Analysis of the influence of the tubes constrained structure of contra-flow vertical shell and tube heat exchanger on heat transfer ability with natural circulation

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Abstract. In order to verify the effect of the combined restraint form of straight strip and corrugated strip on the heat transfer capacity of shell-tube heat exchanger with natural circulation. The physical model and mathematical model of the local heat exchange section of the heat exchanger are established by using ICEM and Fluent to carry out numerical simulation. The calculation results of three-dimensional flow field and temperature field of the heat exchange section are solved and analysed to obtain the resistance characteristics of the tube bundle of shell-and-tube heat exchanger under the restriction of corrugated strip and straight strip with natural circulation and the influence at the temperature field.

1. Introduction

The primary and secondary side fluid of shell-tube heat exchanger maintains at a low flow rate with natural circulation, and the driving pressure head of the fluid comes from the density difference between the inlet and outlet media of the heat exchanger. The whole heat transfer process is a coupling process of the temperature, flow and pressure of the shell side fluid and the tube side fluid. The heat exchange system is affected by the resistance significantly, so the form drag of the equipment is strictly limited in the design process of the heat exchanger used in the natural circulation system. The restraint of tube bundle is an indispensable part in the design of heat exchanger, and it is also the main part of shell side resistance of shell-tube heat exchanger.

The tube bundle of contra-flow vertical shell and tube heat exchanger adopts a constraint structure composed of corrugated strip and straight strip to maintain the tube spacing and shell side flow passage, and to prevent flow induced vibration of heat exchanger. The resistance of the shell side is mainly composed of the shape resistance of the inlet and outlet windows and corrugated strips and the resistance along the tube bundle. This paper mainly studies the local resistance characteristics of corrugated strip and straight strip composite structure and its influence on heat transfer performance, and analyzes whether the design of the structure meets the requirements.

2. Physical model

The tube bundle of heat exchanger is composed of 285 heat transfer tubes, which are divided into five layers. The number of heat exchange tubes decreases layer by layer from the outside to the inside. There are 69 tubes in the outermost layer, with a difference of 6 tubes in each layer. The tube center spacing
is 65mm. The diameter of the heat exchange tube is 22mm. The thickness of tube wall is 1.4mm. The width and thickness of corrugated support structure belt are 40mm and 1mm respectively, and the width and thickness of paired straight plate belt are 30mm and 3mm respectively. The external and inner wall diameter of the primary cylinder is 860mm and 508mm.

In order to analyze the influence of the support structure on the pressure drop and the coupled heat transfer between the shell and tube sides, a section of fluid with or without the original support structure (total length 240 mm, support structure 40 mm high, upper and lower fluid 100 mm high) is selected for refined numerical simulation. Through ICEM mesh generation and Fluent numerical simulation, the mathematical model of local heat exchange section of heat exchanger is established to solve and analyze the three-dimensional flow field and temperature field of heat exchanger. The physical model is shown in Figure 1.

![Physical model of local supporting structure in heat exchange section.](image)

3. Mathematical model

3.1. Basic control equation

Continuity equation [1]

\[
\frac{\partial(\rho u_i)}{\partial x_i} = 0 \tag{1}
\]

Momentum equation

\[
\frac{\partial(\rho u_i u_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left[ \mu_{\text{eff}} \left( \frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i} \right) - \frac{2}{3} \mu_{\text{eff}} \frac{\partial u_k}{\partial x_k} \right] - \frac{\partial p}{\partial x_j} + \rho g_i \tag{2}
\]

Where \( u_i \) is the velocity in the \( x_i \) direction, \( p \) means pressure, \( g_i \) is the acceleration of gravity in \( x_i \) direction. \( \mu_{\text{eff}} = \mu + \mu_t \) is the effective dynamic viscosity, where \( \mu \) is the dynamic viscosity, \( \mu_t \) is the dynamic viscosity of turbulence.

Energy equation

\[
\nabla \cdot \left( \bar{v} (\rho E + p) \right) = \nabla \cdot \left( \kappa_{\text{eff}} \nabla T + \sum_j h_j J_j + (\tau_{\text{eff}} \cdot \bar{v}) \right) \tag{3}
\]
Where $E$ is the total energy, $k_{eff}$ is the effective thermal conductivity, $T$ means temperature, $h_j$ is the enthalpy of split direction $j$, $\tau_{eff}$ is the effective stress tensor.

### 3.2. Turbulence model

This paper uses Realizable $k$-$\varepsilon$ Turbulence model calculation [2, 5], the model has two equations including turbulent kinetic energy equation and turbulent dissipation rate equation, the steady state expression of equation is as follows.

**Turbulent kinetic energy equation**

$$\frac{\partial}{\partial x_j}((\rho u_j)\frac{\partial k}{\partial x_j}) = \frac{\partial}{\partial x_j}[(\mu + \frac{\mu_t}{\sigma_k})\frac{\partial k}{\partial x_j}] + G_k + G_b - \rho \varepsilon$$

**Turbulent dissipation rate equation**

$$\frac{\partial}{\partial x_j}((\rho \varepsilon u_j)\frac{\partial \varepsilon}{\partial x_j}) = \frac{\partial}{\partial x_j}[(\mu + \frac{\mu_t}{\sigma_\varepsilon})\frac{\partial \varepsilon}{\partial x_j}] + C_{1\varepsilon} \varepsilon \frac{\varepsilon}{k} C_{3\varepsilon} G_b - \rho C_{2\varepsilon} \frac{\varepsilon^2}{k + \sqrt{\nu \varepsilon}}$$

Where $k$ is the turbulent kinetic energy, $\varepsilon$ is the dissipation rate of turbulent kinetic energy, $\sigma$ is the turbulent Prandtl number, $G_k$ means the turbulent kinetic energy due to the average velocity gradient, $G_b$ means the turbulent kinetic energy due to buoyancy. $\nu$ means dynamic viscosity, $C_{1\varepsilon}$, $C_{2\varepsilon}$, $C_{3\varepsilon}$ is a constant.

Standard wall function method be used for the region of surface.

### 3.3. Basic assumptions

For the above physical models, the following assumptions are made as:

1. The surface of part of the system is in steady state and without heat exchange with the outside.
2. Unlike forced circulation heat exchangers, Heat exchangers rely on the density difference as the driving head for heat exchange in natural circulation. In this calculation, the inlet flow rate of the shell fluid is given, the shell fluid outlet is set similarly.

### 4. Numerical method

In this paper, fluent 18.0 is used for numerical calculation. The setting of boundary conditions, physical parameters, turbulence model, difference scheme and coupling term of velocity and pressure are shown in table 1-3 [3, 4]. Finally, the three-dimensional temperature field and flow field are obtained.

#### Table 1. Boundary condition setting.

| Parameter name      | Unit | Shell side | Tube side |
|---------------------|------|------------|-----------|
| Rate of flow        | kg/s | 84         | 50        |
| Inlet temperature   | °C   | 489.69     | 425.75    |
| Outlet pressure     | Pa   | 0          | 0         |

#### Table 2. Setting of physical property parameters.

| Parameter name                      | Unit                  | Numerical value                  |
|-------------------------------------|-----------------------|----------------------------------|
| Density                             | kg/m³                 | 1015.25-0.247                    |
| Thermal conductivity                | W/(m·K)               | 107.788-0.0576°F                 |
| Specific heat at constant pressure  | J/(kg·K)              | 1658.2-0.8479°F+0.00045417°F²    |
| Dynamic viscosity                   | Pa·s                  | 0.0006188-5.626×10⁻⁷°F+9.84×10⁻¹°F² |
| Thermal conductivity of wall material | W/(m·K)              | 13.976+0.0144(T-273.15)         |
Table 3. Solver settings.

| Parameter name                        | illustration            |
|---------------------------------------|-------------------------|
| Turbulence model                      | Standard k-epsilon      |
| Pressure                              | Second-order            |
| Momentum                              | Second-order upwind     |
| Turbulent kinetic energy              | First-order upwind      |
| Turbulent dissipation rate             | First-order upwind      |
| Pressure-velocity coupling            | SIMPLE                  |

5. Calculation results

5.1. Coupling results of shell-tube heat transfer

In this paper, the coupling calculation of shell-tube side is carried out to evaluate the influence of support structure on flow rate, heat transfer and pressure drop. Because there are a lot of tangent points between the supporting structure and the pipe wall in the tube bundle area, it is very difficult to generate the grid or even cannot be generated. Considering that the support structure has a great influence on the calculation of the coupling heat transfer of the shell-tube sides, the support structure model is simplified. The calculation results are shown in Table 4 and Figure 2.

Table 4. Comparison of fluid characteristic section results with and without support structure.

| Section  | Mass-average temperature°C | Mass-average velocity(m/s) | Mass-average pressure(Pa) |
|----------|----------------------------|----------------------------|----------------------------|
|          | Without                    | With                       | Without                    | With                       |
| Inlet-S     | 490                        | 490                        | 0.381                      | 0.381                     | 9                          | 23                          |
| Outlet-S    | 478                        | 478                        | 0.383                      | 0.388                     | 0                          | 0                           |
| D-value     | 12                         | 12                         | 0.002                      | 0.007                     | 9                          | 23                          |
| Inlet-T     | 426                        | 426                        | 0.720                      | 0.720                     | 71                         | 71                          |
| Outlet-T    | 446                        | 446                        | 0.735                      | 0.735                     | 0                          | 0                           |
| D-value     | 20                         | 20                         | 0.015                      | 0.015                     | 71                         | 71                          |

(a) Pressure distribution with supporting structure  
(b) Pressure distribution without supporting structure
From the calculation results, the pressure and velocity distribution of the shell side fluid are obviously influenced by the supporting structure. Because of the existence of the supporting structure, the flow area of the shell side fluid suddenly decreases in this area, so the fluid velocity increases and the pressure decreases. The pressure drop on the shell side increases from 9Pa to 23Pa.

Compare Figure 2(e) with (f), there is no obvious change in the temperature distribution of the primary side, which indicates that the support structure has little influence on the heat transfer of the primary side fluid, which can also be proved by the outlet temperature of the shell side in table 4. For the tube side, the pressure, temperature and velocity distributions are almost not affected by the constrained structure.

5.2. Resistance characteristics of shell side restraint structure
Through the above analysis, it can be seen that the support structure mainly affects the flow characteristics of the shell side, but has little effect on the flow and heat transfer characteristics of the tube side. However, in the above analysis, the support structure region is simplified to some extent, and the resistance characteristics of the shell side fluid in the support structure region can not be accurately obtained. In order to preferably analyse the influence of the support structure on the pressure drop of the shell side, a section of the shell side fluid region containing only the original support structure is to be calculated. The shell side entrance boundary conditions and exit boundary conditions are consistent with
the above boundary conditions, ignoring the heat transfer. The calculation results are shown in Table 5 and Figure 3.

Table 5. Three Scheme comparing.

| Section  | Mass-average velocity(m/s) | Mass-average pressure(Pa) |
|----------|---------------------------|---------------------------|
|          | With                      | Without                   | With  | Without |
| Inlet-S  | 0.380                     | 0.381                     | 27    | 9       |
| Outlet-S | 0.389                     | 0.383                     | 0     | 0       |
| D-value  | 0.009                     | 0.002                     | 27    | 9       |

The results show that the existence of support structure has obvious influence on the axial pressure and velocity distribution of the shell side fluid, the existence of support structure increases the primary pressure drop by 18 Pa.
6. Conclusion

The local disturbance of the shell fluid does not affect the heat transfer of the heat exchanger, and there is no local heat transfer deterioration. The existence of the confinement structure has no obvious effect on the temperature field of the fluid in the tube side. The combination of corrugated strip and straight strip is feasible for the design of heat exchanger.

The resistance of a single group of corrugated strips reaches 23pa at the low flow rate, and the resistance value of five groups of corrugated strips of heat exchanger accounts for one third of the total shape resistance. Therefore, it has a great influence on the establishment of natural circulation. Under the condition of ensuring the constraint of the heat exchange tube bundle in the operation state, the resistance value of the single group of corrugated strips accounts for one third of the total shape resistance.

On the premise of ensuring the performance of the heat exchanger, the restraint structure should be reduced as much as possible to reduce the shell side resistance.

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