**Boiling heat transfer of R32 and R22 in spiral corrugated tube with a vertical groove**

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**Abstract.** In this paper, a three-dimensional steady-state boiling model of the spiral corrugated tube with a vertical groove is established by ANSYS Fluent 18.0. The model is a double tube structure, the water flow in the outer tube, the refrigerant flow in the inner tube, and the inner tube is a new type of spiral corrugated tube with a vertical groove inside the groove. Firstly, the physical properties difference between R32 and R22 is analyzed. Subsequently, the boiling heat transfer characteristics of R22 and R32 in the vertical groove inner spiral corrugated tube with different pipe diameters are studied, including the velocity of the inner pipe, the Reynolds number of the outer pipe, and the saturation temperature. Therefore these studies have laid the foundation for the application of R32 in double-tube heat exchangers.

1. **Introduction**

Comparing the physical parameters of R32 and R22, it can be found that the relative molecular mass of R32 is 60% of that of R22, and according to the conversion relationship between the filling volume and the relative molecular mass, it is clear that the filling volume of R32 is only 60% of that of R22. The dynamic viscosity of R32 is 69% of that of R22, and smaller dynamic viscosity facilitates smaller heat transfer pressure drop. In terms of environmental protection, the ODP (ozone depletion potential) of R32 and R22 are similar, and the GWP (global warming potential) is 675, which is only 35% of that of R22, which is conducive to CO₂ emission reduction. It has low flammability and is rated A2L by ASHRAE, and the production cost of R32 is low, and its production process is similar to that of R22 [1].

Domestic and foreign studies on boiling heat transfer of R22 and R32 mainly focus on smooth tubes and micro-fin tubes. Zhang et al [2] conducted an experimental study of boiling heat transfer of R22 in smooth tubes and inner groove tubes. It was found that the boiling heat transfer coefficient of the inner groove tube was 1.32-1.475 compared with that of the smooth tube, where the distance between the inner groove and the angle of the inner recesses were the main factors of boiling enhancement. Zan et al [3] conducted experiments on boiling heat transfer of R22 in a 5 mm smooth tube and micro-fin tube. He et al [4] conducted an experimental study of boiling of R32 in 5 and 7 mm horizontal smooth tube and micro-fin tubes. It was found that the boiling heat transfer coefficient of both 5 mm smooth tube
and micro-fin tube were about 20% higher than those of 7 mm under the condition that the heat flux was kept constant, and the smaller pipe diameters allow for larger boiling heat transfer coefficients, and the heat flux was the main factor influencing the boiling heat transfer in the smooth pipe. The experimental study of R32 flow pattern and boiling heat transfer in a horizontal finned tube of 3.48 mm equivalent inner diameter was conducted by Daisuke et al [5]. The results showed that the flow pattern of R32 in the horizontal finned tube was divided into wavy plug flow and annular flow, and the nucleation boiling in boiling heat transfer was mainly dependent on the liquid film inside the groove. Then it conducted an experimental study of boiling heat transfer of R32 in spiral fluted finned tubes with equivalent diameters of 2.1, 2.6, and 3.5 mm. It was found that the boiling heat transfer coefficient of R32 increased with decreasing pipe diameter at constant heat flux, and the friction pressure drop increased greatly with decreasing pipe diameter.

2. In this paper, the flow boiling characteristics of R22 and R32 in spiral corrugated tube with vertical groove are analyzed based on CFD Numerical simulation

2.1. Physical model and grid division

The structure of the internal spiral corrugated tube with vertical groove is shown in Figure 1. The structural parameters of the corrugated tube include the tube length (L=300mm), inner diameter (D=5mm), tube-wall thickness (tl=0.2mm), corrugation height (hl=0.3mm), corrugation pitch (pl=5mm), and the outer tube inner diameter (D=8.2mm), where the vertical groove height and the internal corrugation height keep the same, the number of vertical groove is 0, 4, 8, 12, 16, 18, 20. The double tube with 16 vertical grooves is shown in Figure.1a and Figure.1b the inner tube is an inner helical corrugated structure. To better capture the flow boiling inside the tube, the boundary layer treatment is applied to the inner tube wall surface and the outer tube inner wall surface. The boundary layer mesh was shown in Figure.1(c).

![Figure 1. Inner spiral corrugated tube structure.](image)

2.2. Mixture model

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:

\[
\frac{\partial (\rho_m \vec{v}_m)}{\partial t} + \nabla (\rho_m \vec{v}_m \vec{v}_m) = m, \tag{2-1}
\]

\[
\vec{v}_m = \frac{\sum_{k=1}^{n} \alpha_k \rho_k \vec{v}_k}{\rho_m}, \tag{2-2}
\]

\[
\rho_m = \sum_{k=1}^{n} \alpha_k \rho_k, \tag{2-3}
\]
Where $\bar{v}_m$ is mass average velocity, m/s; $\rho_m$ is mixing density, kg/m$^3$; $\alpha_k$ is phase k volume fraction, $\rho_k$ is phase k density, kg/m$^3$; $\bar{v}_k$ is phase k velocity, m/s.

(2) momentum equation:
\[
\frac{\partial (\rho_m \bar{v}_m)}{\partial t} + \nabla (\rho_m \bar{v}_m \bar{v}_m) = -\nabla q_m + \nabla (\mu_m (\nabla \bar{v}_m + \nabla \bar{v}_m^T)) - \nabla (\alpha_p \rho_p \bar{v}_{dr,p} \bar{v}_{dr,p}) - \rho_m \rho_m g \Delta T \quad (2-4)
\]
where $\bar{v}_{dr,p}$ is the drift speed of the second phase p and $\beta_m$ is coefficient of volumetric thermal expansion
\[
\bar{v}_{dr,p} = \bar{v}_{qp} - \sum_{k=1}^{n} \frac{\alpha_k \rho_k}{\rho_m} \bar{v}_{qk}, \quad (2-5)
\]
\[
\bar{v}_{qp} = \bar{v}_p - \bar{v}_q, \quad (2-6)
\]

(3) energy equation
\[
\frac{\partial (\sum_{k=1}^{n} \alpha_k \rho_k E_k)}{\partial t} + \nabla (\sum_{k=1}^{n} \alpha_k \bar{v}_k (\rho_k E_k + p)) = \nabla (K_{eff} \nabla T) + S_E, \quad (2-7)
\]
Where $K_{eff}$ is effective thermal conductivity, W/(m·K) and $S_E$ is energy source term for all phases, J second phase volume fractional equation:
\[
\frac{\partial (\alpha_p \rho_p \bar{v}_m)}{\partial t} + \nabla (\alpha_p \rho_p \bar{v}_m) = -\nabla (\alpha_p \rho_p \bar{v}_{dr,p}), \quad (2-8)
\]

phase change model equation:
\[
S_{energy} = -ML, \quad (2-9)
\]
\[
L = (h_q^s - h_p^s), \quad (2-10)
\]
\[
h_p^s = h_p^f + \int_{T_{ref}}^{T} cp_p dT, \quad (2-11)
\]
\[
h_q^s = h_q^f + \int_{T_{ref}}^{T} cp_q dT, \quad (2-12)
\]
Where $M$ is mass transfer rate, mol/(m$^2$·s); $L$ is the latent heat of vaporization, J/kg; $h_q^s, h_p^s$ is the enthalpy of saturation of the first phase q and the second phase p, J/kg; $h_q^f, h_p^f$ is the enthalpy of generation of the first phase q and the second phase p, J/kg; $T_{ref}$ is reference temperature of refrigerant physical properties, °C; $T_{sat}$ is saturation temperature of the refrigerant, °C.

2.3. Turbulence model

The turbulence model is chosen as the RNG $k-e$ model with the standard near-wall treatment, which contains the following equations:
\[
\frac{\partial (\rho_k u_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial \bar{u}_i}{\partial x_i} \right] + G_k + \epsilon - \frac{\rho_k}{Y_M} + S_k, \quad (2-13)
\]
\[
\frac{\partial (\rho_k e)}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \left( \mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial \varepsilon}{\partial x_i} \right] + C_1 \varepsilon \left( \frac{G_k}{k} + G_3 \rho_b \right) - C_2 \rho_b \varepsilon^2 + \frac{S_k}{k}, \quad (2-14)
\]
Where $h$ is surface heat transfer coefficient, W/(m$^2$·K); $\mu$ is turbulent viscosity, Pa·s; $k$ is turbulent pulsation kinetic energy, J; $G_k$ is turbulent kinetic energy from laminar velocity gradients, J; $G_b$ is turbulent kinetic energy from buoyancy, J; $\varepsilon$ is turbulent pulsation kinetic energy dissipation rate, $Y_M$ is fluctuations from turbulent diffusion, and $C_1$ and $C_2$ are constant factors.

2.4. Boundary conditions and algorithm settings

Boundary conditions of the model and algorithm setting steps:

(1) In the two-phase flow model, the liquid phase of R32 is set as phase 1, the gas phase of R32 is set as phase 2, and the water in the outer tube is set as phase 3. Due to the effect of surface stress, the "implicit body force" option in the Mixture model is activated, and the "Evaporation-Condensation" model is selected in the Mixture model for the boiling heat exchange model.

(2) Both the inlets of the inner and outer tube are velocity inlets, and the outlets are pressure outlets. 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.
(3) The inner and outer tube wall material is copper, and the inner and outer tube are coupled boundaries.

(4) The Semi-Implicit Method for Pressure Linked Equations (SIMPLE) algorithm is chosen to couple the pressure and velocity, and the least squares cell-based method is used for the discretization of the gradient, it is used to select Solve N-Phase Volume Fraction Equations, and the first-order upwind equation is used for the energy equation, volume equation, and turbulent kinetic energy equation.

3. Results and analysis

3.1. Numerical simulation feasibility verification

The numerical simulation result is validated by the Liu and Winterton correlation [6]. Boiling heat transfer is divided into nucleate boiling and membrane boiling. Based on the linear addition and subtraction of the two boiling heat transfer coefficients, they are corrected by the film boiling enhancement factor and the nucleate boiling weakening factor, respectively.

\[
h_3 = [(Eh_2)^2 + (Sh_1)^2]^{0.5},
\]

\[
h_2 = 0.023Re^{0.8}Pr^{0.4}d^{-0.2},
\]

\[
h_1 = 55Pr^{0.12}(-logPr)^{-0.55}M^{-0.5}Q^{0.67},
\]

\[
E = \left[1 + xPr\left(\frac{p_r}{p_w} - 1\right)\right]^{0.35},
\]

\[
S = \left(1 + 0.055E^{0.1}Re^{0.16}\right)^{-1},
\]

\[
Re = \frac{\rho d v}{\mu},
\]

\[
P_r = \frac{c_p \mu}{\lambda},
\]

Where \(h_3\) is surface heat transfer coefficient, W/(m²·K); \(\lambda\) is the thermal conductivity of fluids, W/(m·K); \(d\) is equivalent diameter of the inner tube, m; \(M\) is relative molecular mass, kg/mol; \(Q\) is heat flow density, W/m²; \(\rho\) is the fluid density, kg/m³; \(v\) is fluid speed, m/s; \(\mu\) is fluid viscosity, Pa·s; \(c_p\) is the specific heat at a constant pressure of the fluid, J/(kg·K).

When refrigerant saturation temperature is 283K, water velocity is 5m/s, water inlet temperature is 320K, refrigerant inlet temperature is 280K, the results of simulation values and theoretical values from the Liu and Winterton correlation are shown in Table 1. It is found that the surface heat transfer coefficient maximum deviation between the simulation values and theoretical values is 9%, the results of present study are in good agreement with the Liu and Winterton correlation [6].

| Refrigerant velocity m/s | Theoretical value W/(m²·K) | Simulation value W/(m²·K) | Deviation % |
|-------------------------|---------------------------|---------------------------|-------------|
| 1                       | 5056                      | 5414                      | 7           |
| 2                       | 7697                      | 8235                      | 7           |
| 3                       | 11316                     | 11768                     | 4           |
| 4                       | 15503                     | 16278                     | 5           |
| 5                       | 18775                     | 20465                     | 9           |

3.2. Inner tube velocity

The inner tube velocity is an important factor affecting the heat transfer effect. The trend of the surface heat transfer coefficient of R32 and R22 with the velocity of the inner pipe for inner pipe diameters 3, 4, 5, 6, and 7 mm as shown in Figure2A, and the trend of inner tube pressure drop with inner tube velocity for R32 and R22 with inner tube diameters of 3, 4, 5, 6 and 7 mm as shown in Figure2B.

It is found that the surface heat transfer coefficient of R32 and R22 side are proportional to the inner
tube velocity at a saturation temperature of 283K and an outer tube velocity of 5m/s. The maximum and minimum increase of surface heat transfer coefficient of R32 side are 181% and 176%, respectively. The maximum and minimum increase of surface heat transfer coefficient of R22 side are 204% and 190%, respectively. Both of them increase with the increase of tube diameter, the reason for this phenomenon is that with the increase of tube diameter, the increase of inner tube velocity strengthen the convection of the inner tube and the disturbance of the bubbles, but enhance the boiling heat exchange in the tube.

And due to the difference between the liquid and gas phase densities of R32 and R22, the surface heat transfer coefficient of R32 is 1.3 times and 1.1 times that of R22 at inner tube velocities of 3 m/s and 5 m/s, respectively. The pressure drop of the inner tube increase with the increase of the inner tube velocity, the disturbance of the inner tube wall surface increases, the friction between R32, R22 and the inner tube wall surface increases, and the shear stress of the inner tube wall surface increases [7, 8]. At the inner tube velocity of 1-3 m/s, the inner tube pressure drop between R32 and R22 is close; however, at the inner tube velocity of 4-5 m/s, the inner tube pressure drop of R22 is 1.1 times that of R32.

Figure 2. Relationship between surface heat transfer coefficient, inner tube pressure drop, and inner tube velocity for different tube diameters.

3.3. Reynolds number outside the tube
Reynolds number characterizes the flow of the fluid. The trend of surface heat transfer coefficient of R32 and R22 with Reynolds number outside the tube for inner tube diameters of 3, 4, 5, 6, and 7 mm as shown in Figure 3A, and the trend of inner tube pressure drop of R32 and R22 with Reynolds number outside the tube for inner tube diameters 3, 4, 5, 6 and 7 mm as shown in Figure 3B.

It is found that the surface heat transfer coefficients of R32 and R22 side are proportional to the Reynolds number outside the tube at a saturation temperature of 283K and an inner tube velocity of 1m/s, The maximum and minimum increase of surface heat transfer coefficient of R32 side are 34% and 24%, respectively. The maximum and minimum increase of surface heat transfer coefficient of R22 side are 27% and 16%, respectively. When the Reynolds number outside the tube increases to 19000, R32 has stronger heat transfer effect compared with R22, and the surface heat transfer coefficient is about 1.48 times of R22. The reason is that when the Reynolds number outside the tube increases, the heat provided by the outer tube increases too, and more refrigerant boils in the inner tube, so the heat transfer coefficient of R32 and R22 side surfaces all increase with the Reynolds number outside the tube. When the Reynolds number outside the tube changes, the pressure drop of the inner tube of R32 and R22 changes very little and the change can be neglected, but when the Reynolds number outside the tube increases to 19000, the pressure drop of the inner tube of R22 is 1.21 times that of R32, at the same time, the pressure drop of the inner pipe increases with the decrease of the pipe diameter. Both R32 and R22 have the maximum inner pipe pressure drop at 3 mm pipe diameter and are 225% and 204% of the pressure drop at 7 mm pipe diameter, respectively. According to the expression of frictional resistance, Fanning’s formula:
\[ h_f = \frac{1}{4} \left( \frac{\nu^2}{l} \right) \]

It shows that, in other conditions remain unchanged, the larger the diameter of tube and the smaller the resistance along the way, which lead to the smaller the friction between R32 and the wall of inner tube, so the pressure drop of inner tube increases with the tube diameter decreases [9, 10].

![Figure 3](image_url)

**Figure 3.** Relationship between surface heat transfer coefficient, inner tube pressure drop, and outer tube Reynolds number for different tube diameters.

### 3.4. Saturation temperature

Saturation temperature is an important factor affecting boiling heat transfer. The trend of surface heat transfer coefficient of R32 and R22 with saturation temperature for inner pipe diameters 3, 4, 5, 6, and 7 mm as shown in Figure 4A, and the trend of inner tube pressure drop of R32 and R22 with saturation temperature for inner tube diameters 3, 4, 5, 6 and 7 mm as shown in Figure 4B.

In the case of inner tube flow rate of 1 m/s and outer tube flow rate of 5 m/s, it is found that as the saturation temperature increases, the surface heat transfer coefficient of R32 and R22 side of the slightly decreases and decrease stays below 10%, at the same time the surface heat transfer coefficient of R32 is always about 1.5 times that of R22. Taking R32 as an example, the reason for this phenomenon is that the generation and detachment of bubbles during the phase change of the inner tube is the key point, the factors that affect the growth and detachment of bubbles from the wall are surface stress, the buoyancy of the fluid and resistance of the fluid. When the saturation temperature increases, the reduction of surface stress is conducive to the generation of bubbles with a smaller diameter and thus break away from the wall, and bubble detachment will lower the wall temperature, which is conducive to heat transfer [11]. The increase in gas density is much greater than the decline in liquid density, it can be judged that the effect of fluid buoyancy on the bubble is decreasing, the decline in liquid viscosity leads to the effect of fluid resistance on the bubble is decreasing. At the same time, owing to the decrease of thermal conductivity, the increase of gas density and the decrease of liquid density lead to the increase of gas phase velocity, the decrease of liquid phase velocity and the decrease of average velocity of mixed phase, which lead to the weakening of convective heat transfer.

It can be seen that the promotion effect on the heat transfer effect in this experiment is slightly less than the inhibition effect, and so is R22, so the side surface heat transfer coefficient of R32 and R22 decreases slightly with the increase of saturation temperature. The pressure drop in the inner tube of R32 and R22 increases slightly with the decrease of saturation temperature, because with the increase of saturation temperature, the liquid viscosity decreases slightly, the wall shear stress increases and the wall disturbance increases, so the pressure drop in the inner tube increases slightly [12].
Figure 4. Relationship between surface heat transfer coefficient, inner tube pressure drop, and saturation temperature for different tube diameters.

4. Conclusion
The following conclusions were obtained from the three-dimensional steady-state simulation of the boiling flow of R32 and R22 in the casing:

(1) The surface heat transfer coefficient of R32 and R22 and the pressure drop of the inner tube are proportional to the flow velocity of the inner tube and increase with the increase of the tube diameter. When the flow velocity of the inner tube is 1-3 m/s, the surface heat transfer coefficient of R32 is about 1.3 times that of R22. When the velocity of the inner tube is 4-5 m/s, although the pressure drop of the two inner tubes is similar, that is, the pressure drop in the tube of R22 is 1.1 times that of R32, but the surface heat transfer coefficient of R22 is about 1.3 times that of R32.

(2) The surface heat transfer coefficient of R32 and R22 is proportional to the Reynolds number outside the tube, and the change of pressure drop of the inner tube is not related to the Reynolds number outside the tube, but the pressure drop of the inner tube decreases with the increase of the tube diameter. When the Reynolds number outside the tube increases to 19000, the surface heat transfer coefficient of R32 is about 1.48 times that of R22, while the pressure drop of the inner tube of R22 is about 1.21 times that of R32. R32 has a larger surface heat transfer coefficient and smaller pressure drop of the inner tube compared with R22.

(3) The surface heat transfer coefficient of R32 and R22 decreases slightly with the increase of saturation temperature, and the pressure drop of both inner tubes increases slightly with the decrease of saturation temperature. When the saturation temperature increases from 273 K to 283 K, the surface heat transfer coefficient of R32 is always about 1.5 times that of R22, and the pressure drop of the inner tube of R22 is about 1.25 times that of R32. And compared with R22, R32 has a smaller inner tube pressure drop.

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