Performance Simulation of Plate Heat Exchanger Based on ANSYS ICEM

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Abstract: A three-dimensional geometric model was established by ANSYS ICEM according to the structure of a plate heat exchanger. And the thermodynamic performance under different working conditions is simulated. The pressure field, temperature field, velocity field and flow pattern were analyzed based the simulated results. The heat transfer performance and resistance performance of heat exchanger under different flow rates and inlet temperature difference of cold and hot fluids are compared and analyzed. The results provide support for the more efficient use of plate heat exchangers under different working conditions.

1. Introduction
Plate heat exchanger is a common heat exchanger in marine desalination equipment for its compact structure, high heat transfer coefficient and easy maintenance. The efficiency of desalination equipment is affected by the performance of plate heat exchanger. It is important to study the performance of heat exchanger for improving the efficiency of seawater desalination equipment[1].

Heat transfer coefficient is the main technical index to evaluate the performance of heat exchanger. And the Nusselt (NU) number is usually used to evaluate the heat transfer coefficient. The bigger NU is, the more obvious the convective heat transfer effect is. The larger the heat transfer coefficient and the smaller the resistance coefficient, and the better the heat transfer performance. However, in general, the heat exchanger with large heat transfer coefficient, its resistance coefficient is generally greater. In order to meet the requirements of practical application, it is necessary to seek the balance between heat transfer coefficient and resistance coefficient. And it is important to analyze the performance of the heat exchanger under different working conditions to optimize the design and improve the efficiency of heat exchanger. The study of plate heat exchanger performance has been done by some scholars, and some theoretical advice of heat exchanger optimization design is provided[2-6]. It is necessary to find an efficient and feasible method to study the internal flow and heat transfer of heat exchanger because of its complicated flow channel and high experimental cost. In this paper, the flow and heat transfer characteristics of plate heat exchanger under different operating conditions are simulated and analyzed, which provides theoretical suggestions for the performance improvement and design of plate heat exchanger.
2. Simulation Model

2.1 Geometric model and Mesh Generation
Plate heat exchanger is generally made up of a number of corrugated plates, it will consume a huge amount of computing resources to simulate the whole heat exchanger, and the computing efficiency is low. In order to make the simulation of plate heat exchanger simple and convenient, only one of the flow channels of plate heat exchanger is simulated in this paper. The comprehensive heat transfer performance of plate heat exchanger under different working conditions is studied by the simulation. The parameters of plate heat exchanger used in the simulation are shown in table 1.

| plate form | Plate length (mm) | Plate width (mm) | Corrugated depth (mm) | Corrugated pitch (mm) | Ripple dip(°) |
|------------|-------------------|------------------|-----------------------|-----------------------|---------------|
| W          | 120               | 80               | 4                     | 14                    | 60            |

A geometric model of single flow passage of plate heat exchanger is established by using ANSYS ICEM. The unstructured tetrahedral Mesh is used to mesh the fluid model in this paper for the complex structure of the fluid flow in the herringbone plate heat exchanger and the contact points between the upper and lower plates, and the structured grid method is not suitable. Which is shown in figure 1. The total number of grid is about 670,000, and the grid quality of the whole model is found to be good and accord with the requirement of calculation through the grid checking tool of ICEM and FLUENT.

2.2 governing equations and boundary conditions
Mass conservation, momentum conservation and conservation of energy must be observed in heat exchanger simulation. The governing equations are as follows:

\[ \frac{\partial p}{\partial t} + \nabla (\rho \vec{v}) = 0 \]  
\[ \frac{\partial}{\partial t} (\rho \vec{v}) + \nabla (\rho \vec{v} \vec{v}) = -\nabla p + \nabla (\vec{f}) + \rho \vec{g} + \vec{F} \]  
\[ \frac{\partial}{\partial t} (\rho E) + \nabla (\vec{v}(\rho E + p)) = -\nabla (\sum h_j j_j) + S_h \]

The turbulent flow in plate heat exchanger is considered as turbulent flow. RNG k-ε model is used to simulate the turbulent flow in plate heat exchanger. The two equations are as follows[7]:

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

The boundary conditions are set as shown in Table 2. The medium is water, the inlet is velocity inlet, the inlet temperature of cold water is set at 300K; the outlet is pressure outlet, and the outlet static pressure is set at atmosphere. The upper and lower surfaces are set as non-slip constant temperature conditions, and the lateral surfaces are set as non-slip adiabatic wall conditions. The heat effect due to viscous dissipation and the buoyancy force due to gravity and density difference are neglected in the calculation. The SIMPLE algorithm is used to calculate the velocity field and the pressure field, and the second order upwind scheme is used to improve the accuracy of the results.
2.3 simulation settings
In order to analyze the working characteristics of plate heat exchanger under different working conditions, 20 different working conditions of plate heat exchanger were simulated. The specific operating mode settings are shown in table 3.

| Temperature difference | Flow rate | 20℃ | 30℃ | 40℃ | 50℃ | 60℃ |
|------------------------|-----------|-----|-----|-----|-----|-----|
| 55 L/h                 |           | √   | √   | √   | √   | √   |
| 110 L/h                |           | √   | √   | √   | √   | √   |
| 192 L/h                |           | √   | √   | √   | √   | √   |
| 275 L/h                |           | √   | √   | √   | √   | √   |

3. Numerical simulation and analysis

3.1 Numerical simulation
The pressure distribution on the whole and on different sections is shown in Figure 2. The results show that there is pressure loss when the fluid flows through the plate heat exchanger. And there is a pressure gradient along the direction of fluid flow, while the pressure gradient perpendicular to the direction of flow is very small.

The velocity distribution in the middle plane and in different sections is shown in Figure 3. As can be seen from the diagram, the velocity distribution is periodic, and the velocity of fluid is obviously decreased behind the contact point of the plate heat exchanger, and the maximum velocity is located in the middle of the contact point. There exist both maximum and minimum velocity in the same region, and the non-uniform distribution of the velocity increases the disturbance of the fluid and makes the temperature distribution of the fluid in the same region more uniform, the heat exchanger has better heat transfer performance.

The temperature distribution in different cross-sections is shown in Figure 4. When the cold fluid flows through the plate heat exchanger, its temperature increases gradually. And the temperature near the contact is the highest, approaching the temperature of hot fluid. This is due to contact closer to the plate heat exchanger wall, and contact near the cold fluid flow rate is slow, heat transfer is more adequate. But in general, the temperature gradient along the direction of fluid flow is large, while the temperature gradient perpendicular to the direction of fluid flow is small.

![Pressure distribution](image)
3.2 Performance analysis of heat exchanger

Fig. 5 shows the variation of the temperature difference between the inlet and outlet of the cold fluid with the temperature difference between the hot and cold fluid under different flow velocity of cold fluid. The results show that the smaller the flow velocity of the cold fluid, the larger the temperature difference between the inlet and outlet of the cold fluid. In other words, the slower the cold fluid velocity, the more adequate heat transfer, the cold fluid is heated at a higher temperature. At the same time, in order to make the cold fluid be heated to a certain temperature, the temperature difference between the cold fluid and the hot fluid can be increased.

Fig. 6 shows the variation of NU with heat transfer temperature difference under different operating conditions. The results show that the NU increases with the increase of temperature difference between hot and cold fluids. The NU of cold fluids with flow velocity of 55 L/H, 110 L/H, 192 L/H and 275 L/h increased by 28.9%, 29.6%, 25.6% and 28.3% respectively when the inlet temperature difference of cold and hot fluids increased from 20 °C to 60 °C. This shows that the larger the temperature difference between hot and cold fluid inlet, the better heat transfer performance. It can be achieved by increasing the flow capacity of cold fluid and increasing the temperature difference between cold and hot fluid to ensure a larger heat transfer coefficient.
A fitting formula of the average Nusselt number of plate heat exchanger when the working fluid on both sides of the exchanger is respectively hot and cold water is obtained by Das\cite{8}:

$$Nu = 0.317 \cdot Re^{0.703} \cdot Pr^{1/3} \quad (6)$$

The simulated Nusselt number is compared with the predicted value according to formula (6). As you can see in figure 7, NU's simulation is smaller than predicted, but the overall deviation is between 0% and 20%. The larger the temperature difference between cold and hot fluids, the closer the simulation value of NU is to the predicted value; And the smaller the temperature difference between cold and hot fluids, the more the simulation value of NU deviates from the predicted value. The possible reason for the error is that formula (6) shows the fitting relation of the average NU number of the plate heat exchanger, which cannot be accurately predicted. And there are some errors between simulation and experiment also. But this deviation is within the engineering acceptable range, which shows that the simulation results are reliable.

The pressure difference between the inlet and outlet of the plate heat exchanger is the key factor to affect its performance. The simulation results show that the simulated pressure drop is basically the same under the same flow capacity of cold fluid and different temperature difference between cold and hot fluid. The reason is that the pressure drop in the plate heat exchanger is mainly caused by the pressure loss in the plate heat exchanger. The viscosity coefficient of water is different at different temperatures. The resistance to the flow of fluid through the plate heat exchanger is relatively small for the viscosity coefficient of water is relatively small. The difference of pressure loss caused by different dynamic viscosity of water due to different temperature is negligible.

The relationship of NU number and pressure loss with Re number is shown in Fig.8. It can be found that the NU number and pressure loss $\Delta p$ both increase with the increase of Re number. This means that the performance of heat exchanger is positively correlated with the flow capacity of the cold fluid, but the increase of the pressure loss means that more mechanical work is consumed.
4. Conclusion

The results of the study show that the NU number and the pressure difference between the inlet and outlet of the cold fluid will increase with the increase of Re. It is necessary to take into account the increase of pressure loss and consume more mechanical work when using the heat exchanger; the larger the flow capacity of the cold fluid and the greater the NU number, the better the performance of heat exchanger. It can be achieved by increasing the flow rate of cold fluid and increasing the temperature difference between cold and hot fluid to ensure a larger heat transfer coefficient.

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