Flow field simulation of the spiral-wound heat exchanger

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Abstract. The flow conditions and heat transfer processes in the spiral-wound heat exchanger were studied. The flow characteristics at different sections were also analysed. Numerical calculation results demonstrate that the velocity gradient of the fluid on the shell-side of the heat exchanger increases with the inlet flow velocity increasing, and the increased velocity gradient enhances the turbulence of the shell side fluid.

1. Introduction

As a new type of highly efficient and compact heat exchanger, the spiral-wound heat exchanger fits the development trend of heat exchange equipment well. The spiral-wound heat exchanger also has many advantages over other heat exchangers. For instance the tubes twisted on the center cylinder, and the parts of the exchanger such as spacer and pipe clamp affect the flow state of the shell-side fluid. In the meantime, due to the special spiral winding structure, the fluid in the tube forms a strong secondary flow, which greatly improves the efficiency of the heat transfer. In order to investigate the heat transfer coefficient on the shell-side of the spiral-wound heat exchanger, Sampour [1] established an experimental model of the coiled tube heat exchanger, and calculate the heat transfer coefficient on the shell side of the heat exchanger by the "Wilson plot" method. Srbislav et al. [2] studied the effect of structural parameters such as pitch and winding angle on Nussle number Nu. The following relationship has been obtained:

\[ Nu = 0.5 Re^{0.55} Pr^{1/3} (\eta/\eta_w)^{0.14}, \, 1000<Re<9000, \, 2.6<Pr<6.0 \]

Ma et al. [3] implemented a test for the structural parameters of the spiral wound tube shell-side fluid. The weighted order of the parameters is \( Re > \eta > Nu > Pr > P1 > P2 > D \). Nephron et al. [4] established a spiral-wound heat exchanger model which has six-layer coiled tubes with five-wraps. The numerical calculation was also carried out based on the experimental result. Subsequently, an experimental model of finned spiral-wound heat exchanger was established and compared with non-finned spiral-wound heat exchanger. The results show that the finned heat exchanger has higher heat transfer efficiency. Pan et al. [5] studied the relationship among the oscillating flow, heat transfer, and pressure drop in the winding pipe by computational fluid dynamics. The relationship among oscillation frequency, inlet velocity, Nussle number and drag coefficient was also analyzed according to the simulation results. Nerada’s et al. [6] established an experimental model of the spiral-wound LNG heat exchanger, the heat-transfer coefficients and pressure drop for gas flow at the shell-side of heat exchanger were measured. Deng et al. [7] established a new type of corrugated tube heat exchanger, and compared the heat transfer with two different types of heat exchangers through experiments and
simulations. Hammed et al. [8] implemented an experimental test on spiral wound tube heat exchangers and studied the effects of flow parameters, thermodynamic parameters, and geometric parameters.

2. Numerical Approach

2.1. Geometrical configuration

The main parameters of the heat exchange tube include winding radius, pitch, radius of the heat exchange tube, and the number of turns, the heat transfer inner tube is shown in Fig 1.

Fig 1. Heat transfer tube structure of the wound tube heat exchanger.

The calculation model of a three-dimensional spiral-wound heat exchanger was established by SolidWorks 2016. The length of the wended pipe bundle is 600mm. It is composed of two layers of six wending pipes, including three inner and outer layers.

Fig 2. The schematic diagram of shell model of spiral-wound heat exchanger.

2.2. Mathematical model

2.2.1. Basic equation. Computational fluid dynamics (CFD) is used to solve fluid mechanics based on control equations, including continuity equation, momentum equation and energy equation. The turbulent state of fluid occurred in the spiral-wound tube heat exchanger, and the control equations are shown as follow [9, 10]:

Continuity equation:

$$\frac{\partial \rho}{\partial t} + \frac{\partial (\rho u_x)}{\partial x} + \frac{\partial (\rho u_y)}{\partial y} + \frac{\partial (\rho u_z)}{\partial z} = 0$$  \hspace{1cm} (1)

Momentum equation:

$$\rho \frac{Du_x}{D\theta} = pX - \frac{\partial P}{\partial x} + \mu \left( \frac{\partial^2 u_x}{\partial x^2} + \frac{\partial^2 u_x}{\partial y^2} + \frac{\partial^2 u_x}{\partial z^2} \right) + \frac{\mu}{3} \left( \frac{\partial u_x}{\partial x} + \frac{\partial u_y}{\partial y} + \frac{\partial u_z}{\partial z} \right)$$  \hspace{1cm} (2)

$$\rho \frac{Du_y}{D\theta} = pY - \frac{\partial P}{\partial y} + \mu \left( \frac{\partial^2 u_y}{\partial x^2} + \frac{\partial^2 u_y}{\partial y^2} + \frac{\partial^2 u_y}{\partial z^2} \right) + \frac{\mu}{3} \left( \frac{\partial u_x}{\partial x} + \frac{\partial u_y}{\partial y} + \frac{\partial u_z}{\partial z} \right)$$  \hspace{1cm} (3)
Energy equation:

$$\frac{\partial (\rho E)}{\partial t} = \nabla \cdot \left[ \left( \rho \left( E + p \right) \right) \nabla T \right] - \nabla \cdot \left( \sum \sigma_f J_f + \left( \tau_{eff} \cdot \nabla \right) \right) + S_h \tag{4}$$

2.2.2. Turbulence model. Take the notational flow into consideration, the curved streamlines and so on were revised by the RNG k-ε model. Better results were obtained for complex flow phenomena such as secondary flow, swirling flow and so on. The equations of k and ε are given respectively [10]:

$$\frac{\partial (\rho k)}{\partial t} + \frac{\partial (\rho ku)}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_i} \right] + G_k + G_b - \rho \varepsilon - Y_m \tag{5}$$

$$\frac{\partial (\rho \varepsilon)}{\partial t} + \frac{\partial (\rho \varepsilon u_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \left( \mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_i} \right] + C_{\varepsilon}^* \frac{\varepsilon}{k} \left( G_k + C_{\gamma} G_b \right) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} - R \tag{6}$$

2.2.3. Wall functions. In view of the functions and models applied in this study, the scale wall function is chosen to avoid the discontinuity of the simulation results caused by the small boundary layer grid.

2.3. Grid and operating conditions

Based on the Workbench software, the geometry model was imported into Meshing function and generated meshes automatically.

Boundary conditions including the velocity inlet, the pressure outlet and the non-slip wall boundary were chosen. The SIMPLE (Semi-Implicit Method for Pressure Linked Equations) algorithm and the first-order upwind scheme were used to solve the pressure-velocity coupling problem.

3. Results and Discussions

The working medium of the heat exchanger is water with 20°C, the velocities of the inlet fluid were set at 0.1m/s, 0.3m/s, 0.5m/s and 0.7m/s, respectively.

3.1. The Velocity distribution simulation of X-Y shell section under different inlet velocity

Fig. 3 shows the velocity distribution of the shell-side fluid in the heat exchanger at X-Y cross-section under different inlet velocities. It can be seen from the figure that there are low velocity regions below the tubes of the heat exchanger. The low velocity regions are not conducive to heat transfer. With the increase of the inlet flow velocity, the area of the low velocity regions gradually decreases. There are high velocity regions between the heat exchanger tubes and the outer heat exchanger wall. Because of the linear motion of the fluid flow in the high velocity regions, it also has negative impact on heat transfer. There exists radial flow from the outer high velocity region to the winding pipes. The radial flow helps to promote the heat exchange between the outer cold fluid and the heat exchanger tube wall. With the increase of the inlet flow velocity, the radial velocity also increases. From Fig. 3A, it can be seen that when the inlet velocity is 0.1m/s, the radial velocity is about 0.15m/s. From Fig. 3B, it can be seen that when the inlet velocity is 0.7m/s, the radial velocity is about 1.2m/s.

3.2. The flow field simulation of X-Z cross section under different inlet velocities

3.2.1. The analysis on velocity field. Fig. 5A, 5C and 5E shows the velocity distribution of X-Z section at y=120mm under different inlet flow rates. Fig. 5B, 5D and 5F are the velocity distribution of X-Z section at y=600mm under different inlet flow rates. It can be seen that there are velocity boundary layers and low velocity zone (lower than 0.5m/s) near the heat exchange tube. Because of the turbulent
effect, the velocity distribution in the outlet section, which has high velocity region, is more complex than that in the inlet section.

3.2.2. Temperature distribution. Fig. 6A, 6C and 6E shows the temperature distribution of X-Z section at y=120mm under different inlet flow rates. Fig. 6B, 6D and 6E are the temperature distribution of X-Z section at y=600mm under different inlet flow rates. The cold water was set to flow in the shell side while the hot water was set to flow in the tube side. Compared A with C and E, it can see that when the inlet velocity becomes smaller, the cold fluid stays longer in the shell side, and mixes with the hot fluid well. It can be observed that the area of low temperature region is smaller from Fig. 6B. From Fig. 6D and 6F, it can be seen that there are more and larger low temperature regions.

![Fig 3. Velocity distribution of X-Y section under different inlet velocity.](image)

![Fig 4. Velocity distribution of X-Z section at different inlet velocity.](image)
Fig 5. Temperature distribution of X-Z section at different inlet velocity.

4. Conclusions
In this paper, the flow conditions and heat transfer processes in the shell-side of a spiral-wound heat exchanger were investigated numerically. The velocity and temperature distributions of different sections were analyzed.

On the one hand, the flow field distribution of shell side X-Z section under different inlet velocity were analyzed. With the increase of the inlet flow velocity, the velocity gradient in the shell increases. The flow velocity also fluctuates automatically, and it is beneficial to heat transfer. There is a radial flow from the high speed zone of the outer layer to the winding pipes. The radial flow promotes the heat exchange between the outer layer of cold fluid and the wall of the tubes. With the increase of inlet velocity, the radial flow also increases. For the temperature distributions, the high-temperature zones and the low-temperature zones are distributed in, which promotes the heat transfer. As the inlet flow velocity increases, the shell-side temperature gradient decreases gradually.

On the other hand, the flow distribution of shell-side X-Z sections under same inlet velocity was analyzed. The results show that: With the increase of the inlet velocity, the area of the high velocity zone on the shell increases gradually. The temperature increases with the fluid flowing, through the heat transfer tube. The high and low temperature zones are distributed irregularly. The high temperature zone diffuses from the heat transfer tube to the surrounding regions. On the shell side flow field, the fluid is strongly affected by the winding tube, and the heat transfer is improved.

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