Influence of Coiling Direction of Helical Tube Bundles on the Thermal-Hydraulics of the HTGR Steam Generator

Weikai Gao¹, Xiaoyang Xie¹, Xiaowei Li¹*, Xinxin Wu¹

¹Key Laboratory of Advanced Reactor Engineering and Safety of Ministry of Education, Collaborative Innovation Center of Advanced Nuclear Energy Technology, Institute of Nuclear and New Energy Technology, Tsinghua University, Beijing 100084, PR China
²China Academy of Launch Vehicle Technology, Beijing 100076, PR China

*lxiaowei@tsinghua.edu.cn

Abstract. Helical tube bundles were usually adopted in the steam generators (SGs) or intermediate heat exchangers (IHXs) of high temperature gas-cooled reactors (HTGRs). Heat transfer tubes in neighboring tube layers can be coiled in the same direction or in the opposite direction. The coiling direction has influences on the thermal-hydraulic performances of the SGs or IHXs. The cross flow convection over helical tube bundles with neighboring tube layers having the same coiled direction and opposite coiled direction were numerically investigated. Reynolds stress model with standard wall functions was used for the turbulence modeling. For a helical tube bundle with neighboring layers coiled in the same direction (parallel tube layers), the tangential velocity along the coiled circumferential direction could be observed obviously. For a helical tube bundle with neighboring layers coiled in the opposite direction (crossed tube layers), there is no average tangential velocity of the whole flow field. And the streamlines of the fluid are very complex. The flow resistances and heat transfer coefficients over helical tube bundles with crossed tube layers and crossed tube layers were compared. Although the heat transfer over helical tube bundles with crossed tube layers was 9.39% smaller than that with parallel tube layers, the pressure drop over tube bundle with crossed tube layers was much smaller compared with those with parallel tube layers.

1. Introduction
Steam generator (SG) is the most important heat transfer equipment for High Temperature Gas-cooled Reactor (HTGR) [1, 2]. Helical tube bundles are usually adopted in the SG to increase the heat transfer area and improve the compactness due to the low heat transfer coefficient in the gas side. As the SG adopts once through flow pattern, its heat capacity is very small. The once through steam generator (OTSG) is sensitive to thermal hydraulic parameter fluctuation and geometric deviations [3]. Analysis of the HTR-PM OTSG shows that the flow and heat transfer characteristics in the primary side have more influences on its overall thermal hydraulic performance [4]. The basic flow pattern in the primary side of the SG is turbulent cross flow over helical tube bundles. The geometrical parameters such as longitudinal tube pitches, transverse tube pitches and tube diameter, will influence the cross flow and heat transfer characteristics over tube bundles. 1D and 2D tube and shell side coupled methods were proposed by Li et al. for HTGR OTSG [5]. The overall thermal performance of HTR-PM OTSG was simulated using a 1D method. The thermal non-uniformity or thermal hydraulic deviations caused by the geometrical and thermal hydraulic deviations were analyzed using a 2D method. It is shown that a 1% change of secondary side water flow rate of a certain layer of tubes will result in a 5-6°C variation of the...
outlet steam temperature of this layer of tubes. A helical diameter variation of 1 mm will result in a 6-7°C variation of the outlet steam temperature. An inlet helium temperature profile with a 10°C variance will result in a 2°C variation of the outlet steam temperature.

The influences of the bounding walls on the cross flow and heat transfer characteristics were numerically investigated using CFD method [6]. It is found that the wall effects depress turbulence intensities of the flow in the near wall flow passages. So the heat transfer coefficients of the near wall tubes are 10% lower than those of the tubes in the middle of the bundle. The influence of fabrication tolerance on the cross flow and heat transfer characteristics over tube bundles in the HTR-PM OTSG was numerically investigated using the 2D tube and shell side coupled method and the 2D URANS method [4]. It is found that a fabrication tolerance of 0.5 mm in helical diameter will result in 1.4% pressure resistance variation and 0.7% flow rate variation among the 19 units. And it will induce about 5°C steam temperature variance among the 5 layers of tubes. The influence of the of helical tube bundles on the thermal hydraulic performance of the helical tube bundles was numerically investigated. The turbulent cross flow and heat transfer characteristics over medium scale and large scale helical tube bundles were compared [7]. It is found that the temperature non-uniformity in the large scale helical tube bundles is larger than that in the medium scale helical tube bundles.

There are two types of coiling directions of helical tube bundles in the commonly used steam generators. Heat transfer tubes on neighboring tube layers can be coiled in the same direction or in the opposite direction. The coiling direction has influences on the thermal-hydraulic performances of the SGs or IHXs. The angles between the tube axes of helical tubes and the plane perpendicular to the fluid flow directions, which are called the flow attack angle or the yaw angle of the tubes, also affects the cross flow and heat transfer characteristics over helical tube bundles. For example, the IHX of High-temperature Engineering Test Reactor (HTTR) adopted helical tube bundles with neighboring tube layers coiled in the same direction [8]. The overall flow resistance, heat transfer characteristics and flow induced vibrations was experimentally studied. The helix angle of the HTTR IHX is 12°. It is found that the flow resistance and heat transfer coefficient were both smaller than those calculated using the experimental correlations. HTR-PM SG adopted helical tube bundles with neighboring layers coiled in the opposite direction. The helix angle of HTR-PM SG is about 7°.

It was found that the flow resistance and Nusselt number measured in the experiment were smaller than that calculated using the experimental correlations [9]. Groehn [10] experimentally studied the influence of the yaw angle on heat transfer and pressure drop of straight tube bundles in the heat exchangers. The yaw angle of the tubes varied between 15° and 90° (15°≤φ≤90°). And Reynolds number varied between 2×10^4 and 10^6 (2×10^4≤Re≤10^6). It was found that the correlations for flow attack angle of 90° could be applied to predict the heat transfer coefficient of yawed tube banks if the velocity is corrected for calculating the Reynolds number. However, it is not applicable to predict the hydraulic resistance.

Groehn [11] experimentally studied the flow and heat transfer characteristics over yawed tube banks with neighboring layers having opposite sense of inclination. Moreover, a full-scale test of a helical-type heat exchanger has been done to verify the fundamental research. It was found that opposite-sense inclination of neighboring tube layers of inline tube banks leads to a small reduction of the heat transfer coefficient while the hydraulic resistance decreases substantially. The opposite sense inclination becomes more evident with smaller transverse tube pitches. It was also shown that the results from the investigations of inclined straight tube banks are transferable to helical coiled tube bundles.

Due to the high compactness and sensitivity of HTR-PM SG, the turbulent cross flow and heat transfer characteristics over helical tube bundles with different coiling direction should be studied in more detail. Because of the complexity of the geometry of the helical tube bundle and the huge computational resources required for the simulation, only the overall flow characteristics over helical tube bundles was simulated. The detailed flow and heat transfer was simulated using the geometry of straight tube bundle with inclination. It requires less computational resources and more accurate models can be used. Then the influences of coiling direction on the cross flow and heat transfer characteristics over tube bundles was studied.
2. Numerical methods

2.1. Geometric model

The geometric models of helical tube bundles were shown in Figure 1 and Figure 2. They were three-dimensional wall bounded helical tube bundles with 20 rows of tubes in the axial direction (stream wise direction) and 5 layers of tubes in the radial direction (transverse direction). The geometric model of helical tube bundles coiled in the same direction was shown in Figure 1. The geometric model of the 1st and 2nd tube layers of helical tube bundles coiled in the opposite direction was shown in Figure 2(a). The geometric model of the 3rd, 4th and 5th tube layers was shown in Figure 2(b). Layer No.1 and layer No.5 tubes were adjacent to the inner wall and outer wall. There were 5, 6, 7, 8, 9 rows of tubes in layer No. 1, 2, 3, 4, 5, respectively. The structure of the helical tube bundles was very complex. In order to make it clearly understood, only part of the helical tubes in each layer of the tube bundles was shown in the figures. The tube diameter was 19 mm. The stream wise tube pitch was 25 mm, and the transverse tube pitch was 30 mm. The helix angle was 7.3°.

![Figure 1. Geometric model of helical tube bundles coiled in the same direction](image1)

![Figure 2. Geometric model of helical tube bundles coiled in the opposite direction](image2)
The geometric models of the straight tube bundles were shown in Figure 3. The tube bundles were in similar arrangements as the helical tube bundles. There were 5 layers of tubes in the transverse direction and 8 rows of tubes in the stream wise direction. The tube diameter was 19 mm. The stream wise tube pitch was 25 mm, and the transverse tube pitch was 30 mm. The yawed angle was 7.3°.

![Figure 3. Geometric model of straight tube bundles with inclination](image)

(a). Tube bundle with parallel tube layers  
(b). Tube bundle with crossed tube layers

2.2. Governing equations and boundary conditions

Unstructured grid was generated for the computational domain of helical tube bundles. The total element number of the tube bundle with neighboring layers coiled in the opposite direction was 18,079,231. The total element number of the tube bundle with neighboring layers coiled in the same direction was 18,146,556. Structured grid was generated for the computational domain of straight tube bundles with inclination. The total element number of tube bundle with parallel tube layers was 18,709,600. The total element number of tube bundle with crossed tube layers was 51,443,136.

Velocity inlet boundary condition was set for the inlet. The outlet was fully developed boundary condition. Adiabatic and non-slip wall boundary conditions were used for the inner wall and outer wall. Constant temperature and non-slip wall boundary conditions were set for the tube walls. The tube wall temperature was 953.15 K. The working fluid was helium with constant properties. The physical properties of helium are shown in Table 1. The inlet velocity of the helical tube bundles is 7.134 m/s. The operating pressure is 7 MPa.

| Properties                  | Units     | Value    |
|-----------------------------|-----------|----------|
| Density [\(\rho\)]         | kg/m³     | 3.46     |
| Heat capacity [\(c_p\)]    | J/(kg·K)  | 5189.67  |
| Thermal conductivity [\(\lambda\)] | W/(m·K)   | 0.36     |
| Viscosity [\(\mu\)]        | kg/(m·s)  | 4.51E-05 |

The governing equations were the time-dependent three-dimensional Reynolds Averaged Navier-Stokes equations and the energy equation. Li et al.[6] numerically investigated the turbulent cross flow and heat transfer in a wall bounded tube bundle. Standard \(k-\varepsilon\) model, SST \(k-\omega\) model and Reynolds stress model were used for the simulation. It was found that both standard \(k-\varepsilon\) model and Reynolds stress model showed good agreements with the experiments. Thus, Reynolds stress model was used for the turbulence modeling in this paper. Standard wall function was used for the near wall treatment of helical tube.
bundles. And enhanced wall treatment was used for the near wall treatment of straight tube bundles. The governing equations were solved using the commercial CFD software ANSYS Fluent. Velocity-pressure coupling was treated with the SIMPLEC algorithm. The convection terms were discretized with the QUICK scheme. The time step is 0.0001 s

3. Results and discussion

3.1. Overall flow characteristics over helical tube bundles

Figure 4 shows the transient velocity contour of the helical tube bundle on z=0 plane. Figure 4(a) is the velocity contour of the tube bundle with neighboring layers coiled in the same direction. Figure 4(b) is the velocity contour of the tube bundle with neighboring layers coiled in the opposite direction. It is shown that the fluid in the tube bundle sways as it passes the tube bundles. The turbulent wakes after the tubes swing towards upper right or lower right or backward. The bounding walls (inner cylinder wall and outer cylinder wall) suppress the swing of the wake. So the turbulent wakes after the tubes adjacent to the bounding walls could hardly sway towards them. The transient periodic zigzag flow appeared in wall bounded inline tube bundles was also observed in the helical tube bundles with neighboring layers coiled in both opposite direction and the same direction.

It is shown that the wake swing in the helical tube bundle with neighboring layers coiled in the opposite direction has no obvious periodicity compared to that in the tube bundles with neighboring tube layers coiled in the same direction and that in the straight tube bundle. It is mainly because that the velocity components along the tube axis influences the main flow and the vortex shedding. For the helical tube bundle with neighboring tube layers coiled in the same direction, the tube arrangements in different cross section along the circumferential direction are the same. The maximum velocity in these cross sections are equal. The velocity components along the tube axis have the same influence on the vortex shedding. For the helical tube bundle with neighboring tube layers coiled in the opposite direction, the tube arrangements in different cross section along the circumferential direction are different. The flow passages in these sections are slightly different. The maximum velocities in the different cross sections are not equal. The velocity components along the tube axis have different influences on different tubes. Thus, the wake swing in tube bundle with crossed tube layers are not periodic.

![Figure 4. Transient velocity contour](image-url)
The circumferential velocity contours of the helical tube bundles are shown in Figure 5. The cross section shown in Figure 5 is illustrated in Figure 8 and Figure 9. Figure 5(a) shows the circumferential velocity in helical tube bundles with neighboring layers coiled in the same direction. It is shown that the circumferential velocities of all tube layers in the upper section are negative. And the circumferential velocities of all tube layers in the lower section are positive. Thus, due to the same coiling direction of the helical tubes, the direction of the circumferential velocities in different tube layers are all the same. Figure 5(b) shows the circumferential velocity in helical tube bundles with neighboring layers coiled in the opposite direction. It is shown that the circumferential velocities near the 1st, 3rd and 5th tube layers in the upper section are negative. However, the circumferential velocities near the 2nd and 4th tube layers are positive. Similarly, the circumferential velocities near the 1st, 3rd and 5th tube layers in the lower section in Figure 5(b) are positive. The circumferential velocities near the 2nd and 4th tube layers are negative. Thus, due to the opposite coiling direction of the helical tubes, the direction of the circumferential velocities in different tube layers are different.

![Circumferential Velocity Contour](image)

(a). Helical tube bundle coiled in the same direction

(b). Helical tube bundle coiled in the opposite direction

**Figure 5.** Circumferential velocity contour

The transient velocity vector in tube bundle with neighboring layers coiled in the same direction is shown in Figure 6. The overall velocity field of section $x=0.275$ m is shown in Figure 6(a). The detailed velocity field of the local area marked by the red box in Figure 6(a) is shown in Figure 6(b). It is shown that almost all the velocity vector on the cross-section are in the same direction. For a helical tube bundle with neighboring layers coiled in the same direction, the tangential velocity along the coiled circumferential direction could be obviously observed.
The transient velocity vector in tube bundle with neighboring layers coiled in the opposite direction is shown in Figure 7. The overall velocity field of section $x=0.275$ m is shown in Figure 7(a). The detailed velocity field of the local area marked by the red box in Figure 7(a) is shown in Figure 7(b). It is shown that the velocity vector in the neighboring layers are in the opposite direction. For example, the flow direction of the fluid in different radial position of the 2nd flow passage (the flow passage between the 1st tube layer and the 2nd tube layer) is different. The fluid near the 1st tube layer flow towards upper left and the fluid near the 2nd tube layer flow towards lower right. The fluid flow is more complex than that in the cross section of tube bundle with neighboring layers coiled in the same direction. And there is no average tangential velocity of the whole flow filed.
The circumferential velocity and streamlines of the cross section of helical tube bundle with neighboring layers coiled in the same direction is shown in Figure 8. The helical tubes were coiled in the same direction and the attack angles of the tubes in the neighboring tube layers were all 7.3°. Therefore, the directions of the circumferential velocity are all the same. As a result, the streamlines in the tube bundle bends along the coiling direction of the helical tubes. And the tangential velocity along the coiled circumferential direction could be obviously observed. The main fluid flows in a helical direction in tube bundles with neighboring layers coiled in the same direction.

Figure 8. Circumferential velocity contour and streamlines in tube bundle with neighboring layers coiled in the same direction

The circumferential velocity and streamlines of the cross section of helical tube bundle with neighboring layers coiled in the opposite direction is shown in Figure 9. The absolute value of the attack angle of the neighboring tube layers is equal. But one is +7.3° and the other is -7.3°. Therefore, the direction of the circumferential velocity in different tube layers is opposite. As a result, the streamline shown in Figure 9 are almost in the same cross section, and a certain degree of circumferential bending occurs. This will make the thermal mixing in the tube bundle with neighboring layers coiled in the opposite direction better than that in the tube bundle with neighboring layers coiled in the same direction.

Figure 9. Circumferential velocity contour and streamlines in tube bundle with neighboring layers coiled in the opposite direction
3.2. Time-averaged flow characteristics in straight tube bundles

The turbulent cross flow over tube bundles is periodic. The transient flow and heat transfer characteristics are extracted from the data files of each time step. Then the time averaged flow and heat transfer characteristics are obtained by averaging the transient flow and heat transfer characteristics for several periods.

Figure 10 shows the drag coefficient of the tube bundles with parallel tube layers and crossed tube layers. The horizontal coordinates are the row numbers of the tubes. The drag coefficient of the tube is calculated using Eq. (1).

\[ C_d = \frac{F_{p,x} + F_{v,x}}{0.5 \rho u_\infty^2 d H} \]

where \( F_{p,x} \) is the x direction pressure force, \( F_{v,x} \) is the x direction viscous force. The forces are calculated by integrating along the wall surfaces. \( d \) is the diameter of the tube; \( H \) is the length of the tube.

It is found that the drag coefficient of the tube bundle with crossed tube layers is smaller than that of the tube bundle with parallel tube layers in the fully developed region. The drag coefficients of the tubes near the inlet and outlet, which are influenced by the inlet effect and outlet effect, are neglected. The averaged drag coefficient of the tube bundle was calculated by averaging the drag coefficients of the tubes in the fully developed region. The averaged drag coefficients were shown in Table 2. It is found that the drag coefficient of the tube bundle with parallel tube layers is 15.62% larger than that of the tube bundle with crossed tube layers. The opposite inclination effect will lower the flow resistance of the tube bundles.

![Figure 10. Drag coefficient of straight tube bundles](image)

**Table 2.** Time-averaged flow resistance coefficient

| Tube bundle          | Drag coefficient | Difference  |
|----------------------|------------------|-------------|
| with crossed tube layers | 0.2209           | -           |
| with parallel tube layers | 0.2554           | 15.62%      |

3.3. Time-averaged heat transfer characteristics in straight tube bundles

Figure 11 shows the Nusselt number of the tube bundles with crossed tube layers and parallel tube layers. The horizontal coordinates are the row numbers of the tubes. It is found that the Nusselt number in the inlet region increases along the flow direction. The Nusselt number of the tube bundle with crossed tube
layers is smaller than that of the tube bundle with parallel tube layers in the fully developed region. The Nusselt numbers of the tubes near the inlet and outlet, which are influenced by the inlet effect and outlet effect, are neglected. The averaged Nusselt number of the tube bundle is calculated by averaging the Nusselt number of the tubes in the fully developed region. The averaged Nusselt number are shown in Table 3. The Nusselt number of the tube bundles with parallel tube layers in the fully developed region is 9.39% larger than that of the Nusselt number of the tube bundles with crossed tube layers. The opposite inclination effect will lower the heat transfer of the tube bundles.

![Nusselt number of straight tube bundles](image)

**Figure 11.** Nusselt number of straight tube bundles

**Table 3.** Time-averaged flow resistance coefficient

| Tube bundle                  | Nusselt number | Difference |
|------------------------------|----------------|------------|
| Tube bundle with crossed     | 119.56         | -          |
| Tube bundle with parallel    | 130.79         | 9.39%      |

**4. Conclusions**

The overall flow characteristics over helical tube bundles with neighboring layers coiled in the same direction and opposite direction were simulated. The detailed flow and heat transfer characteristics over straight tube bundles with parallel tube layers and crossed tube layers were numerically investigated. The influences of coiling direction on the cross flow and heat transfer characteristics over tube bundles was studied. The main conclusions are:

1. For a helical tube bundle with neighboring layers coiled in the same direction, the tangential velocity along the coiled circumferential direction could be observed obviously.
2. For a helical tube bundle with neighboring layers coiled in the opposite direction, there is no average tangential velocity of the whole flow filed. However, the steam lines or flow paths of the fluid are very complex, this may enhance the thermal mixing among different tube layers of the helical tube bundles.
3. The heat transfer over tube bundles with crossed tube layers is 9.39% smaller than that with parallel tube layers, the pressure drop over tube bundles with crossed tube layers is 15.62% smaller than that with parallel tube layers.

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