STRUCTURE DESIGN AND THERMAL-HYDRAULIC PERFORMANCES STUDY ON TUBE-ON-SHEET HEAT EXCHANGER

by

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A numerical investigation is performed to study the thermal-hydraulic performances of tube-on-sheet heat exchanger. Especially, this paper is focused on the effects of the structure parameters (fin angles, fin lengths, fin pitches, fin widths, and tube arc radius) on the thermal-hydraulic characteristics of the heat exchanger. The heat exchanger optimized structural parameters are obtained. The results reveal that main structure parameters on thermal-hydraulic performance are fin angle, fin length, fin pitch, and fin width. Furthermore, decreasing the fin angle, fin pitch, and fin width or increasing the fin length cause thermal hydraulic performance to increase. In addition, main structure parameters on low temperature corrosion are tube arc radius and fin length. Decreasing the tube arc radius and fin length can mitigate low temperature corrosion. Furth, stainless steel is choosing as the material for the heat exchanger to reduce corrosion.

Key words: thermal-hydraulic performance, numerical simulation, elliptical tube, corrugated fin, stainless heat exchanger

Introduction

For compact and high thermal hydraulic performance, tube-on-sheet heat exchanger is important for gas-fired on-demand water heater and boiler [1, 2]. To enhance efficiency of the heat exchanger, it is necessary to lower flue temperature and decrease air coefficient. Especially under lower flue temperature, lower air coefficient and cold water temperature, low temperature corrosion must be happened, so corrosion resistant material and new structure must be selected and designed. Common material of water heater heat exchanger are copper [3], aluminum and casting aluminum-silicon alloy [4, 5], carbon steel [3, 6], stainless steel [5, 7], and polymer heat exchanger [8].

Tubes of water heater heat exchanger always are circle tube [9, 10] or elliptic tube [2, 10, 11]. The elliptic tube structure has a better aerodynamics configuration and a lower total drag force than circle tube. Jang *et al.* [12] experimentally determined that the elliptic tube heat transfer coefficient is 35-50% of the corresponding circular tube, while the drag force is only

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25-30% of the circular tube. Erek *et al.* [1] concluded that fin pitch has a considerable influence on drag force, and increasing ellipticity of the fin tube increase the heat transfer and an important reduction in pressure drop.

Many papers have revealed the relationship between thermal hydraulic characteristics and various fin structures, such as plate fin [1, 13], strip fin [14-18], wavy fin [19], louvered fin [4, 19], H-type fin, spiral fin, and slotted fin. Tao *et al.* [19] found that the heat exchanger with larger wavy angles, smaller fin pitch and tube row number, has better heat transfer performance and higher pressure drop. Kim *et al.* [4] show that the fins pitch and angle affect the thermal hydraulic performance significantly.



Figure 1. Stainless corrugated finned flat-elliptical tube heat exchanger

Model descriptions and numerical method

Physical model

Therefore, the problems to be solved in this paper can be stated: Firstly, some measures should be taken to meet the low temperature corrosion and secondly, the structure parameters of heat exchanger should be optimizing to enhance the thermal hydraulic performance.

According aforementioned questions and studies, a new type heat exchanger is designed for gas fired water heater, showed in fig. 1. In this work, the main purpose of 3-D numerical simulation is to study effects of fin length, fin angle, fin pitch, fin width, tube arc radius, especially tube arc radius and fin width, on thermal hydraulic performance, to obtain a higher efficient and a lower corrosion heat exchanger for gas-fired water heater.

Figure 2 shows a schematic diagram of one row elliptical tube and plate fin heat exchanger. Furthermore, the effects of fin structure parameters have been investigated; thermal hydraulic behavior has been predicted on gas side only. Table 1 shows the dimensions of tube and fin heat exchanger.



Figure 2. Structure parameters of corrugated finned flat-elliptical tube

Sample No.	<i>S</i> [mm]	<i>W</i> [mm]	δ [mm]	α [°]	<i>H</i> [mm]	R_1 [mm]	R_2 [mm]
1	3.32	3.98	0.2	0	13	2.9	3.5
2	3.32	3.98	0.2	10	13	2.9	3.5
3	3.32	3.98	0.2	20	13	2.9	3.5
4	3.32	3.98	0.2	30	13	2.9	3.5
5	3.32	3.98	0.2	40	13	2.9	3.5
6	4.12	3.98	0.2	10	13	2.9	3.5
7	4.92	3.98	0.2	10	13	2.9	3.5
8	5.72	3.98	0.2	10	13	2.9	3.5
9	3.32	3.98	0.2	10	12	2.9	3.5
10	3.32	3.98	0.2	10	14	2.9	3.5
11	3.32	3.98	0.2	10	15	2.9	3.5
12	3.32	3.98	0.2	10	16	2.9	3.5
13	3.32	4.18	0.2	10	13	2.9	3.5
14	3.32	4.38	0.2	10	13	2.9	3.5
15	3.32	4.58	0.2	10	13	2.9	3.5
16	3.32	3.98	0.2	10	13	2.4	3.0
17	3.32	3.98	0.2	10	13	1.9	2.5
18	3.32	3.98	0.3	10	13	1.4	2.0

Table 1. Main geometry data for simulated tube and fin heat exchangers

Governing equations and boundary conditions

For simplifying the mathematical model, some assumptions suggested in some articles [6, 18, 20], are made in the numerical simulation:

- Viscous, steady, incompressible transition turbulent flow is presented.
- Thermal radiation, viscosity heating and natural-convection are neglected.
- The working fluid properties is function of composition and temperature.
- Inner wall surface are constant temperature.
- The governing equations in the fluid domain are expressed:
- Continuity equation

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} \left(\rho u_i \right) = 0 \tag{1}$$

- Momentum equation

$$\frac{\partial}{\partial t}(\rho u_i) + \frac{\partial}{\partial x_j}(\rho u_i u_j) = -\frac{\partial p}{\partial x_i} + \frac{\partial}{\partial x_j} \left[\mu \left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} - \frac{2}{3} \delta_{ij} \frac{\partial u_l}{\partial x_l} \right) \right] + \frac{\partial}{\partial x_j} \left(-\rho \overline{u'_i u'_j} \right)$$
(2)

Energy equation

$$\frac{\partial}{\partial t}(\rho E) + \frac{\partial}{\partial x_i} \left[\rho u_i \left(\rho E + P\right)\right] = \frac{\partial}{\partial x_i} \left(k_{\rm eff} \frac{\partial T}{\partial x_i}\right)$$
(3)

– Turbulent kinetic equation

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_i}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k + G_b - \rho \varepsilon - Y_M + S_k$$
(4)

Heat conduction equation in the solid domain



Figure 3. Computation domain



simulation and experiment

The hydraulic diameter, D_h , is defined:

$$D_{h} = \frac{4\left[F_{c}H + \frac{1}{2}S\left(Wd + d^{2} - \frac{1}{8}\pi d^{2}\right)\right]}{F_{0}}$$
(6)

where *d* is tube external diameter, $d = 2R_2$.

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial v^2} + \frac{\partial^2 T}{\partial z^2} = 0$$
(5)

Figure 3 represents the computational domain with boundary conditions for tube-on-sheet heat exchanger. The figure also shows the co-ordinate system, where X is the stream wise direction, Y is the span wise direction, and Zstands for the fin pitch direction. The fluid region comprises of the flow inlet, pressure outlet and the solid region includes the tube wall and fin. The outlet zone is extended with two times of the length of the bank to prevent the backflow while the entrance zone is extended with 150 mm to keep the uniform inlet velocity. The gas inlet is specified with the massflow rate inlet or constant velocity-inlet boundary condition, the gas outlet with the pressure outlet boundary condition.

The numerical simulation is carried out using FLU-ENT, with RNG *k*- ε , SIMPLE pressure-velocity coupling algorithm, and the second upwind discretization scheme for momentum, energy, turbulent kinetic energy, and dissipation energy [6, 18, 20]. The convergence criterion is satisfied when the residuals of variables are less than $1 \cdot 10^{-6}$ except for the energy where a value of $1 \cdot 10^{-7}$ is used. Under-relaxation factors of turbulent kinetic energy, turbulent dissipation rate and turbulent viscosity are changed within the range between 0.2 and 0.4, others take the default values.

Model validation

Test section of sample No. 2 is show in fig. 1, results of simulation and test are showed in fig. 4, deviation of outlet temperature and pressure drop between are within $\pm 5\%$, so simulation models are valid.

Data reduction

To more accurately compare the thermal-hydraulic performances of tube-on-sheet heat exchanger under different structural parameters and operating conditions, some parameters are defined. The Nusselt number, Nu, is defined:

$$Nu = \frac{h_o D_h}{\lambda} = \frac{Q D_h}{F_0 \Delta T_m \lambda}$$
(7)

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where

$$\Delta T_m = \frac{\left(T_{\rm in} - T_{\rm wall}\right) - \left(T_{\rm out} - T_{\rm wall}\right)}{\ln\left(\frac{T_{\rm in} - T_{\rm wall}}{T_{\rm out} - T_{\rm wall}}\right)}$$
(8)

$$Q = mc_{p} \left(T_{\rm in} - T_{\rm out} \right) \tag{9}$$

The friction factor, *f*, is calculated:

$$f = \frac{F_c}{F_0} \frac{\rho_m}{\rho_i} \left[\frac{2\Delta P \rho_i}{G_c^2} - \left(1 + \sigma^2\right) - \left(\frac{\rho_i}{\rho_0} - 1\right) \right]$$
(10)

where

$$\sigma = \frac{F_c}{F_{\rm fr}}$$
$$F_{\rm fr} = \frac{1}{2} (d + W) S$$

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where ΔP is pressure drop, $F_{\rm fr}$ – the frontal flow area, ρ_m – the mean density, ρ_o – the outlet density, and G_c – the mass flux at minimum flow area.

The Reynolds number is defined:

$$\operatorname{Re} = \frac{\rho_i u_c D_h \frac{F_{\mathrm{fr}}}{F_c}}{\mu} \tag{11}$$

Heat transfer and flow performance ratio, θ , is defined:

$$\theta = \frac{\frac{\mathrm{Nu}}{\mathrm{Nu}_0}}{\left(\frac{f}{f_0}\right)^{1/3}} \tag{12}$$

where Nu_0 and f_0 are based on the condition of the optimum structure, which is showed as sample No. 1 in tab. 1.

Result and discussion

The details of operating conditions are list in tab. 2. In this paper, the influence of geometrical parameters, as α , S, H, R_1 , and W, on the thermal-hydraulic performances are studied by analyzing three factors, which are Nu, f, and θ .

Parameter	α	S	Н	R_1	W
Re	350-400	350-400	350-400	350-400	350-400
Mass-flow rate, \dot{m} [kgm ⁻² s ⁻¹]	0.846	0.846	0.846	0.846	0.846
Flue gas velocity [ms ⁻¹]	_	3.6	-	3.6	-
Flue gas inlet temperature, T _{flue,in} [K]	1000-1500	1000-1500	1000-1500	1000-1500	1000-1500
Inner wall temperature, T _{wall} [K]	350	350	350	350	350

Table 2. Details of operating conditions



Figure 5. Thermal-hydraulic performance with arc radius under constant inlet mass-flow rate and velocity







Figure 7. Thermal-hydraulic performance with different fin width under constant inlet mass-flow rate

Tube arc radius

The data is divided into two groups. The first is in the condition of constant inlet massflow rate, in which $\dot{m} = 0.846 \text{ kg/m}^2\text{s}$. The factors are marked Nu_m, f_m , and θ_m , respectively. The second is in the condition of constant inlet flue gas velocity, in which v = 3.6 m/s. The factors are marked Nu_v, f_v , and θ_v , respectively. To illuminate the influence of tube arc radius, four circular diameters are compared. The samples from No. 16 to 18 and two are showed in tab. 1. With circular diameter increases, elliptical width and outer tube area enlarge, whereas fin structure does not change.

From fig. 5, it can be shown that Nu_m and f_m increase rapidly, when θ_m increases slowly with R_1 increases in the first group date. It can be concluded that the exchanger with smaller R_1 has lower pressure drop and little reduction in thermal-hydraulic performance under constant mass-flow rate. In the second group date, it can be found that Nu_m and f_m increase rapidly, when θ_m decrease with R_1 increases. It suggests that the exchanger with smaller R_1 has lower pressure drop and better thermal-hydraulic performance under constant mass-flow rate. In the second group date, it can be found that Nu_m and f_m increases rapidly, when θ_m decrease with R_1 increases. It suggests that the exchanger with smaller R_1 has lower pressure drop and better thermal-hydraulic performance under constant flue gas velocity.

As shown in fig. 6, No. 2 has larger R_1 than No. 18, and the low temperature zone area at the end of the elliptical tube in No. 2 is larger than No. 18, when the lowest temperature of this area is lower than No. 18. It demonstrates that, it will generate a low temperature zone area in the larger R_1 exchanger. The low temperature zone area could occur gas condensation and it would result in metal corrosion. To obtain better thermal-hydraulic performance, smaller area of low temperature zone and lower pressure drop, smaller R_1 should be selected. To reduce corrosion, stainless steel is choosing as the material for the heat exchanger.

Fin width

To illustrate the influence of fin width, four fin widths are compared under constant inlet mass-flow rate. Samples from No. 13 to 15 and two are showed in tab. 1. With fin width increases, uncovered tube outer area does not

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change, while fin plate area increases. Fin width increases from 3.98-4.58 mm and total outer area increase 8.97%. From fig. 7, Nu, *f*, and θ decrease with the increasing fin width. Under constant inlet mass-flow rate and inner temperature, inlet velocity decreases, when outlet temperature decreases slightly and heat transfer rate increases slightly, but pressure drop decreases obviously. Smaller fin width should be selected.

Fin angle, fin pitch, and fin length

Figures 8-10 have presented the variation of Nu, f, and θ with fin angle α , fin pitch S, and fin length H. According to analyzing the curves, three conclusions are verified which had been concluded in [21]. First, with smaller α , as $\alpha = 0 \sim 20^\circ$, the heat exchanger has better thermal-hydraulic performances and lower pressure drop. Second, with smaller S and H, it has better thermal-hydraulic performance, but pressure drop becomes larger. Third, with larger Hthe low temperature of the outlet might lead to gas condensation.



Figure 8. Thermal-hydraulic performance with different fin angles



Figure 9. Thermal-hydraulic performance with different fin pitch



Figure 10. Thermal-hydraulic performance with different fin length

Conclusions

In this study, thermal-hydraulic performances of tube-on sheet heat exchanger are investigated numerically. The conclusions can be summarized as follows.

- Main structure parameters of tube-on sheet heat exchanger on thermal-hydraulic performances are fin angle, fin length, fin pitch and fin width. Smaller fin angle, pitch and width, higher thermal hydraulic performance. Higher fin length, higher thermal performance.
- Main structure parameters on low temperature corrosion are tube arc radius and fin length. Smaller arc radius and smaller fin length should be selected for reducing low temperature corrosion.
- To reduce corrosion, stainless steel is choosing as the material for the heat exchanger.

The numerical data for thermal-hydraulic performance are validated by test, which can be helpful to engineering applications of heat exchanger for gas-fired water heater and others.

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Nomenclature

- d tube external diameter, [mm]
- D_h hydraulic diameter, [m]
- G_c mass flux at minimum flow area, [kgm⁻²s⁻¹]
- H fin length, [mm]
- h_0 heat transfer coefficient F_c – minimum flow area, [m²]
- F_0 total outside superficial area, [m²]
- f friction factor f_0 -f based on tube without any fin
- ΔT_m log-mean temperature difference, [K] $T_{\rm flue,in}$ – inlet temperature of flue gas, [K]
- T_{wall} inner wall temperature of tube, [K]
- u_c velocity at the minimum cross-sectional area, $[ms^{-1}]$ Nu – Nusselt number
- Nu₀ Nu based on tube without any fin
- ΔP pressure drop, [Pa]
- Q total heat transfer rate, [W]

- R_1 tube inner arc radius
- R_2 tube outer arc radius
- Re Reynolds number
- S = fin pitch, [mm]
- \tilde{W} fin width, [mm]

Greek symbols

- α fin angle, [°]
- fin thickness, [mm] δ
- θ - fin coefficient
- λ thermal conductivity, [Wm⁻¹K⁻¹]
- μ dynamic viscosity, [kgm⁻¹s⁻¹]
- ρ density, [kgm⁻³]
- ρ_i inlet density, (kgm⁻³)
- ρ_m mean density, [kgm⁻³]
- $\rho_{\rm o}$ outlet density, [kgm⁻³]

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