Investigation of the shell-side flow in a shell-and-tube exchanger with impingement plate

Yi Wang¹,², Guoliang Qin¹*, Yong Zhang², Shuhua Yang², Yuying Sun², Qin Cui¹, Cheng Jia ¹

(1. School of Energy and Power Engineering, Xi’an Jiaotong University, Xi’an 710049, China; 2. Shenyang Blower Works Group Corporation, Shenyang 110869, China)

E-mail: glqin@mail.xjtu.edu.cn

Abstract. For studying the influence of impingement plates on the fluid flow within the shell-side of the shell-and-tube heat exchanger, two-dimensional numerical simulation is carried out using the Fluent software. The different distances between the impingement plate and the tube bundle are taken into consideration. Meanwhile the research investigates the flow field of the non-perforated impingement plate and the perforated impingement plates with various hole diameters. The results show that the fluid behind the non-perforated impingement plate contacts the tube insufficiently. The farther the distance between the non-perforated impingement plate and the tube bundle is, the more tubes contact with the fluid poorly. The flow field behind the perforated impingement plates is more uniform. The protective effect from flow impacting for impingement plates is positive. The perforated impingement plates are recommended in the engineering design of the shell-and-tube heat exchanger.

1. Introduction

Shell-and-tube heat exchangers are widely used in the industrial fields such as the energy and power industry, the chemical industry, the metallurgy, the aerospace and so on. Shell-and-tube heat exchangers have many excellent characteristics, including the simple structure, low cost, easy scale cleaning, resistance quality of high temperature and high pressure. Heat exchanger designers usually put an impingement plate at the entrance of the shell side or the tube side of the heat exchanger according to the provisions of GB-T151-2014 [1]. The impingement plate is a protective auxiliary part. Firstly, it can prevent the tube bundle and the tube sheet of the heat exchanger from directly impacting by the working fluids. Meanwhile, it can prevent the tube bundle vibration caused by the high flow rate working fluid. Finally, it can also prevent the tube bundle and the tube sheet from erosion and corrosion sometimes.

At present, the research on shell-and-tube heat exchangers mainly focuses on heat transfer elements and pressure-bearing elements. There are relatively few studies on impingement plates. Fu Shuangquan [2] developed an Excel program to calculate the using criterion of the impingement plate for heat exchange designers. Liu Xiaoling [3] used empirical formulas to analyze the impingement plate on the shell-side of the heat exchanger from the perspective of the tube bundle vibration. Defan Qing [4] took the experiment on heat transfer and flow resistance of elliptic rib and
circular rib. Xiaojun Chen [5] indicated the application of fin tube cooler with bimetallic strip. Bin Xue et al [6] investigated the relationship between resistance and heat transfer of inner finned tube with single or double holes based on measuring and calculating. Feng Su et al [7] used the theory of multi-span beams to study the transverse vibration induced by a tube-side flow. Mingqing Liu [8] summarized the type and character of the heat exchanger used as intercooler of the compressor process.

In recent years, CFD research on impingement plates began to be reported. Wu Yangkuan et al. [9] took the horizontal condenser as the research object, using the Ansys CFX software to study the effect of the slotted impingement plate on the shell side. Liu Xiangbin et al. [10] took the channel of a single-tube-pass heat exchanger as the research object. They simulated the protection effect of impingement plates with the circular flat shape and the angle shape 120° cone by the Ansys CFX software. They found the circular flat impingement plate with holes had the best protection on alleviating the impact of the working fluid within the channel. Salamah S. Al-Anizi et al. [11] developed double perforated impingement plate to stop fouling accumulation on the tubes to prevent under-deposit corrosion.

The flat impingement plate is widely used in shell-and-tube heat exchangers for it is easy to manufacture. The impingement plate at the shell side could be welded on the inner wall of the shell or the spacers of the tube bundle. The installation type is shown in Figure 1. In addition, the flat impingement plate would be made into the non-perforated one or the perforated one, as shown in Figure 2. At present, both the installation type and the perforated structure parameters of the impingement plate depend on the designer’s experience. There are no standards or guidelines about the impingement plate.

In a shell-and-tube heat exchanger heat transfers from one fluid in the tube to another fluid in the shell through the tube wall. The effective heat transfer means that the fluid in the shell-side flows fluently through the gaps or the tube bundle. However, few researches have been done about the flow behind the impingement plate in the shell, especially by numerical simulation. Therefore the research is taken into consideration in this paper on how the shell-side flow field of the tube bundle area is effected by the impingement plate.

In this paper, the research object focuses on the fluid flow through the tube bundle area behind the impingement plate at the shell side of the shell-and-tube heat exchanger. Two-dimensional numerical simulations are performed with the Fluent software to investigate the flow field on the different installation type and the various perforated structure of the impingement plate. The conclusions aim to provide references for the designers of shell and tube heat exchangers.
2. Description of the engineering case
This research is based on a shell-and-tube heat exchanger designed by Shenyang Blower Works Group Corporation. All the simulation data comes from the engineering application. Considering the commercial confidentiality, only basic public data of the shell-and-tube heat exchanger is shown in the paper.

2.1. Calculation of the Reynolds Number
According to the formula in the reference [12], when the tubes layout are designed in the form of equilateral triangle, the equivalent diameter of shell-side flow area is calculated by:

\[
d_e = \frac{4 \left( \frac{\sqrt{3}}{2} l^2 - \frac{\pi}{4} d_o^2 \right)}{\pi d_o} = 17 \text{ (mm)}
\]  

where the center distance between two adjacent tubes is 25mm, and the tube outer diameter is 19mm.

In the heat exchanger, the baffle spacing is 500mm. The shell diameter is 1600mm. The maximum flow area in the shell-side is calculated by:
\[ A_{s,\text{max}} = BD_s \left( 1 - \frac{d_s}{l} \right) = 0.192 \ (m^2) \] (2)

The working fluid in the shell-side is water. The mass flow of water is 243289kg/hr. The water velocity in the shell-side is calculated by:

\[ u_i = \frac{Q}{\rho_i A_{s,\text{max}}} = 0.35 \ (m/s) \] (3)

The Reynolds number of the shell-side is calculated by:

\[ Re = \frac{d u_i \rho_i}{\mu_i} = 5932 \] (4)

According to the reference [12], the flow pattern at the shell side is determined by the Reynolds number. When the Reynolds number at the shell-side fluid is greater than 100, the flow is turbulent. Therefore, water flows turbulently at the shell-side in the sample.

2.2. Criterion of the Impingement Plate

According to GB/T151-2014[1], the minimum flow area of the shell-side inlet nozzle is calculated by:

\[ A_{s,\text{min}} = \pi D_{in} H_{it} + F_1 \left( \frac{\pi D_{in}^2}{4} \right) \left( \frac{l - d_s}{F_2 l} \right) = 42461.95 \ (mm^2) \] (5)

where \( D_{in} \) is the inner diameter of the shell-side inlet nozzle. In the case, the inner diameter of the shell-side inlet nozzle is 255mm. \( H_{it} \) is the averaged free height measured between the tube layout and the inlet nozzle, \( H_{it} = 50 \)mm. The coefficients \( F_1 \) and \( F_2 \) are dimensionless coefficients, which are taken as recommended in GB/T151-2014[1], \( F_1 = 1 \) and \( F_2 = 1 \).

It has been stated in GB/T151-2014[1] that an impingement plate is needed when the value of \( \rho \nu^2 \) is greater than 2230kg/(m·s²). In the sampled heat exchanger, the value of \( \rho \nu^2 \) is calculated by:

\[ \rho \nu^2 = \rho \left( \frac{Q}{3600 A_{\text{min}}} \right) = 2523.55 \ (kg/(m \cdot s^2)) \] (6)

Then, an impingement plate is needed at the shell-side of the heat exchanger.

2.3. Averaged Velocity at the Shell-side Inlet Nozzle

According to the fluid flow data in the shell-side, the average velocity at the shell-side inlet nozzle is calculated by:

\[ u_{in} = \frac{Q}{3600 \rho_i \left( \frac{\pi D_{in}^2}{4} \right)} = 1.33 \ (m/s) \] (7)

Various velocities are studied in the simulation for investigating its influence on the flow field. All values of the velocity in the study is based on the values calculated above.

3. Numerical simulation

3.1. Geometric Model

Flow change is neglected along the longitudinal direction of the shell in the research. The tube layout is simplified for simulation. Figure 3 shows the geometric model of the simplified numerical calculation area. The tube bundle is quite far from the inlet and the outlet to reduce the influence of the inlet and outlet boundary on the flow field. The tubes are arranged in the form of a regular triangle as stated in GB/T151-2014[1]. The outer diameter of the tube is 19mm. The spacing of the tube is 25mm.
In order to investigate the effect of the impingement plate on the flow field in the tube bundle area, there are two kinds of geometric models— with or without the impingement plate. Many variables are considered in the model with the impingement plate. The different distances between the impingement plate and the tube bundle are investigated first. Then, the research displays the flow field behind the impingement plates perforated or not. In addition, flow field behind the perforated impingement plates with various diameters is studied.

Assuming that the holes on the perforated impingement plate are round, the distance between the two holes center is 2 times the hole diameter. The cross-section is selected as the calculation area where the center of the hole is located. The variables for simulation are shown in Table 1.

It is recommended that the tubes should be as many as possible in the heat exchanger when designing. Therefore, the vertical distance is basically not more than 120mm between the center line of the tubes in the first row and the intersecting line of the shell and the inlet nozzle. Meanwhile, the distance between the impingement plate welded on the shell and the intersecting line of the tubes in the first row is about 60~70mm based on the design experience. Based on the above statement, the maximum distance between the first row of the tubes and the impingement plate is 51mm in the research, which can cover the actual distances of most engineering cases.

![Figure 3. Geometric Model (mm).](image)

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**Table 1** The variables for simulation (mm).

| name | value | denotation |
|------|-------|------------|
| h    | 1     | The impingement plate is welded on the spacers. |
|      | 21    | The impingement plate is welded on the shell. |
|      | 51    | The distance between the impingement plate and the tube bundle is 21mm. |
|      | 0     | The impingement plate is welded on the shell. |
|      | 10    | The distance between the impingement plate and the tube bundle is 51mm. |
|      | 20    | The non-perforated impingement plate |
|      |       | The perforated impingement plate with the hole diameter 10mm |
|      |       | The perforated impingement plate with the hole diameter 20mm |
3.2. Mathematical model
In this paper, the numerical simulations of the above geometric models are carried out with the Fluent software. The working fluid is assumed as water. The flow is assumed to be a two-dimensional steady-state incompressible flow. The mathematical models used in the research are the continuity equation, the momentum conservation equation, and the standard k-ε turbulence model. The second-order upwind difference format is used to discretize the above-mentioned equations. The COUPLED algorithm is used to deal with the coupling of pressure and velocity. The convergence condition is the equation residual less than 10^{-5}.

The boundary conditions are imposed as follows. The inlet boundary is the velocity inlet. The outlet boundary is the free flow. The fixed and non-slip wall surface is used on the surface of the tubes and the impingement plate, as well as the two side walls.

3.3. Meshing
Unstructured grids are used in the entire calculation area. Mesh encryption is carried out in the tube bundle and the impingement plate area for the calculation accuracy. The sparse mesh is used in the area of inlet and outlet where little influence of these areas is imposed on the result.

In this study, the grid independence is verified by different geometric models. Taking the geometric model in Figure 3(a) as an example, the relationship is studied between the nodes and the averaged outlet velocity on the y-direction, as shown in Figure 4. In order to save the calculation cost, this case adopts a meshing scheme with 53674 nodes for calculation.

![Grid independence verification](image)

Figure 4. Grid independence verification.

4. Results and discussions

4.1. The Flow With or Without the Impingement Plate
Two cases are modeled in order to investigate the influence of the impingement on the flow field of the tube bundle area. Case 1 is without the impingement plate. Case 2 is with the non-perforated impingement plate welded on the spacers. The water inlet velocity is 0.09m/s. The flow velocity contour is shown in Figure 5.

When the water flows through the tube bundle, the flow field shows a symmetrical distribution along the vertical centerline of the model’s area in both cases. Vortexes appear between the tubes in the last row and the outlet. Actually, the gap between the tube bundle and the shell wall is very small in the design of the heat exchanger, so that there is no space for large vortex generation.

The main purpose of this research is to study the fluid flow in the tube bundle area, not too much blank area. Therefore, in the subsequent discussions, the appropriate space around the tube bundle is
intercepted for flow field analysis. Meanwhile, half of the area along the vertical centerline is cut for the symmetry. Then the streamline and velocity contour of these two cases are shown in Figure 6.

The fluid can smoothly pass through the gap between the tubes in case 1 as shown in Figure 6 (a). In case 2 as shown in Figure 6 (b), an obvious vortex is generated around the right tube of the first row after the impingement plate. Affected by the vortex, the fluid bypassing the impingement plate cannot smoothly flow into the windward side of the tubes in the first row. The fluid flows from the leeward side of the tubes in the first row into the gaps of the tubes in the second row and the third row. The non-perforated impingement plate welded on the spacer will prevent the first row of tubes from convection with the fluid.

- **Figure 5.** Velocity contour at inlet velocity 0.09 m/s.

- **Figure 6.** Streamline and velocity contour at inlet velocity 0.09 m/s.

### 4.2. The Distance between the Impingement Plate and the Tube Bundle

Three cases are studied with various distances between the impingement plate and the tube bundle in this section. The distance between the impingement plate and the tube bundle is 1 mm in case 2. The distance between the impingement plate and the tube bundle is 21 mm in case 3. The distance between the impingement plate and the tube bundle is 51 mm in case 4. The impingement plate is non-perforated. The water inlet velocity is 0.09 m/s for three cases. The streamline and velocity contours of these three cases are shown in Figure 7.
The flow field of case 2 shown in Figure 7 (a) is discussed in the section 3.1 when analyzing the flow behind the non-perforated impingement plate welded on the spacer shown in Figure 6 (b).

When the distance between the impingement plate and the tube bundle is 21mm as shown in Figure 7 (b), a large vortex is formed between the impingement plate and the first tube on the right side of the first row. A number of small vortices are formed between the impingement plate and the other tubes of the first row. These vortices make most of the streamlines pass through the leeward side of the tubes in the first row, flowing into the area of the second row and the third row. Similar to the flow of case 2, the fluid does not fully contact with the first row bypassing the impingement plate.

When the distance between the impingement plate and the tube bundle is 51mm as shown in Figure 7 (c), two large vortices are formed between the first row of tubes and the impingement plate. These two vortices cause many small vortices in the gaps between the tubes in the first row. Affected by these vortices, the fluid basically does not contact the tubes in the first row. Meanwhile, the fluid does not contact some tubes in the second row near the vertical centerline for these vortices.

![Streamline and velocity contour](image)

(a) case 2 (h=1mm). (b) case 3 (h=21mm). (c) case 4 (h=51mm).

Figure 7 Streamline and velocity contour at inlet velocity 0.09m/s.

4.3. The Inlet Velocity of the Shell-side Fluid

When the inlet velocity of the water is 0.35m/s, the streamline and velocity contours behind the non-perforated are shown in Figure 8. Three cases are studied with various distance between the impingement plate and the tube bundle. The distance between the impingement plate and the tube bundle is 1mm in case 5. The distance between the impingement plate and the tube bundle is 21mm in case 6. The distance between the impingement plate and the tube bundle is 51mm in case 7. The impingement plate is non-perforated.

The flow field after the non-perforated impingement plate welded on the spacer still shows that the first row tubes are in insufficient contact with the fluid, as shown in Figure 8 (a). For case 6 and case 7, the vortices become larger than that in case 5, as shown in the Figure 8 (b) and (c). Not only the tubes in the first row, but also some tubes in the second and third row are in the vortex zone. That means more tubes cannot effectively contact the fluid for the cases that the non-perforated plate is welded on the shell when the inlet velocity of the water is 0.35m/s.

When the inlet velocity of the water is 1.33m/s, the vortices fill the flow area regardless of the distance between the impingement plate and the tube bundle, as shown in Figure 9. All tubes in Figure 9 contact the fluid insufficiently.
Although the non-perforated impingement plate can prevent the tube bundle from fluid impacting, it will also somewhat affect the flow behind it causing the fluid cannot fully contact the tubes. By comparing the flow field of the various inlet velocities in Figure 7, Figure 8 and Figure 9, it can be seen that the flow field behind the non-perforated impingement plate welded on the spacer is relatively fluent in a wide inlet velocity range. The fluid bypassing the non-perforated impingement plate welded on the spacer could contact more tubes.
Figure 10. Streamline and velocity contour at inlet velocity 0.09m/s.

4.4 The Hole Diameter of the Impingement Plate
A common situation in engineering design of the impingement is to punch holes when the impingement plate is welded on the spacers of the tube bundle. Based on the engineering foundation, this section investigates the flow field behind the perforated impingement plate when $h=1\text{mm}$. The diameter of the hole is specified as 10mm in case 11 and 20 mm in case 12. The streamline and velocity contours when the inlet velocity is 0.09m/s are shown in Figure 10.

It can be seen in Figure 10 that the fluid velocity on the windward side of the perforated impingement plate is reduced. It indicates that even if the impingement plate is punched, it can still play a role in protecting the tube bundle from the fluid impacting. It can be seen that the streamlines are uniform and the low-velocity area at the tube leeward is very small in Figure 10 (a). However, the streamlines are in the uneven distribution somewhere between the gaps of the tubes in case 12. The low-velocity area in the tube bundle extends from the leeward of the tubes to the gaps of the tubes in Figure 10 (b). Based on the discussion between these two cases, the hole diameter is recommended to be 10mm when the perforated impingement plate is designed.

5. Conclusion
In this paper, the fluid flow behind the impingement plate in the shell-side of the shell-and-tube heat exchanger is studied. The main conclusions obtained are as follows:

(1) The flow field behind the non-perforated impingement plate on the shell-side is inevitably uneven. Vortices exist in the flow field causing the fluid contacts the tubes insufficiently. By changing the distance between the non-perforated impingement plate and the tube bundle, it is found that the smaller the distance is, the smaller the vortices are and the more tubes the fluid contacts. According to the above analysis, for an impingement plate is necessary for a heat exchanger according to GB/T151-2014[1], it is recommended to weld on the tube bundle spacers.

(2) Not only can the perforated impingement plate welded on the tube bundle spacers prevent the tube bundle from the fluid impacting, but also it improves the effect of contact between the tubes and the fluid. Meanwhile, the flow field of the perforated impingement plate is more uniform than that behind the non-perforated impingement plate.

(3) Heat exchangers with perforated impingement plates are not suitable for the engineering limitations sometimes. The designer should take into consideration the case of the fluid contact the tubes behind the non-perforated impingement plate insufficiently and the possibility of the insufficient heat transfer capacity. Increasing the heat transfer margin of the heat exchanger is a recommended way to avoid the failure of the heat exchanger caused by insufficient heat transfer capacity.

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