mass flow rate of water	kg/s	8.72	8.72
Mass flow rate of air	kg/s	16	16
Total number of tubes	-	235	235
Shell inner diameter	m	0.8	0.8
Tube length	m	1.8	1.8
Central distance of tubes	m	0.045	0.045
Outer diameter of the tube	m	0.032	0.032
Inner diameter of the tube	m	0.028	0.028
Outer diameter of blocked core	m	0.008	0.008
Water-side pressure drop	MPa	0.000492	0.000492
Air-side pressure drop	MPa	0.076	0.0071
Effectiveness	-	0.40	0.40
Inlet temperature of air	K	390	390
Outlet temperature of air	K	300	300
Inlet temperature of water	K	290	290
Outlet temperature of water	K	330	330

4. Conclusions

Two heat exchangers were designed under different heat source pressures. In the two heat exchangers, $T_{a,i}$, $T_{a,o}$, $T_{w,i}$, $T_{w,o}$ and cold source pressure were identical. The thermal design results indicate that the geometry of the two heat exchangers are the same, but the air-side pressure drop under 0.1 MPa is significantly larger than that under 1.0 MPa. The results show that the pressure of fluid has little effect on the structure. This work provides valuable advice for rapid production in industry. However, to make the discoveries into real application, much work remains to be done.

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