Boiling simulation and PEC evaluation of helically corrugated tubes in vertical grooves with different pipe diameters

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Abstract. The double tube structure is simulated, R32 in the inner helical corrugated tubes with vertical grooves is heated by the water flowing in the outer tube to make it boil. The influence of pipe diameter and number of vertical grooves on the performance of double tube is analyzed. The evaporation temperature of the refrigerant is 283K. The temperatures of inlet refrigerant and inlet water are 280K and 320K, respectively. The flow velocity of inlet refrigerant and inlet water are 1m/s and 5m/s, respectively. The simulation results show that, compared with the smooth pipe, when the inner pipe diameter is 3, 4, 5, 6, 7mm, the surface heat transfer coefficients of the double tube have been increased by 160%, 150%, 147%, 138%, and 130%, respectively. when PEC=1.691, the performance of the inner helical corrugated tube is the best, whose inner pipe diameter is 3 mm, outer pipe diameter is 6.2 mm, number of vertical grooves is 10.

1. Introduction

Heat exchangers were widely used heat transfer equipment, of which double tube heat exchanger was the most simple and applicable heat exchanger, due to its simple structure and low operating cost was widely used in pasteurization, reheating, preheating, boiler heating, and wastewater heating process [1], so the enhanced heat transfer of double tube heat exchanger was widely studied [2].

The research on the enhancement of double-tube heat exchangers focuses on the enhancement of the internal fluid flow. Some scholars have enhanced the heat transfer efficiency of double tube heat exchangers by changing the structure of the inner tube. Timothy et al [3] conducted an experimental study of advection and counterflow structures in a double tube helical heat exchanger and showed that there was no difference in the total heat transfer coefficient between the two flow modes. Wei et al [4,5] carried out a multi-objective optimization study of turbulent heat transfer in a new outward helical corrugated tube, and the results showed that a larger Reynolds number and helical corrugation height and smaller corrugation pitch were obtained when carrying out the pipe design, after which their numerical study of fully developed turbulence and heat transfer on the inner pipe side of the inner corrugated tube was carried out, and the results showed that the pipe with 200 mm pipe length and 20 mm inner pipe diameter, the pipe with 2 mm corrugation height and 3 mm corrugation pitch was a better...
geometric parameter, and the PEC of helically corrugated pipe was greater than that of transverse corrugated pipe.

The change in the structure of the inner tube increases the flow of fluid inside the tube, thus enhancing the heat transfer. Therefore, some scholars have thought of adding fillers to the inner tube to directly change the flow of fluid inside the tube to enhance heat transfer. Sheikholeslami et al [6] conducted an experimental study of flow and heat transfer in a waterless double tube heat exchanger with perforated and non-perforated discontinuous helical turbulators and showed that the friction coefficient and Nuss number decreased with increasing the opening ratio and pitch ratio. Thejaraju et al [7] conducted an experimental study of the turbulent behavior of a novel para winglet tape in an air-to-air double-pipe heat exchanger, and the results showed that the heat transfer in the enhanced tube of the novel para winglet tape was 1.09-2.69 times that of a normal tube. Hamed et al [8] performed a simulation study of a double pipe heat exchanger with a combined vortex generator and twisted tape and showed that the combined vortex generator increased the pressure drop while enhancing the heat transfer.

Current research has focused on convective heat transfer in double tube heat exchangers, and the techniques used to enhance heat transfer were generally the addition of helical lines and the addition of filler to the inner tube, and the pretreatment of the filler. In this paper, a new helical corrugated structure was proposed based on which the helical corrugated structure of the inner tube was cut, and the research was transferred from convective heat transfer to boiling heat transfer by CFD technology, and the PEC evaluation method was introduced to obtain the optimal structure under different inner pipe diameters to provide a theoretical basis for the new design of double tube heat exchangers.

2. Numerical simulation

2.1. Physical model and meshing
The structure of the inner helical corrugated tube with vertical grooves is shown in Figure 1. The total length of the inner helical corrugated tube is 300 mm, the inner tube inner diameter is 3, 4, 5, 6, 7 mm, the wall is 0.2 mm, the outer tube inner diameter is 6.2, 7.2, 8.2, 9.2, 10.2 mm, the height of the inner helical is 0.3 mm, where the height of the vertical grooves is consistent with the height of the inner helical, the number of vertical grooves is 0, 4, 8, 12, 14, 16, 18, 20. The double tube with 16 vertical grooves is shown in Figure 1(a) and Figure 1(b). The inner tube is an inner helical corrugated structure, and the boundary layer treatment is applied to the inner tube wall surface and the outer tube inner wall surface to better capture the flow boiling inside the tube. The boundary layer mesh was shown in Figure 1(c).

![Figure 1. Inner helical corrugated tube structure.](image-url)
2.2. Mixture Models
The Mixture model is chosen for the two-phase flow model, this model contains the continuity equation, momentum equation, and energy equation, which are as follows:

(1) Continuity equation:
\[
\nabla (\rho_m \vec{v}_m) = m \\
\vec{v}_m = \sum_{k=1}^{n} \alpha_k \rho_k \vec{v}_k \\
\rho_m = \sum_{k=1}^{n} \alpha_k \rho_k
\]
where \( \vec{v}_m \) — Unit Mass average speed, m/s; \( \rho_m \) — Mixing density, kg/m\(^3\); \( \alpha_k \) — Volume fraction of the kth phase; \( \rho_k \) — Density of the kth phase, kg/m\(^3\); \( \vec{v}_k \) — Velocity of the kth phase, m/s.

(2) Momentum equation:
\[
\nabla (\rho_m \vec{v}_m \vec{v}_m) = -\nabla q_m + \nabla [\mu_m (\nabla \vec{v}_m + (\nabla \vec{v}_m)^T)] \\
-\nabla (\alpha_p \rho_p \vec{v}_{dr.p} \vec{v}_{dr.p}) - \rho_m \beta_m g \Delta T
\]
where \( \vec{v}_{dr.p} \) — The drift speed of p in the second phase, m/s; \( \beta_m \) — Coefficient of volumetric thermal expansion.

(3) Energy equation:
\[
\nabla \sum_{k=1}^{n} (\alpha_k \vec{v}_k (\rho_k E_k + p)) = \nabla (K_{eff} \nabla T) + S_E
\]
where \( K_{eff} \) — Effective thermal conductivity, W/(m K); \( S_E \) — Energy source term for all phases, J.

Second phase volume fraction equation:
\[
\nabla (\alpha_p \rho_p \vec{v}_m) = -\nabla (\alpha_p \rho_p \vec{v}_{dr.p})
\]

Phase change model equation:
\[
S_{energy} = -ML \\
L = (h_q^s - h_p^s) \\
h_p^s = h_p^f + \int_{T_{ref,p}}^{T_{sat}} cp_p dT \\
h_q^s = h_q^f + \int_{T_{ref,q}}^{T_{sat}} cp_q dT
\]
where \( M \) — Mass transfer rate, mol/(m\(^2\) s); \( L \) — Latent heat of vaporization, J/kg; \( h_q^s \), \( h_p^s \) — The enthalpy of saturation of the first phase q and the second phase p, J/kg; \( h_q^f \), \( h_p^f \) — Enthalpy of generation of the first phase q and the second phase p, J/kg; \( T_{ref} \) — Reference temperature of refrigerant physical properties, °C; \( T_{sat} \) — Saturation temperature of the refrigerant, °C.

2.3. Turbulence model
The turbulence model is chosen as the RNG \( k - \varepsilon \) model with the standard near-wall treatment, which contains the following equations:

\[
\frac{\partial (\rho k \mu_d)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \frac{\mu}{\sigma_k} \frac{\partial h}{\partial x_i} \right) + G_k + G_k - \rho \varepsilon - Y_M + S_k
\]
\[
\frac{\partial (\rho \varepsilon)}{\partial x_i} = \frac{\partial}{\partial x_i} \left( \frac{\mu}{\sigma_\varepsilon} \frac{\partial \varepsilon}{\partial x_i} \right) + C_1 \frac{\varepsilon}{k} (G_k + G_{3\varepsilon} G_b) - C_2 \rho \frac{\varepsilon^2}{k} + S_\varepsilon
\]
where \( h \) — Surface heat transfer coefficient, W/(m\(^2\) K); \( \mu \) — Turbulent Viscosity, Pa S; \( k \) — Turbulent pulsation kinetic energy, J; \( G_k \) — Turbulent kinetic energy generated by laminar velocity gradients, J; \( G_b \) — Turbulent kinetic energy generated by buoyancy, J; \( \varepsilon \) — Turbulent pulsation kinetic energy dissipation rate; \( Y_M \) — Fluctuations from turbulent diffusion; \( C_1 \), \( C_2 \) — Constant Factor.

2.4. Boundary conditions
Boundary conditions of the model and algorithm setting steps:

1. Selected the time type to "steady" model.

2. The evaporation temperature of the refrigerant is 283K. The temperatures of inlet refrigerant and inlet water are 280K and 320K, respectively. The flow velocity of inlet refrigerant and inlet water are 1m/s and 5m/s, respectively.

3. The number of the iterations is 10000, the convergence scale is defined $10^{-7}$ and $10^{-6}$ for energy variables and flow variables respectively.

4. In the two-phase flow model, set the liquid phase of R32 as phase 1, the gas phase of R32 as phase 2, and the water in the outer tube as phase 3, and consider the effect of surface stress, activate the "implicit body force" option in the Mixture model, and select the boiling heat exchange model. Activate the "implicit body force" option in the Mixture model, and select the "Evaporation-Condensation" model in the Mixture model.

5. Both the inlets of the inner and outer tube are velocity inlets and the outlet is a pressure outlet. The inner tube is R32 liquid phase inlet and the outer tube is water inlet, where the volume fraction of water in the outer tube inlet is 100% and the flow of R32 and water is downstream.

6. The inner and outer tube wall material is copper, and the inner and outer tubes are coupled boundaries.

7. The SIMPLE algorithm is used to select Solve N-Phase Volume Fraction Equations, and the first-order upwind equations are used for the energy equation, volume equation, and turbulent kinetic energy equation.

3. Results and Discussion

3.1. Numerical simulation feasibility verification

The numerical simulation results were validated using the Liu and Winterton empirical correlation [9]. Boiling heat transfer is divided into nucleate boiling and membrane boiling and there are interactions. Based on the linear addition and subtraction of the two boiling heat transfer coefficients, they are corrected by the membrane boiling enhancement factor and the nucleate boiling weakening factor, respectively.

$$h_3 = [(E h_2)^2 + (S h_2)^2]^{0.5}$$  
$$h_2 = 0.023 R_e^{0.8} P_r^{0.4} \lambda / d$$  
$$h_1 = 55 P_f^{0.12} (-\log P_f)^{-0.55} M^{-0.5} Q^{0.67}$$  
$$E = [1 + x P_r (\rho_l / \rho_v - 1)]^{0.35}$$  
$$S = (1 + 0.055 E^{0.1} R_e^{0.16})^{-1}$$  
$$R_e = \rho d v / \mu$$  
$$P_r = C_p \rho / \lambda$$

where $h_3$ — Surface heat transfer coefficient, W/(m$^2$·K); $\lambda$ — Thermal conductivity of fluids, W/(m·K); $d$ — Equivalent diameter of the inner tube, m; $M$ — Relative molecular mass; $Q$ — Heat flow density, W/m$^2$; $\rho$ — Fluid Density, kg/m$^3$; $v$ — Fluid velocity, m/s; $\mu$ — Fluid viscosity, Pa·S; $C_p$ — Constant pressure specific heat capacity of the fluid, J/(kg·K).

The results of the comparison between simulated and theoretical values were shown in Table 1. It can be seen that the deviation between simulated and theoretical values is kept at 6%-10%, the results of present study are in good agreement with empirical correlation obtained data [9].
Table 1. Comparison of simulated value and theoretical value of helically corrugated tube in different number of vertical grooves.

| Number of grooves | Theoretical value | Simulated values | Deviation/% |
|-------------------|-------------------|-----------------|-------------|
| 0                 | 5 056             | 5 414           | 7           |
| 4                 | 5 093             | 5 522           | 8           |
| 8                 | 5 128             | 5 648           | 8           |
| 12                | 5 170             | 5 792           | 8           |
| 14                | 5 206             | 6 207           | 9           |
| 16                | 5 238             | 6 716           | 10          |
| 18                | 5 260             | 6 072           | 7           |
| 20                | 5 287             | 5 660           | 6           |

3.2. Surface heat transfer coefficient analysis

The surface heat transfer coefficient is the primary objective for the design of the inner tube reinforcement. The surface heat transfer coefficients of helically corrugated tubes in different numbers of vertical grooves were shown in Figure 2. The surface heat transfer coefficient increases and then decreases with the increase of the number of vertical grooves for each pipe diameter, and there is a maximum value, and the maximum value increases with the increase of the pipe diameter.

When the inner helical corrugated structure is added first, the heat transfer coefficients of the inner tube surface of 3, 4, 5, 6, and 7mm pipe diameters are 160%, 150%, 147%, 138%, and 130% of the corresponding pipe diameter smooth tube, respectively. On the one hand, the inner helical corrugated structure increases the heat transfer surface area of R32, which gives R32 a larger boiling surface. On the other hand, it is the internal helical structure that changes the flow of R32 in the tube, and the internal helical structure produces a secondary flow along the radial direction, and the flow of R32 in the tube is enhanced [10]. The maximum values of the heat transfer coefficients on the surface of the inner tube for 3, 4, 5, 6, and 7mm pipe diameters were 175%, 165%, 157%, 148%, and 140% of the corresponding smooth tubes when the vertical grooves structure was added later. With the addition of the vertical grooves structure, the inner helical structure is cut along the radial direction, and the flow at the cut of the vertical grooves structure and the inner helical corrugated structure is enhanced and the degree of turbulence is strengthened, but with the further increase of the vertical grooves structure, the radial length of the inner helical structure decreases, the radial secondary flow is weakened, and the degree of turbulence decreases, so the surface heat transfer coefficient of the inner tube increases and then decreases [11].

Vertical groove structure for different pipe diameter of the inner helically corrugated tubes surface heat transfer coefficient strengthening degree is not the same, 3, 4, 5, 6, 7mm pipe diameters of the degree of strengthening are 175%, 165%, 157%, 148%, 140%. The reason for this phenomenon is mainly that when the inner pipe diameter increases, the heat provided by the outer tube water increases, and the heat flow density on the inner tube wall surface increases, and the increase in heat flow density on the inner tube wall surface leads to the formation of more vaporization cores on the inner tube wall surface, which generates more bubbles and promotes boiling heat transfer in the inner tube [12].
3.3. Pressure drop analysis

The reinforced structure of the inner tube can lead to a large pressure drop loss in the inner tube. As shown in Figure 3, the pressure drop of the inner tube of helically corrugated tubes varies with different pipe diameters under different numbers of vertical grooves. When the diameter of the tube is certain, the pressure drop of the inner tube increases and then decreases with the increase of the number of vertical grooves, and there is a maximum value, and the maximum value of the pressure drop of the inner tube decreases with the increase of the diameter of the inner tube.

When the internal helical corrugated structure is added first, the pressure drop of the inner tube for 3, 4, 5, 6, and 7 mm pipe diameters is 240%, 230%, 207%, 202%, and 200% of the corresponding pipe diameters of the smooth pipe, respectively. When the vertical groove structure is added later, the maximum pressure drop of the inner tube of 3, 4, 5, 6, and 7 mm pipe diameters is 260%, 255%, 248%, 240%, and 220% of the corresponding smooth tube, respectively. The added inner helical corrugated structure obviously increases the pressure drop of the inner tube, because the fluid at the wall produces radial secondary flow along with the inner helical structure, which increases the turbulence at the wall, but with the later addition of the vertical groove structure cutting the inner helical structure, the degree of turbulence at the cut is strengthened, and gradually forms a vortex, and the number of vertical grooves further increases, the secondary flow weakens, and the vortex gradually disappears, so the pressure drop of the inner tube. The pressure drops in the inner tube increase and then decrease with the increase of the number of vertical grooves [13].

The pressure drop of the helically corrugated tube in the vertical groove decreases with the increase of the inner pipe diameter, according to the expression of frictional resistance, Fanning's formula ($h_f = \frac{\varepsilon}{d} \left( \frac{v^2}{2} \right)$, $h_f$ is the along-travel resistance loss, $\varepsilon$ is the along-travel friction coefficient, $l$ is the pipe length, $d$ is the pipe diameter, and $v$ is the average velocity of the fluid in the pipe), it is known that, when other conditions remain constant, the smaller the pipe diameter, the along-travel resistance. The greater the friction between R32 and the wall of the inner pipe, so the pressure drop in the inner pipe decreases with the increase of the inner pipe diameter [14].

Figure 2. Heat transfer coefficient of helically corrugated tubes surface in different pipe diameters with different number of vertical grooves.
3.4. Performance Evaluation

PEC [15] is a standard method of performance evaluation defined as the ratio of the reinforcement effect of the helically corrugated tube in the vertical grooves to the pump power loss, the value of which determines the relationship between the surface heat transfer coefficient and the pressure drop.

\[
\text{PEC} = \frac{U_C}{U_S} \left[ 1 - \frac{\Delta P_C + \Delta P_{AC}}{\Delta P_S + \Delta P_{AS}} \right]^{1/3}
\]  

(22)

where \( U_C \) — Total heat transfer coefficient of a double tube heat exchanger with the corrugated tube, \( W/(m^2 \cdot K) \); \( U_S \) — Total heat transfer coefficient of a two-tube heat exchanger with smooth tubes, \( W/(m^2 \cdot K) \); \( \Delta P_C \) — Pressure drop in the double tube heat exchanger in the internal corrugated tube, Pa; \( \Delta P_{AC} \) — Pressure drop of the heat exchanger on one side of helically corrugated tube, Pa; \( \Delta P_S \) — Pressure drop of the double tube heat exchanger on the inner smooth tube, Pa; \( \Delta P_{AS} \) — Pressure drop in a double tube heat exchanger with smooth tubes on the circular side, Pa.

The PEC of each specification of vertical grooved internal helical corrugated tubes with an internal (external) wall thickness of 0.2mm and an internal helical height of 0.3mm is shown in Figure 4. It can be seen from the figure that when the inner diameter of the inner tube is 3 mm, the inner diameter of the outer tube is 6.2 mm, and the number of vertical grooves is 10, the PEC value of the helical corrugated tube in the vertical groove is the largest; When the inner diameter of the inner tube is 4mm, the inner diameter of the outer tube is 7.2mm, and the number of vertical grooves is 12, the PEC value of the helical corrugated tube in the vertical groove is the largest; When the inner diameter of the inner tube is 5mm, the inner diameter of the outer tube is 8.2mm and the number of vertical grooves is 16, the PEC value of the helical corrugated tube in the vertical groove is the largest; When the inner diameter of the inner tube is 6 mm, the inner diameter of the outer tube is 9.2 mm and the number of vertical grooves is 18, the PEC value of the helical corrugated tube in the vertical groove is the largest; When the inner diameter of the inner tube is 7mm, the inner diameter of the outer tube is 10.2mm, and the number of vertical grooves is 20, the PEC value of the helical corrugated tube in the vertical groove is the largest.
Figure 4. PEC values of internal helical corrugated tubes for different numbers of vertical grooves and different pipe diameters.

4. Conclusion
Based on boiling simulation of helically corrugated tubes in vertical grooves with different pipe diameters, the surface heat transfer coefficient and pressure drop of the inner helical corrugated tubes with different pipe diameters were analyzed, and the inner helical structure was optimized by PEC evaluation, the results are as follows:

1. The inner helical corrugated structure increases the heat transfer surface area and circulation area (per unit length) of the inner tube and produces radial secondary flow. Compared with the inner helical corrugated tubes, the surface heat transfer coefficient and pressure drop of the vertical groove inner helical corrugated tubes both increase and then decrease with the increase of the number of vertical grooves.

2. Compared with the smooth tube, the surface heat transfer coefficients of tubes, whose pipe diameters are 3, 4, 5, 6, and 7 mm, are increased to 160%, 150%, 147%, 138%, 130%, respectively. Compared with the smooth tube, the pressure drop of tubes, whose pipe diameters are 3, 4, 5, 6, and 7 mm, are increased to 240%, 230%, 221%, 202%, and 200%, respectively.

3. According to PEC evaluation, the optimal structures with the inner (outer) tube wall thickness of 0.2 mm and inner helical height of 0.3 mm are as follows: The inner tube is 3 mm, the outer tube is 6.2 mm, and the number of vertical grooves is 10; The inner tube is 4 mm, the outer tube is 7.2 mm, and the number of vertical grooves is 12; The inner tube is 5 mm, the outer tube is 8.2 mm, and the number of vertical grooves is 16; The inner tube is 6 mm, the outer tube is 9.2 mm, and the number of vertical grooves is 18; The inner tube is 7 mm, the outer tube is 10.2 mm, and the number of vertical grooves is 20.

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