

# *Flow Field Simulation of the Spiral Plate Heat Exchanger*

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**Abstract:** The flow field and the heat transfer of spiral plate heat exchanger were studied. The distributions of temperature, pressure and velocity with different flow velocities were obtained. The numerical simulation results show that the temperature difference between the inlet and outlet of the equipment becomes smaller with the increase of velocity besides the pressure drop and the turbulence intensity increases with velocity increasing.

## 1. Introduction

The spiral plate heat exchanger was firstly represented by Rosemblad. It has the advantages of compact structure, high heat transfer coefficient, strong self-cleaning ability, low cost of equipment, small floor area, and so on. Liu et al. established a three-dimensional mathematical model of flow field and heat transfer in a spiral plate heat exchanger by using computational fluid dynamics and numerical heat transfer method. The influence of flow velocity, flow direction, flow state, heat exchanger height, channel spacing, and other flow and structural parameters on heat transfer coefficient, pressure drop and heat transfer-pressure drop performance coefficient  $E_k$  were analyzed. Sathiyam et al. predicted the heat transfer coefficient of the liquid-liquid two phase spiral plate heat exchanger based on the correlation. The experiment was carried out under different velocity and temperature conditions, and the velocity of the thermal fluid kept constant. The prediction results show that the variation range is less than 12% in the laminar flow. Lin et al. applied the computational fluid dynamics and numerical heat transfer method to simulate the heat transfer characteristics of high viscosity media in spiral plate heat exchangers. Rangasamy RAJAVEL studied the heat transfer and flow characteristics of the spiral plate heat exchanger by means of experiment and numerical simulation. The effects of the geometric structure and fluid properties of the spiral plate heat exchanger on the heat transfer characteristics were studied. Three kinds of spiral plate heat exchangers with different plate spacing were designed, assembled, and tested. The physical model under different flow conditions was tested, and the influence of mass flow rate and Reynolds number on heat transfer coefficient was obtained. And the Nusselt number obtained by numerical simulation was compared with the experimental data. Fax and Mills use the Lagrange multiplier method to optimize the heat exchangers with constraints, which require that the target

function is differentiable within the specified range, and the total number of constraints is less than the total number of variables.

## 2. Numerical method

### 2.1 Physical configuration.

The main components of the spiral plate heat exchangers include spiral plates, fixed distance columns, central baffles, and end covers. The heat transfer plate consists of two long strip metal plates welded on both sides of the central baffle and rolled into a pair of concentric spiral passages. The two ends of the channel are staggered and closed, and the nozzles of the inlet and outlet are installed at the side and the spiral center of the two channels, respectively. The medium flows through the channel and carries out heat exchange. The geometric model of the spiral plate is shown in Fig.1. In view of the complexity of the structure of the heat exchanger, reasonable simplification is made to ignore the influence of the inlet and outlet of the outer side of the heat exchanger. The flow passage section is directly used as the inlet and outlet. The specific size parameters of the spiral plate heat exchanger used in the simulation calculation are listed in Table 1. The geometric model of the spiral plate heat exchanger is shown in Fig. 2.

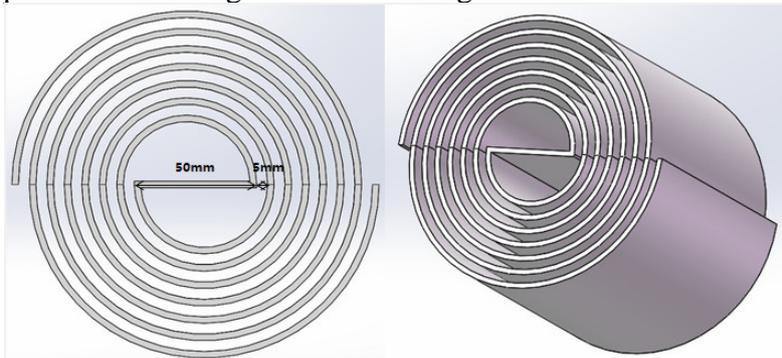


Fig 1 Geometric model of spiral plates

Table 1 Structure parameters of Spiral plate heat exchanger

Parameter name	numerical value	unit
Heat exchanger height	150	mm
External diameter	160	mm
Base circle diameter	50	mm
Channel width	9	mm
Plate thickness	3	mm

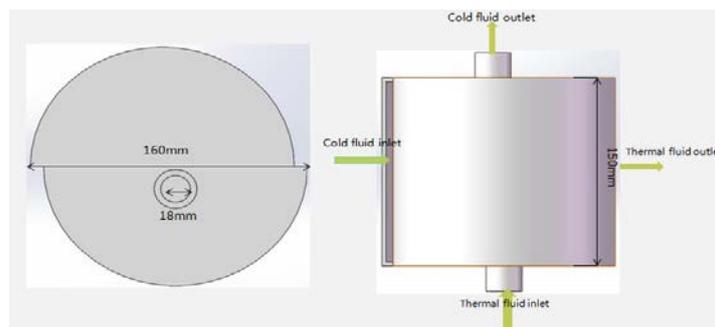


Fig 2 The geometric model of spiral plate heat exchanger

## 2.2 Governing equation.

The governing equations of fluid flow and heat transfer in spiral plate heat exchangers include continuity equation, momentum equation, energy equation and RNG  $k - \varepsilon$  equation, which are used to solve velocity, pressure and temperature variables. The governing equations are discretized by the second-order upwind difference scheme and coupled by the SIMPLE algorithm.

### 2.2.1 Basic equation.

When describing the flow field and heat transfer in the spiral plate heat exchanger, the continuity equation, energy equation and momentum conservation equation are used. These equations are expressed in rectangular coordinates when the equations are as follows:

Continuity equation:

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

Energy equation:

$$\frac{\partial(\rho E)}{\partial t} = \nabla \cdot [\bar{u}(\rho E + p)] = \nabla \cdot [k_{eff} \nabla T - \sum_j h_j J_j + (\tau_{eff} \cdot \bar{u})] + S_h \quad (2)$$

Momentum equation:

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

$$\rho \frac{Du_y}{D\theta} = \rho Y - \frac{\partial P}{\partial y} + \mu \left( \frac{\partial^2 u_y}{\partial x^2} + \frac{\partial^2 u_y}{\partial y^2} + \frac{\partial^2 u_y}{\partial z^2} \right) + \frac{\mu}{3} \frac{\partial}{\partial y} \left( \frac{\partial u_x}{\partial x} + \frac{\partial u_y}{\partial y} + \frac{\partial u_z}{\partial z} \right) \quad (4)$$

### 2.2.2 Turbulence model.

The flow in spiral plate heat exchanger is a turbulent flow accompanied by convection and conduction heat transfer. The RNG  $k - \varepsilon$  double equation model of the simulated rotating flow is simulated by Yakhot and Orzag.

The  $k$  equation and the epsilon equation can be expressed as:

$k$  equation:

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j} \left[ \alpha_k \mu_{eff} \frac{\partial k}{\partial x_j} \right] + G_k + G_b - \rho \varepsilon - S_k \quad (5)$$

Epsilon equation:

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_j} \left[ \alpha_k \mu_{eff} \frac{\partial \varepsilon}{\partial x_j} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} (G_k + C_{3\varepsilon} G_b) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} - R_\varepsilon + S_\varepsilon \quad (6)$$

### 2.2.3 Grid and operating condition.

The hexahedron structured meshes are used for the spiral plate heat exchangers. The total number of grids is about 8577000. In the simulation system, the hot fluid is 323k water and the cold

fluid temperature is 293k water. The cold fluid enters from the side and the hot fluid enters from the middle. The velocities of the inlet fluid were set at 0.4m/s, 0.6m/s, 0.8m/s, 1.0m/s and 1.2m/s, respectively.

### 3. Results and discussion

#### 3.1 Simulation results of temperature field.

As shown in Fig.3, the simulation results show that the temperature distributions of X-Y section at  $z=60\text{mm}$  with velocities of 0.4m/s, 0.6m/s, 0.8m/s, 1.0m/s and 1.2m/s, respectively. As can be seen from the figure, the heat transfer in the middle part of the heat exchanger is more intense and a more obvious change in temperature gradient is shown, which means that the heat transfer efficiency of the spiral plate heat exchanger is the highest in the middle part. Fig. 4 shows the temperature difference between the inlet and outlet of the thermal flow side with different velocities. With the increase of inlet velocity, the temperature difference of the fluid decreases. The higher the inlet velocity, the shorter the time the fluid stays in the heat exchanger, moreover, the contact time with the heat exchanger plate, and the heat transfer becomes weaker.

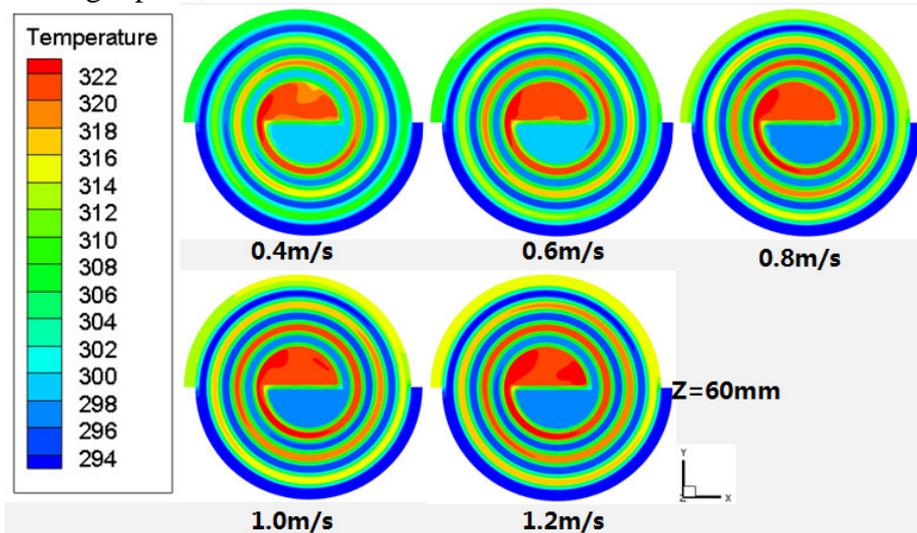


Fig 3 Simulation results of temperature field distribution in X-Y section

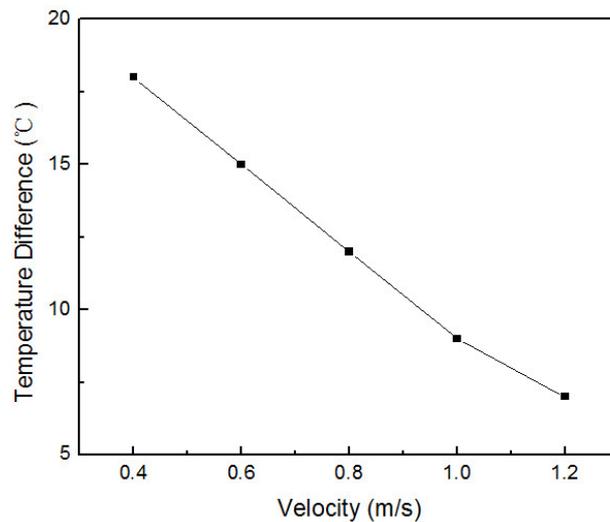


Fig 4 Influence of velocity variation on inlet and outlet temperature difference

### 3.2 Simulation results of pressure field distribution.

Fig. 5 shows the pressure distribution at the X-Y section  $Z=60\text{mm}$  with velocities of  $0.4\text{m/s}$ ,  $0.6\text{m/s}$ ,  $0.8\text{m/s}$ ,  $1.0\text{m/s}$  and  $1.2\text{m/s}$ , respectively. It can be seen from the figure that the pressure gradient in the heat exchanger is relatively uniform. Because of the continuous transition of the curvature of the spiral passage, the resistance in the passage is relatively small, and there is no sudden change in the pressure in the heat exchanger passage. With the increase of flow velocity, the greater the resistance, the greater the pressure loss.

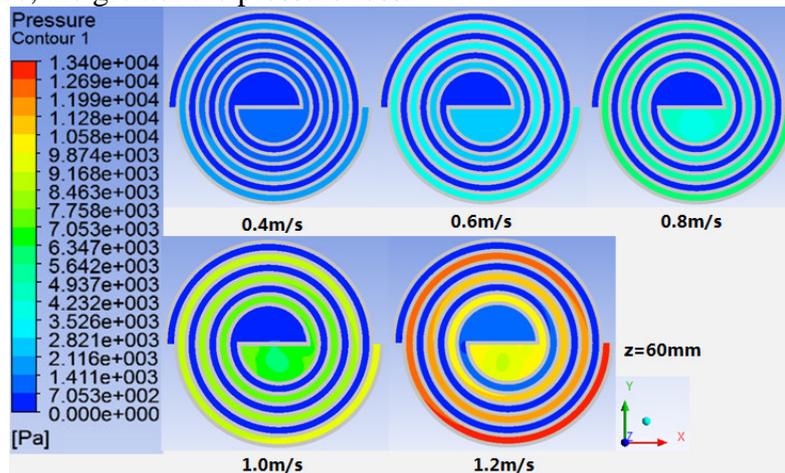


Fig 5 Simulation results of pressure field distribution in X-Y section

### 3.3 Simulation results of velocity field distribution.

Fig. 6 shows the velocity distribution at Z-X section  $y=-20\text{mm}$  and  $y=20\text{mm}$  at different inlet velocities. Fig. 6a and 6b are velocity profiles of Z-X section at inlet velocity of  $0.8\text{m/s}$  respectively. Fig. 6c, 6d, and 6e,6f are the velocity profiles of Z-X cross sections at the inlet velocity of  $1.0\text{m/s}$  and  $1.2\text{m/s}$  respectively. As can be seen from the above velocity profiles, the velocity of hot fluids is less than that of cold fluids. There is a large low speed zone (lower than  $0.6\text{m/s}$ ) in the middle part of the heat exchanger. It can be clearly seen from the comparison in Fig. 6a, 6c, 6e and Fig. 6b, 6d, 6f that the increase of inlet velocity causes more high-speed zones (the speed is above  $1.0\text{m/s}$ ) in the heat exchanger and increases the turbulence flow rate. Besides the increasing velocity improves the heat transfer efficiency of the heat exchanger.

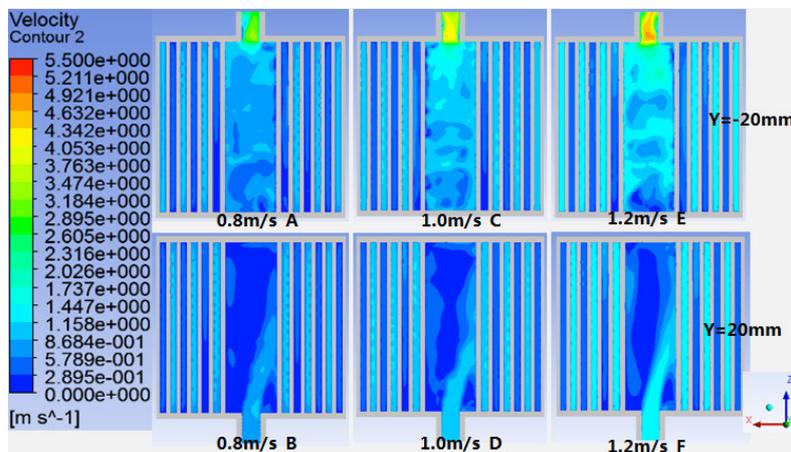


Fig 6 Simulation results of velocity field distribution in z-x section

## 4. Conclusion

In this paper, a mathematical model of flow field and heat transfer in a spiral plate heat exchanger was established, and the distributions of temperature, pressure and flow field in the heat exchanger was simulated.

(1) The heat transfer efficiency of the heat exchanger is highest in the middle part, because of the large temperature difference but also the high curvature of the spiral passage in the middle part, and the high turbulent flow in the central passage.

(2) The velocity of the hot fluid varies slightly, and it smaller than that of the cold fluid. This is because the inlet of the hot fluid is perpendicular to the axis of the cylinder, and the fluid will turn 90 degrees when it enters the spiral passage through the nozzle, and the flow resistance will be larger when the flow direction changes abruptly. Cold fluids enter to the heat exchanger tangentially and there is no sudden change in flow direction when the fluid enters the channel from the side, so that the resistance is small.

(3) With the increase of fluid velocity, the dwell time of fluid that in heat exchanger is shortened, the time of fluid heat transfer are shortened. Also, the temperature difference between inlet and outlet of fluid decreases with the increase of flow velocity. The increase of flow velocity will lead to the increase of flow resistance and the increase of pressure drop. The increase of flow velocity results in an increase of the high speed area in the heat exchanger and a higher degree of turbulence, which are conducive to heat transfer.

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