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# Research on the Key Technologies of Boiling Flow and Heat Transfer Simulation Based on Two Fluid Model

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***Abstract***

*Because of its high heat transfer efficiency, compact structure and many kinds of heat transfer characteristics, the plate fin heat exchanger has been widely used in the field of natural gas liquefaction. The interior of the plate fin heat exchanger, mainly composed of many small rectangular channels, and the heat transfer of hydrocarbon mixed refrigerant in the interior. For the research on the rectangular of hydrocarbon mixed refrigerant flow boiling and heat transfer in the minichannels of plate fin heat exchanger, explore the related mechanism of hydrocarbon mixed refrigerant boiling flow and heat transfer in the interior, which will assist in liquefied natural gas field of plate fin heat exchanger thermal performance design; at the same time, it is also driven by the China natural gas liquefaction systems for large thermal equipment, improve the energy saving of heat equipment or system is the key.*

***Keywords:*** *Boiling heat transfer; hydrocarbon mixed refrigerant; flow boiling; plate fin heat exchanger*

## 1. Introduction

Based on the two fluid model, simulating the small channel hydrocarbon mixed refrigerant flow boiling and heat transfer characteristics, obtained the temperature, velocity and other parameters are vapor and liquid phase parameters [1-2]. However, when calculating the boiling heat transfer coefficient, the average temperature of the fluid and the average temperature of the wall are needed, so how to calculate the average temperature and the average temperature of the wall is the key to calculate the boiling heat transfer coefficient [3]. At the same time, small channel hydrocarbon mixed refrigerant flow boiling and the accuracy of the simulation results of heat transfer, directly affected by the vapor liquid phase interface depth effect [4]. Therefore, vapor liquid phase interface depth is accurate simulation of flow boiling and the key parameters of the law of heat exchanger [5-6]. To accurately determine vapor-liquid phase interface depth is accurate simulation of flow boiling and change rule of heat and a key.

## 2. Study on the Calculation Method of the Average Temperature of Fluid

Based on the two fluid model of small channels hydrocarbon mixed refrigerant flow boiling and the simulation of heat exchanger, the simulation results of temperature are the temperature of the vapor and liquid phase; that is to say, the average temperature of the fluid is obtained by the simulation does not [7]. However, it is necessary to calculate the average temperature of the fluid when calculating the boiling heat transfer coefficient, so it is necessary to get the average temperature of the fluid through the theoretical deduction. Take one element $dxdydz$ in a small rectangular channel is shown in Figure 1, there are gas and liquid two-phase flow exist in the micro.

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Assuming that the vapor and liquid phase temperature $T_g$ and $T_l$ respectively, vapor-liquid phase average temperature $T_{ave}$, by the principle of conservation of energy available: the energy of vapor at temperature $T_g$ and the energy of the liquid phase at temperature $T_l$ is equal to the of the energy of vapor and liquid phase at temperature $T_{ave}$, that is:

$$Q_g + Q_l = Q_{g(ave)} + Q_{l(ave)} \tag{1}$$


**Figure 1. Physical Model of a Vertical Rectangular Channel**

In the formula

$Q_g$ ——When the temperature is $T_g$ in micro - vapor with the energy of J;

$Q_l$ ——When the temperature is $T_l$ in the micro liquid phase with the energy of J ;

$Q_{g(ave)}$、$Q_{l(ave)}$ ——When at the average temperature respectively in micro vapor and liquid phase with the energy, J。

When the micro volume tends to zero（$dxdydz \to 0$）, type (1) can be expressed as

$$\lim_{dxdydz \to 0} Q_g + \lim_{dxdydz \to 0} Q_l = \lim_{dxdydz \to 0} Q_{g(ave)} + \lim_{dxdydz \to 0} Q_{l(ave)} \tag{2}$$

The liquid phase saturation point is the reference point, that is, in this state, the enthalpy of the liquid phase is equal to zero, and the type (1) can be further expressed as

$$Q_g = r_g \rho_g \gamma dxdydz + r_g \rho_g C_g T_g dxdydz \tag{3}$$

$$Q_l = (1 - r_g) \rho_l C_l T_l dxdydz \tag{4}$$

$$Q_{g(ave)} = r_g \rho_g \gamma dxdydz + r_g \rho_g C_g T_{ave} dxdydz \tag{5}$$

$$Q_{l(ave)} = (1 - r_g) \rho_l C_l T_{ave} dxdydz \tag{6}$$

Bring the formula (3) - formula (6) into the formula (2), it can be concluded that

 

$$T_{ave} = \frac{r_g \rho_g C_g T_g + (1 - r_g) \rho_l C_l T_l}{r_g \rho_g C_g + (1 - r_g) \rho_l C_l} \tag{7}$$

Through the formula, we can conclude that, in the small channel hydrocarbon mixed refrigerant flow boiling and heat transfer process, a vapor liquid phase average temperature inside is related to not only vapor and liquid phase temperature, but also vapor and liquid phase density, constant pressure than heat and vapor phase volume fraction.

## 3. Study on Calculation Method of Wall Surface Average Temperature

The average temperature of the wall of the small rectangular channel is composed of two parts: one part is the wall temperature which is occupied by the vapor phase and the other is the wall temperature[8-9]. A micro dS in small rectangular channel wall, the micro surface by vapor and liquid occupied. Assuming that the vapor phase occupy wall element surface temperature $T_{w,g}$, liquid phase occupy wall element surface temperature $T_{w,l}$, infinitesimal surface average temperature $T_{ave}$, by the principle of conservation of energy available: vapor phase occupied the element surface temperature $T_{w,g}$ add to the infinitesimal surface materials have energy and liquid phase according to infinitesimal surface temperature for $T_{w,l}$ infinitesimal surface materials have energy is equal to the average element surface temperature $T_{ave}$ infinitesimal surface materials have the energy, *i.e.*

$$Q_{w,g} + Q_{w,l} = Q_{w,ave} \tag{8}$$

When the micro surface area tends to infinity, the formula (8) can be expressed as

$$\lim_{dS \to 0} Q_{w,g} + \lim_{dS \to 0} Q_{w,l} = \lim_{dS \to 0} Q_{w,ave} \tag{9}$$

Taking 0K as the reference point of the enthalpy value of the wall material, the formula (8) can be further expressed as

$$Q_{w,g} = r_g \rho_w C_w T_{w,g} dS \tag{10}$$

$$Q_{w,l} = (1 - r_g) \rho_w C_w T_{w,l} dS \tag{11}$$

After bring the formula (10) (11) into the formula (9) and simplification, can be obtained

$$T_{w,ave} = r_g T_{w,g} + (1 - r_g) T_{w,l} \tag{12}$$

Through the formula (12), we can conclude that, average wall surface temperature is related to wall gas phase volume fraction, steam phase occupy wall element surface temperature and liquid phase occupy wall element surface temperature.

## 4. Study on Determination Method of Interaction Depth of Vapor Liquid Interface

Due to the accuracy of hydrocarbon mixed refrigerant flow boiling and heat transfer simulation in small rectangular channels, influenced by interaction depth of vapor liquid

interface[10]. Therefore to determine reasonable vapor-liquid interface depth is the key to accurately simulate boiling flow and heat transfer of hydrocarbon mixed refrigerant in vertical rectangular minichannels [11-12]. Based on the principle of conservation of energy, the method of determining the interaction depth of the vapor liquid interface is theoretically deduced.

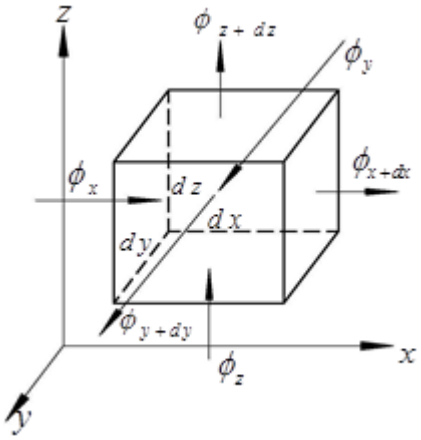


**Figure 2. Micro Control Element**

Assuming the rectangular Mini channels section size for $a \times b$, small rectangular channel length $l$ and the wall heat flux $q$, inlet mass flow rate $m$; in the internal flow field from a micro control volume $dV = dxdydz$ (as shown in Figure 2), the micro control three sides parallel to the X, y, and Z axis. From X, y, and Z into infinitesimal control $dV$ heat $\phi_x$, $\phi_y$ and $\phi_z$, respectively, from $x + dx$、$y + dy$ and $z + dz$ the surface, and outflow of infinitesimal heat control respectively, $\phi_{x+dx}$, $\phi_{y+dy}$ and $\phi_{z+dz}$. By the principle of energy conservation can be drawn into the micro control element is equal to the heat element of the heat flow, *i.e.*

$$\phi_x + \phi_y + \phi_z = \phi_{x+dx} + \phi_{y+dy} + \phi_{z+dz} \tag{13}$$

In the formula, along the X direction of inflow and outflow of fluid control body has two main parts, vapor and liquid phases, therefore, the formula (13) along the X direction the inflow, outflow of the control body heat can be further expressed as

$$\phi_{g,x} = \rho_g u_g H_g dydz - \lambda_g \frac{\partial T_g}{\partial x} dydz \tag{14}$$

$$\begin{aligned} \phi_{g,x+dx} = & \left( \rho_g u_g H_g + \frac{\partial}{\partial x}(\rho_g u_g H_g)dx \right) dydz \\ & - \left( \lambda_g \frac{\partial T_g}{\partial x} + \frac{\partial}{\partial x}\left( \lambda_g \frac{\partial T_g}{\partial x} \right) dx \right) dydz \end{aligned} \tag{15}$$

In the formula: $u_g$ - vapor phase in the X direction of the speed, m/s.

In the same way, for the liquid phase, long the X direction the inflow, outflow of the control body heat can be further expressed as

 

$$\phi_{l,x} = \rho_l u_l H_l dydz - \lambda_l \frac{\partial T_l}{\partial x} dydz \tag{16}$$

$$\begin{aligned} \phi_{l,x+dx} = & \left( \rho_l u_l H_l + \frac{\partial}{\partial x}(\rho_l u_l H_l)dx \right) dydz \\ & - \left( \lambda_l \frac{\partial T_l}{\partial x} + \frac{\partial}{\partial x}\left( \lambda_l \frac{\partial T_l}{\partial x} \right) dx \right) dydz \end{aligned} \tag{17}$$

In the formula: $u_l$ - liquid phase in the X direction of the speed, m/s.

After simplification, it can be concluded that

$$\phi_x = \left( \rho_g u_g H_g + \rho_l u_l H_l \right) dydz - \left( \lambda_g \frac{\partial T_g}{\partial x} + \lambda_l \frac{\partial T_l}{\partial x} \right) dydz \tag{18}$$

$$\begin{aligned} \phi_{x+dx} = & \left( \rho_g u_g H_g + \rho_l u_l H_l + \frac{\partial}{\partial x}(\rho_g u_g H_g)dx + \frac{\partial}{\partial x}(\rho_l u_l H_l)dx \right) dydz \\ & - \left( \lambda_g \frac{\partial T_g}{\partial x} + \lambda_l \frac{\partial T_l}{\partial x} + \frac{\partial}{\partial x}\left( \lambda_g \frac{\partial T_g}{\partial x} \right) dx + \frac{\partial}{\partial x}\left( \lambda_l \frac{\partial T_l}{\partial x} \right) dx \right) dydz \end{aligned} \tag{19}$$

Similarly, the flow of heat from the Y and Z direction, and the flow of the control body can be further expressed as the X direction.

In the small rectangular channel wall, the following relationships will be satisfied.

$$\iint_{\Sigma_{wall}} \rho_g H_g u_g dydz + \rho_g H_g v_g dxdz + \rho_g H_g w_g dxdy = 0 \tag{20}$$

$$\iint_{\Sigma_{wall}} \rho_l H_l u_l dydz + \rho_l H_l v_l dxdz + \rho_l H_l w_l dxdy = 0 \tag{21}$$

In the inlet and outlet of the rectangular channel, the velocity of the flow direction (z direction) is much larger than that of the perpendicular to the flow direction (x, Y axis), so the surface of the inlet and outlet will satisfy the following relationship.

$$u_g = v_g = u_l = v_l = 0 \tag{22}$$

Therefore, on the surface of the fluid corresponding to the inlet and outlet of the small rectangular channel, by simplifying the principle of energy conservation, we can get the following relationship:

$$\overline{d}_{gl} = \sqrt{\frac{\overline{r}_g \overline{r}_l \overline{\lambda}_{gl} \overline{N} u_{gl} \left( \overline{T}_g - \overline{T}_l \right) ab}{2(a+b)q}} \tag{23}$$

In the process of derivation, it is assumed that the fluid with a micro control body increased heat will be completely transformed into the latent heat required for liquid vapor, but in fact the fluid with a micro control body increased heat in part into vapor sensible heat, so that the steam superheating. Therefore, in this definition, a parameter -- vaporization rate, which is equal to the ratio of the heat absorbed by the liquid vapor in a vertical rectangular channel, is the ratio of the total amount of heat absorbed from the wall surface.

$$\varepsilon = \frac{(a+b)lq - abmx_{out}C_{pg}\left(\overline{T_g} - \overline{T_l}\right)}{(a+b)lq} \tag{24}$$

Bring the formula (40) into the formula (39) can be obtained

$$\overline{d}_{gl} = \sqrt{\frac{\overline{r_g}\,\overline{r_l}\,\overline{\lambda}_{gl}\,\overline{Nu}_{gl}\,l(1-\varepsilon)}{2mx_{out}C_{pg}}} \tag{25}$$

It can be drawn by the formula (25), that vapor liquid phase interface average interaction depth is related to material parameters (such as vapor phase constant pressure specific heat, vapor liquid phase mixing coefficient of thermal conductivity), the vapor-liquid phase Nusselt number, vaporization efficiency and export dry and steam phase volume fraction of liquid phase on, so through (25) under different simulation conditions can be determined when the liquid vapor interface depth.

## 5. Validation of the Method for Determining the Interaction Depth of Vapor Liquid Interface

When the inlet mass flow rate is 215kg/($m^2$·s), the dry degree is 0.5, and the wall heat flux is different, the influence of the interaction depth of the vapor liquid interface on the boiling heat transfer coefficient is simulated and analyzed. Figure 3 is the simulation results. The results show that when at different depth of gas liquid interface, the simulation results of boiling heat transfer coefficient are different; the interaction depth of the vapor liquid phase has a great influence on the accuracy of the simulation results of boiling heat transfer coefficient, how to determine the interaction depth of the vapor liquid phase is the key to accurately simulate the boiling heat transfer.

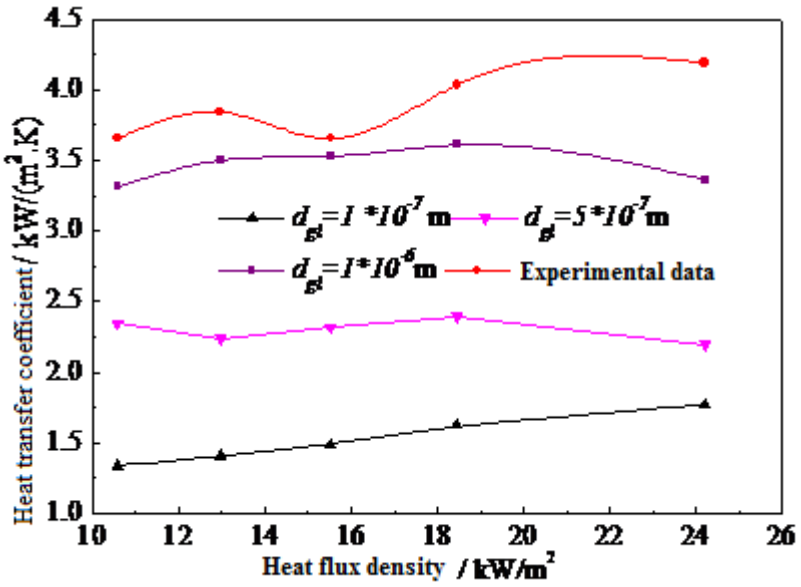


**Figure 3. Comparison of the Simulation Results with the Experimental Results of R21 at the Interface of Different Vapor Liquid Interface**

The theoretical and simulation results show that vapor liquid phase interface depth have a great impact on the accuracy of boiling flow and convective thermal simulation, therefore, it is particularly necessary to verify the rationality of the interaction depth of the vapor liquid phase which is determined by the formula (25).

According to the experimental results of boiling flow and heat transfer in Table 1, and based on the formula (25) to calculate the interaction depth of the vapor liquid interface. Based on the calculation of the interaction depth of the vapor liquid phase, to simulate the experimental conditions, so as to verify the rationality of the method of determining the interaction depth of the vapor liquid interface.

Figure 4 is under these circumstances that inlet mass flow rate for 215kg/($m^2$·s), wall heat flux for 6kW/$m^2$, inlet mass flow rate for 50kg/($m^2$·s), wall heat flux density is 1.1kW/$m^2$ and dry degree of R21 in mini channel flow boiling and heat exchange effect of simulation result and its comparison with experimental results. The comparison results show that both large flow and small flow condition, the boiling heat transfer coefficient of simulation results and the experimental results are in good agreement; Figure. 5 is the result of the error analysis between the simulation results and the experimental data in Figure 3, and the error analysis results show that between the results of the simulation and the experiment error is within + / - 15%.

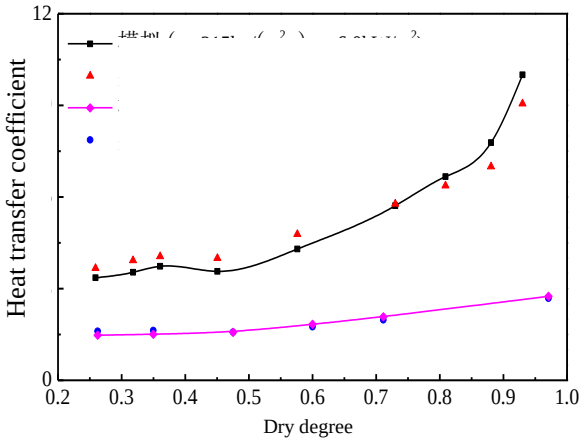


**Figure 4. Comparison of Simulation Results and Experimental Results of R21 in Different Dry Degree**

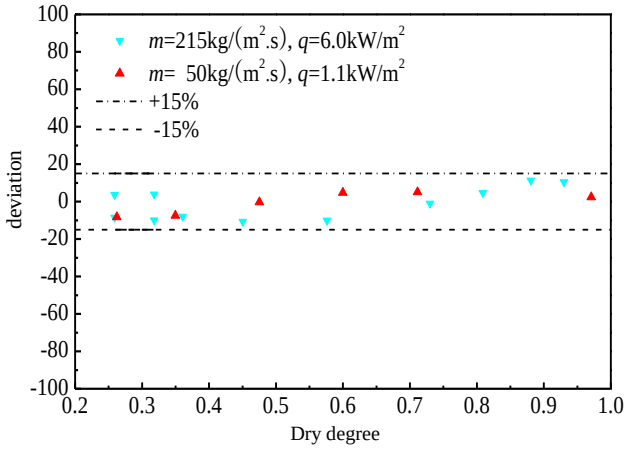


**Figure 5. Error Between the Simulation Results and the Experimental Results at Different R21**

**Table 1. Experimental Results of Heat Transfer Coefficient of R21**

| Inlet mass flow rate / kg/(m$^2$.s) | Wall heat flux density / kW/m$^2$ | Dry degree | Experimental value of literature / kW/(m$^2$.K) |
|---|---|---|---|
| 215 | 6.0 | 0.259 | 3.668 |
| | | 0.318 | 3.923 |
| | | 0.361 | 4.051 |
| | | 0.450 | 3.992 |
| | | 0.576 | 4.779 |
| | | 0.730 | 5.772 |
| | | 0.809 | 6.372 |
| | | 0.881 | 6.991 |
| | | 0.930 | 9.056 |
| 50 | 1.1 | 0.262 | 1.607 |
| | | 0.349 | 1.630 |
| | | 0.475 | 1.576 |
| | | 0.600 | 1.750 |
| | | 0.711 | 1.974 |
| | | 0.971 | 2.686 |
| 215 | 5.662 | 0.5 | 4.787 |
| | 14.612 | | 5.890 |
| | 21.370 | | 5.701 |
| | 28.858 | | 5.984 |
| | 37.260 | | 6.803 |
| 215 | 10.594 | 0.25 | 3.654 |
| | 12.968 | | 3.843 |
| | 15.525 | | 3.654 |
| | 18.448 | | 4.032 |
| | 24.201 | | 4.189 |

The refrigerant in the mini channel flow boiling and heat transfer simulation results show that, it has very high accuracy and applicability to apply the theoretical derivation of the vapor liquid phase interface depth in small channel boiling heat transfer simulation.

## 6. Concluding Remarks

Based on the principle of conservation of energy, Theoretical study on the key technologies of numerical simulation of boiling flow and heat transfer in a vertical rectangular channel, has obtained calculation method of the average temperature of the fluid, small rectangular channel wall mean temperature and vapor-liquid phase interface depth, and validated the vapor-liquid phase interface depth determination method of rationality through the experimental data of this paper draws. Its application in hydrocarbon mixed refrigerant in the mini channel flow boiling and heat transfer simulation is feasible.

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