CFD-based investigation of R22 flow boiling characteristics in corrugated pipes

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Abstract. Double pipes are widely used in the food industry. In order to investigate the influence of different corrugation heights on the heat transfer performance of the corrugated pipe, the corrugated tubes with different sizes have been analyzed. The influence of different corrugation heights on the heat transfer coefficient and the pressure drop has been studied. An effect of the refrigerant velocity on the heat transfer coefficient and the pressure drop has been analyzed. The results show that the highest heat transfer coefficient takes place at the corrugation height 3.3mm. With an increase of the refrigerant velocity, the heat transfer coefficient and the pressure drop grow.

1. Introduction
The DPHx (Double-pipe Heat Exchanger) is one of the most widely used heat exchangers in industries with low fluid flow rates, such as the food industry. DPHx is widely used in industry mainly because of their simple geometry and cost effectiveness, but in industry it is necessary to improve heat transfer performance to achieve energy savings and emission reductions. Therefore, in order to improve the heat transfer performance, fins are inserted in the fluid flow process, or the surface structure of a single tube is modified to increase the heat transfer area, resulting in an increase in convective heat transfer coefficient. The corrugated pipe Heat Exchanger in the DPHx, due to an excellent heat transfer performance compared to other heat exchangers, which is widely used in the field of food quick-freezing, milk and juice sterilization and other processes, especially in the application of cooling, quick-freezing, preheating, reheating, cooking heating and wastewater heat treatment processes. Ouyang [1] et al. found by experiment that the heat transfer coefficient of the threaded tube with increased external cavity structure was 60%-120% higher than that of the smooth tube. Pesteii [2] et al. modified the heat exchanger tube structure and investigated the semi-elliptical convex wave joint geometry by numerical simulation, and found to be quite effective in enhancing heat transfer. Han [3] et al. studied the effect of the geometric parameters of the bellows on the pressure drop and concluded that the pressure drop decreases with increasing pitch P and increases with increasing corrugation height H. Deng [4] et al. investigated the heat transfer performance of semi-elliptical convex wave joints based on CFD, and
found that the corrugated tube with semi-elliptical structure has excellent heat transfer effect. Gorman [5] et al. verified the heat transfer performance of an internally corrugated pipe structure at Reynolds numbers ranging from 420 to 2000 laminar flow conditions by numerical simulations.

In this paper, the effect of different geometrical parameters of female corrugated pipe on heat transfer in turbulent flow region was analyzed using R22 as refrigerant, and a three-dimensional numerical model was designed to simulate the geometrical parameters of female corrugated pipe to compare the flow performance of fluid in each inner pipe and annular hollow side, and to investigate the effect of corrugation shape parameters on heat transfer performance and pressure drop.

2. Numerical model

2.1. Physical model

The corrugated tube is presented in Figure 1. Stainless steel is selected as the material of the corrugated pipe. The outer tube diameter of corrugated pipe is 39mm. The inner tube diameter of corrugated tube is 25mm. The wall thickness of both inner tube and outer tube are 1.5mm. The corrugated length is 2000mm. The double pipe parameters are shown in Table 1. The corrugation heights are 0mm, 2.2 mm, 3.3 mm, and 4.4 mm, respectively.

![Corrugated tube](image)

**Figure 1.** Corrugated tube.

| Case | Diameter (D, mm) | Height (H, mm) | Pitch (P, mm) | Width (W, mm) | H/D | T (H2O) (℃) | Velocity (V, m/s) |
|------|-----------------|----------------|---------------|---------------|-----|-------------|------------------|
| 1    | 22              | -              | -             | -             |     | 60          | 1.0              |
| 2    | 22              | 2.2            | 25            | 4.4           | 0.05| 60          | 1.0              |
| 3    | 22              | 3.3            | 25            | 3.6           | 0.10| 60          | 1.0              |
| 4    | 22              | 4.4            | 25            | 2.8           | 0.20| 60          | 1.0              |

2.2. Mesh division

In this study, an unstructured grid is used. The grid cells are tetrahedral. The fineness of the grid is checked. The accuracy is confirmed to be as required. Due to the large temperature gradient variation
between the inner and outer fluids at the pipe wall, the grid at its boundary is to be encrypted. The boundary layer can capture the temperature and velocity variation of the fluid at the wall. The mesh of corrugated tube is shown in Figure 2.

Figure 2. Grid system of corrugated tube.

2.3 Governing equations
The Mixture Model is chosen as the multiphase flow model. The Mixture Model solves the continuity equation of the mixed phase, the momentum equation of the mixed phase, the energy equation of the mixed phase, and the volume fraction of the second phase.

Continuity equation:

\[ \nabla (\rho m \bar{v}_m) = m \]  
\[ \bar{v}_m = \sum_{k=1}^{n} \alpha_k \rho_k \bar{v}_k \]  
\[ \rho_m = \sum_{k=1}^{n} \alpha_k \rho_k \]  

where \( \bar{v}_m \) is the mass-averaged velocity, \( \rho_m \) is the mixing density, \( \alpha_k \) is the volume fraction of the kth phase, \( \rho_k \) is the density of the kth phase, and \( \bar{v}_k \) is the velocity vector of the kth phase.

Momentum equation:

\[ \nabla (\rho m \bar{v}_m \bar{v}_m) = -\nabla \rho + \nabla [\mu_m (\nabla \bar{v}_m + \nabla \bar{v}_m^T)] + \rho_m g + \bar{F} + \nabla \left( \sum_{k=1}^{n} \alpha_k \rho_k \bar{v}_{\text{dr},k} \bar{v}_{\text{dr},k} \right) \]  

where \( n \) is the number of phases, \( \bar{F} \) is the bulk force, and \( \mu_m \) is the mixed viscosity.

\[ \mu_m = \sum_{k=1}^{n} \alpha_k \rho_k \]  

\( \bar{v}_{\text{dr},k} \) is the drift velocity vector of the kth phase.

\[ \bar{v}_{\text{dr},k} = \bar{v}_m - \bar{v}_m \]  

Energy equation:

\[ \nabla \sum_{k=1}^{n} \left( \alpha_k \rho_k (E_k + p) \right) = \nabla (K_{\text{eff}} \nabla T) + S_E \]  

where \( K_{\text{eff}} \) is the effective thermal conductivity, \( S_E \) is the energy source term for all phases.

Second phase volume fraction equation:

\[ \nabla \sum_{k=1}^{n} \left( \alpha_k \rho_k (p_e + p) \right) = \nabla \left( K_{\text{eff}} \nabla T \right) + S_E \]  

\[ \bar{v}_{\text{dr},p} = \bar{v}_{qp} - \sum_{k=1}^{n} \frac{\alpha_k \rho_k}{\rho_m} \bar{v}_{qk} \]  

\[ \bar{v}_{qp} = \bar{v}_p - \bar{v}_q \]  

The turbulence model is chosen as the RNG model with standard near-wall treatment, which contains the following equations:

\[ \frac{\partial (\rho k \mu_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \left( \frac{\mu_i}{\sigma_k} \right) \frac{\partial h}{\partial x_i} \right] + G_k + G_k - \rho_k - Y_m + S_k \]
\[
\frac{\partial (\rho \varepsilon)}{\partial x_i} = \frac{\partial}{\partial x_i} \left[ \left( \frac{\mu_i}{\sigma_i} \frac{\partial \varepsilon}{\partial x_i} \right) + C_1 \frac{\varepsilon}{K} (G_k + G_3 \varepsilon G_b) - C_2 \rho \varepsilon^2 + S_{\varepsilon} \right]
\]  

(12)

2.4. Boundary conditions and algorithm settings

R22 flows in the inner tube, water flows in the outer tube.

1. The inlet is set as the velocity boundary and the outlet is set as the pressure boundary, where the inner pipe wall system coupling boundary and the outer pipe wall surface is set as adiabatic.

2. The effect of surface stress is taken into account in the model and the option "implicit body force" is activated in the Mixture Model.

3. Copper is the material of the tube wall.

4. The liquid phase of R22 is set as the first phase, the gas phase of R22 is set as the second phase and the water in the outer tube is set as the third phase in the Model.

5. This simulation algorithm is set up: COUPLE algorithm is chosen for the pressure-velocity coupling and QUICK algorithm is chosen for the volume fraction equation.

2.5. Model validation

For validation of numerical results with experimental results, CFD results of surface heat transfer coefficient is evaluated with experimental data. Water mass flux is kept 0.6 m³/h. The evaporation temperature of the refrigerant is kept 278 K. The refrigerant mass flux is changed from 150 kg·s⁻¹·m⁻² to 400 kg·s⁻¹·m⁻². Verification is performed based on the surface heat transfer coefficient of internally grooved tube. A deviation of 8% with Zhang’s [6] experimental data are observed, and it is shown in Figure 3.

![Figure 3. Validation of numerical study with experimental data.](image)

3. Results and discussion

The water velocity is 5 m/s. The evaporation temperature of the refrigerant is 278 K. Four different corrugation heights are chosen to study the effect of corrugation height on heat transfer and pressure drop. The different corrugation heights of each case are presented in Table 1. To investigate the effect of refrigerant velocity on heat transfer and pressure drop, five different refrigerant velocities are selected. Different refrigerant velocities of each case are presented in Table 2.

| Case | Diameter (D, mm) | Height (H, mm) | Pitch (P, mm) | Width (W, mm) | H/D | T (H₂O) (°C) | Velocity (V, m/s) |
|------|-----------------|----------------|---------------|---------------|-----|--------------|------------------|
| 5    | 22              | -              | 25            | 4.4           | -   | 60           | 2.0              |
| 6    | 22              | -              | 25            | 4.4           | -   | 60           | 3.0              |
| 7    | 22              | -              | 25            | 4.4           | -   | 60           | 4.0              |
| 8    | 22              | -              | 25            | 4.4           | -   | 60           | 5.0              |

Table 2. Experimental design parameters for corrugated tubes with varying refrigerant velocities.
The temperature distribution of the corrugated pipe is presented in Figure 4. The red color corresponds to the regions of higher temperature and the blue color corresponds to the regions of lower temperature. The corrugation height has a significant effect on the temperature field of the corrugated pipe. Corrugated pipes of different corrugation heights have different temperature fields. The reason is that different corrugation heights have different enhanced heat transfer effects. The refrigerant velocity also has a significant effect on the temperature distribution of the corrugated tube. Different refrigerant velocities lead to different temperature fields in the corrugated tube. The reason is that the greater the refrigerant velocity leads to the stronger the heat transfer of the corrugated tube.

The distribution of R22 refrigerant vapor volume fraction are presented in Figure 5. The yellow color corresponds to the regions of higher refrigerant vapor volume fraction and the blue color corresponds to the regions of lower refrigerant vapor volume fraction. The corrugation height has a significant effect on refrigerant vapor volume fraction. Corrugated pipes of different corrugation heights have different refrigerant vapor volume fraction fields. The reason is that different corrugation heights have different enhanced heat transfer effects. The refrigerant velocity also has a significant effect on refrigerant vapor volume fraction distribution. Different refrigerant velocities lead to different refrigerant vapor volume fraction fields in the corrugated tube. The reason is that the greater the refrigerant velocity leads to the stronger the heat transfer of the corrugated tube.

![Figure 4. Fluid temperature distribution in the corrugated pipe.](image1)

![Figure 5. R22 vapor volume fraction distribution in the corrugated pipe.](image2)
3.1. Comparison analysis with smooth tube

The surface heat transfer coefficients of Case 1–Case 4 are presented in Figure 6. The surface heat transfer coefficient for all three corrugation heights of the corrugated pipe is higher than the smooth pipe. With the corrugated height rises, the surface heat transfer coefficient of the corrugated pipe rises and then falls. The corrugated tube with a corrugated height of 3.3mm has the highest surface heat transfer coefficient. The surface heat transfer coefficient of corrugated tube with a corrugated height of 3.3mm is 41.3% higher than that of the smooth tube, and 19.5% higher than that of the corrugated tube with a corrugated height of 4.4mm. The results is in coincidence with the results of Liu [7] and Córcoles [8].

The pressure drop of Case 1 to Case 4 are presented in Figure 6. The pressure drop of the corrugated pipe increases linearly with the increase of the corrugation height. The corrugated pipe with a corrugation height of 4.4mm has the maximum pressure drop, which is 1200% higher than that of the smooth pipe. The corrugated pipe with a corrugation height of 2.2 had the lowest pressure drop, which is 200% higher than that of the smooth pipe. The effect of corrugation height on pressure drop is very significant.

The highest surface heat transfer coefficient and moderate pressure drop is for the corrugated pipe with the corrugation height of 3.3 mm.

![Figure 6](image1.png)

**Figure 6.** Comparison of surface heat transfer coefficient and pressure drop for corrugated and smooth tubes.

3.2 Analysis of the effect of refrigerant velocity on the heat transfer characteristics of corrugated tubes

The trend of the surface heat transfer coefficient of the corrugated pipe with the refrigerant velocity is shown in Figure 7. the surface heat transfer coefficient increases with the increase of refrigerant velocity. At the refrigerant velocity of 5m/s, the corrugated pipe has the maximum surface heat transfer coefficient, which was 434% higher than that of corrugated pipe at the refrigerant velocity of 1m/s.

The trend of pressure drop in the corrugated pipe with the refrigerant velocity is shown in Figure 7. The pressure drop increases in a linear trend as the refrigerant velocity increases, which is similar to the results of Li [9] and Mao [10]. At a refrigerant velocity of 5m/s, the corrugated pipe had a maximum pressure drop, which is 1150% higher than pressure drop of the corrugated at the refrigerant velocity of 1m/s.

![Figure 7](image2.png)

**Figure 7.** The effect refrigerant velocity on surface heat transfer coefficient and pressure drop of corrugated pipe.
4. Conclusion
The flow boiling characteristics of internally corrugated tubes with different structural parameters are investigated based on CFD using R22 as refrigerant. The following results are obtained from the numerical study.

1. The surface heat transfer coefficient is significantly increased for corrugated tubes with different structural parameters compared to smooth tubes. The corrugated pipe with corrugation height of 3.3 mm (H/D = 0.15) has the highest heat transfer coefficient, while the pressure drop is lower than the corrugated pipe with a corrugation height of 4.4 mm. The corrugation height of 3.3 mm (H/D = 0.15) is the optimal construction parameter.

2. The refrigerant velocity has a significant effect on both the heat transfer coefficient and the pressure drop of the corrugated tube, which increase with the increase of the refrigerant velocity. The corrugated pipe structure and operating conditions are selected depending on the double pipe heat exchanger application requirements.

Author Contributions
Shenglin Zhu: Conceptualization, Formal analysis, Data curation, Writing - original draft, Review and Editing. Jinfeng Wang: Conceptualization, Funding acquisition, Supervision, Writing original draft, Review and Editing. Jing Xie: Conceptualization, Funding acquisition, Supervision, Writing original draft, Review and Editing. All authors have read and agreed to the published version of the manuscript.

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Conflicts of Interest
The authors declare no conflict of interest.

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