Theoretical and Experimental Research on CO2 Electrical Heating Pool Boiling Heat Transfer Outside a Horizontal Tube

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Theoretical and Experimental Research on CO_2 Electrical Heating Pool Boiling Heat Transfer Outside a Horizontal Tube

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ABSTRACT

Numerical simulation on electrical heating pool boiling heat transfer with CO_2 as refrigerant outside a horizontal tube is carried. A voltage-controlled heating method has been used in the experiment, with the advantages of good stability and adjustability of the experimental heat flux density. After a series of preliminary calculation and pre-work, numerical simulation is carried based on a software FLUENT. Bubble behaviors are observed, the distribution regularity of volume fraction of vapor is obtained and compared with the experimental results. The results show that numerical simulation and experimental results are in good agreement. Furthermore, by changing the heat flux density, the comparison of velocity on center location of experimental tube is analyzed. Varying pattern is satisfying. Evidently, for velocity, the simulation values are relatively higher and the data locate in the range of 1.40~1.52 times higher than the experimental data. This paper makes useful exploration of CO_2 pool boiling heat transfer and the design of evaporator.

1. INTRODUCTION

With the relatively excellent physical properties, such as high evaporation pressure, great specific heat and large heat exchange capacity, CO_2 is widely applied in heat exchangers with small diameter pipes. However, when boiling in evaporators, drying up happens easily that it leads to the deterioration of heat transfer (Hihara E. and Tanaka S., 2000). Meanwhile, a small amount of lubricating oil in the refrigeration cycle would have negative impact on heat transfer (Yun R., 2002). Katsuta M. (2004) and Ding (2009) came to the same conclusion through their researches in recent years. However, Ribatski S. and Jabardo (2009) observed these phenomena would not occur in flooded evaporators, because the heat transfer surfaces are soaked in refrigerant. Moreover, CO_2 is prone to boil at the same evaporation temperature for weak tension on surface. Ye (2007) and Liu *et al*. (2005) proposed that CO_2 boiling heat transfer coefficient is higher. Therefore it has been widely used in daily life and various technical fields of engineering. Systematic studies on this phenomenon have been carrying for over a hundred years, and great achievements have been obtained. However, due to the boiling phenomenon involve a very wide range of time and space scale and complicated mutual-coupling, current relevant knowledge is mainly obtained from empirical correlations of the experiment data, with strong empirical knowledge, which makes it difficult to meet the increasing demand of the application fields of science and technology, like aerospace, micro-electro-mechanism.

The electrical heating pool boiling technology studied in this paper obtains a wide application in heat transfer devices in power, energy *etc*, because of its obvious advantages of compact structure and superiority on heat transfer performance. In this paper, numerical simulation on electrical heating pool boiling heat transfer with CO_2 as refrigerant outside a horizontal tube is carried based on a software FLUENT, and the boiling heat transfer performance outside the tube is studied. Compared to experiments, numerical simulation has its unique advantages, taking cost and time into consideration. With the aid of mathematical model and FLUENT, numerical simulation of CO_2 boiling outside tube is carried. For some parameters, such as velocity, pressure and steam quality, visualization is easy to implement. With essential knowledge of saturated boiling heat, mass transfer and mechanism, it guides the design and experiment to save manpower and material resources and shorten the development cycle.

2. PHYSICAL AND MATHEMATICAL MODELS

2.1 Physical Model

An electricity-controlled heating method is adopted. Changing the heat power of the electrical heating tube is realized by changing the voltage of tube ends, thereby changing the heat flux density. By proposing electrical heating method, a good stability and adjustability of the experimental heat flux density is kept. One K type thermocouple is installed between the electrical heating rod and experimental pipe, then tight connection between electrical heating rod and the test tube is realized by using cold-drawn technology. The temperature measured by the thermocouple is the tube's wall temperature. The outer diameter and effective length of electrical heating tube are 22 mm and 1200 mm. Stainless steel pipe (φ89×10mm) is used as the outer tube in the experimental section. The method applied by Li (2007) is also used to collect the temperature of saturated CO_2. Refrigerant CO_2 enters the experimental section, then steam expel from the top. In order to observe the pool boiling heat transfer characteristics, a couple of light glasses are installed in the middle of the experimental section. For the convenience of the replacement of test tube, flange connections are used on both sides. Structure of test section of CO_2 electrical heating pool boiling heat transfer is shown in Figure1.

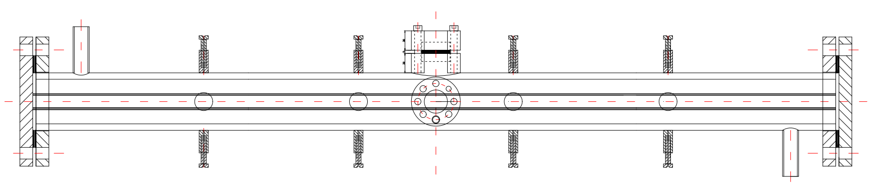


Figure 1: Structure of test section

First of all, a three dimensional model is conducted using 3D modeling software Pro/E. After a series of preliminary calculation, the self-adaptive grid is conducted based on the temperature field. According to Gianluca (2000), the grid of the inner wall is refined. Boundary conditions: the entrance is the velocity inlet, the exit is pressure outlet, the inner wall is considered as constant heat flux density, and the outside wall is heat adiabatic. The vertical and horizontal models of the test section are shown in Figure 2 and Figure 3.

Figure 2: Horizontal model of test section

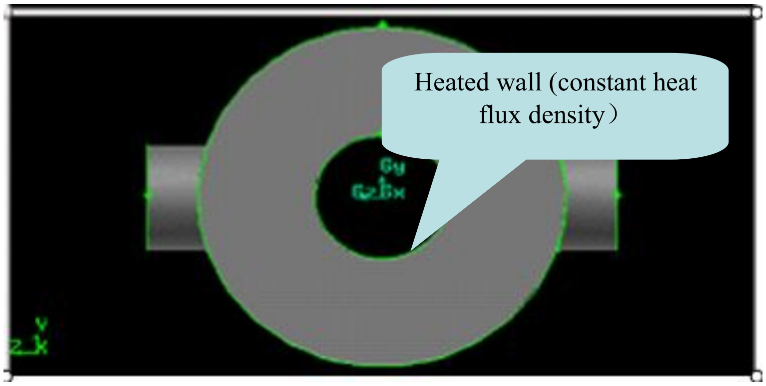


Figure 3: Vertical model of test section

2.2 Mathematical Model

In this paper, considering the standpoint of Tao (2000) about boiling heat transfer flow and heat transfer conditions inside the tube, some hypothesis on fluid flow through the tube are carried as follow:
(1) The flow inside the tube is steady and incompressible;
(2) Phase transition is conducted when fluid's temperature reaches certain value;
(3) The wall is at non-slip boundary condition;
(4) Take the influence of gravity into account;
(5) Ignore the heating effect produced by viscous dissipation when fluid flows.

Specialized model of boiling heat transfer is not available in FLUENT, taking Han's contribution (2009) as reference, it is realized by User-Defined Functions (UDF).The Mixture model in FLUENT is used in this paper. Mixture model is a simplified model of multiphase-flow. It solves the mass, momentum and energy equation, the volume fraction equation of the second phase, and simulation of the volume fraction of two-phase flow.

Continuity equation in Mixture model :

$$\frac{\partial}{\partial \mathrm{t}}(\rho m)+\nabla\cdot(\rho_m\overline{v}_m)=\dot{m} \tag{1}$$

In equation (1), $\overline{v}_m$ is defined as average velocity:

$$\overline{v}_m=\frac{\sum_{k-1}^{n}\alpha_k\rho_k\overline{v}_k}{\rho_m} \tag{2}$$

ρ_m is mixture density:

$$\rho_m=\sum_{k-1}^{k}\alpha_k\rho_k \tag{3}$$

α_k is the volume fraction of phase k and $\dot{m}$ describes the mass transfer of user-defined mass.

Momentum equation in Mixture model can be obtained to sum all momentum equations of all phases:

$$\frac{\partial}{\partial \mathrm{t}}(\rho_m\overline{v}_m)+\nabla\cdot(\rho_m\overline{v}_m\overline{v}_m)=-\nabla p+\nabla\cdot[\mu_m(\nabla\overline{v}_m+\nabla\overline{v}_m{}^T)]+\rho_m\overline{g}+\overline{F}+\nabla\cdot(\sum_{k-1}^{n}\alpha_k\rho_k\overline{v}_{dr,k}\overline{v}_{dr,k}) \tag{4}$$

In equation (4), n is the phase number, $\overline{\mathrm{F}}$ is volume force, μ_m is the viscosity in Mixture model:

$$\mu_m = \sum_{k=1}^{n} \alpha_k \mu_k \tag{5}$$

$\overline{v}_{dr,k}$ is drift velocity of phase k :

$$\overline{v}_{dr,k} = \overline{v}_k - \overline{v}_m \tag{6}$$

Energy equation in Mixture model:

$$\frac{\partial}{\partial t}\sum_{k=1}^{n}(\alpha_k \rho_k E_k) + \nabla \cdot \sum_{k=1}^{n}(\alpha_k \overline{v}_k (v_k E_k + p)) = \nabla \cdot (k_{eff} \nabla T) + S_E \tag{7}$$

In equation (7), k_{eff} is the effective thermal conductivity ,the first item on the right represents the energy transfer caused by the thermal conduction.

$$\mathrm{E}_k = h_k - \frac{p}{\rho_k} + \frac{v_k^{\ 2}}{2} \tag{8}$$

To the compressible phase:

$$\mathrm{E}_k = h_k \tag{9}$$

To the incompressible phase, it is the sensible heat of phase k .

3 RESULTS AND ANALYSIS

3.1 Numerical Method

(1)The entrance condition is velocity inlet boundary conditions.
(2)The exit condition is pressure outlet boundary, the backflow of upper export gas is zero.
(3)The wall boundary is constant heat flux density.
(4)According to the evaporation pressure, corresponding phase-transition temperature in UDF is set.

3.2 Simulation Results Analysis

Flow boundary conditions: inlet velocity v=5m/s, evaporation pressure P set as 2.6, 3.2, and 3.6MPa.The inner wall heat flux density is 10548, 20675, 34177,51054 kW/m^2, the out wall is considered adiabatic. Thermal properties of CO_2 reference to Xue *et al*. (2004).

The contours of volume fraction under different pressure and heat flux density are shown in Figure 4. The wall temperature rises rapidly along the flow direction due to the heating effect. When it reaches the superheat of boiling, bubbles appear. The wall temperature increases rapidly, contributing to the growing of active steam pocket rate and the bubbles, which results in the rapidly increasing of steam quality. Generated gas will gather in the upper portion of the annular space due to the gravity effect (Xiong *et al*., 2005). Under the same pressure, the gas volume rate increases with the increasing of heat flux density. This is mainly because when heat flux density increases, it contributes to higher superheat of the wall and more bubbles, which intensifying the bubbles disturbance and attenuating the flow boundary layer and heat boundary, therefore the boiling heat transfer is enhanced.

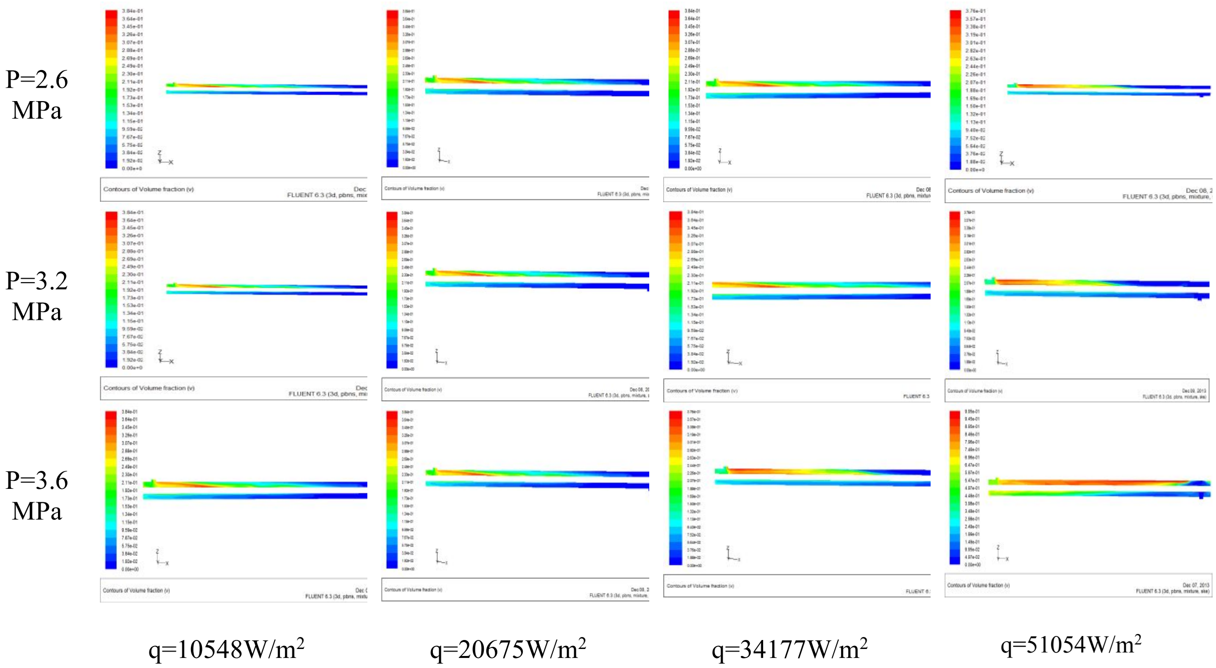


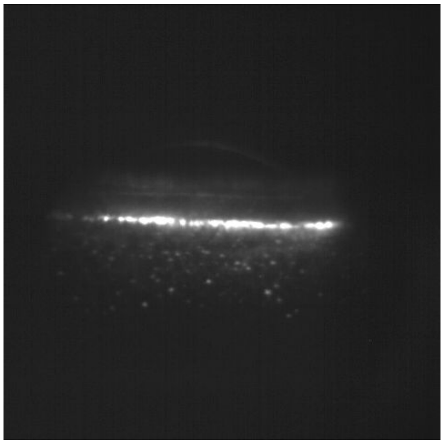
Figure 4: Contours of volume fraction under different pressure and heat flux

3.3 Comparison of Velocity on Center Location of Experimental Tube between Simulation Results and Experimental Results

Bubbles' moving direction and velocity characteristics during the process of pool boiling heat transfer are recorded by a high-speed camera (a maximum resolution of 1280*1024 pixels, a maximum speed of 100000 FPS).

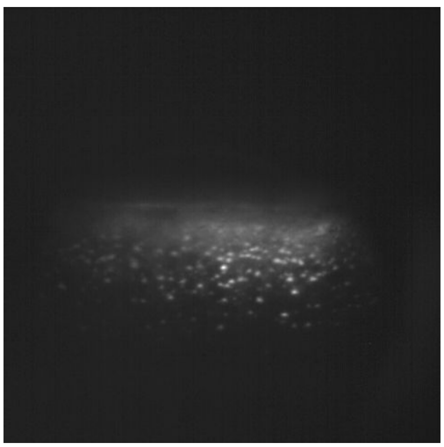
CO_2 pool boiling images took by digital camera at 3.6 MPa under different heat flux density are shown in Figure 5. By analyzing photographs, it is found that the distribution of bubbles on the surface of electrical heating tube is uneven, namely, the gasification core position in the process of boiling is not stable. For the following reasons, gasification cores are divided into stable ones and unstable ones, the stable ones can generate bubbles periodically, and the unstable ones have bubbles occasionally. The existence of two different gasification cores makes the distribution uneven, and different heat flux density and velocity aggravate the inhomogeneity of distributing. According to the images, when heat flux density is low, the quantity of gasification cores and bubbles are few and increases along with heat flux density. With software Insight-3g, the camera images are effectively analyzed. By setting the unit length time, instantaneous velocity is calculated when contrasting the variation of displacement in two consecutive photos. The comparison of velocity on center location of pool boiling experimental tube between simulation results and experimental results under different heat flux density is shown in Figure 6. The velocity increases with the increasing of heat flux density. When heat flux density increases, disturbance enhances, then as to the expel velocity of bubbles.

The trend of simulation results and experimental results is also shown in Figure 6. The agreement is much satisfying. Evidently, the simulation values are relatively higher and the data locate in the range of 1.40~1.52 times higher than experimental data. In the process of the simulation, the tube is completely smooth and horizontal. However, during experiment, levelness and operating environment cannot be totally realized, which results in the decreasing of steam velocity and steam cannot be expelled in time, leading to the low experimental data.



$q=2.6kW/m^2$

$q=6.75kW/m^2$



$q=15.2kW/m^2$

$q=23.73kW/m^2$

Figure 5 : CO_2 pool boiling images at 3.6 MPa under different heat flux density

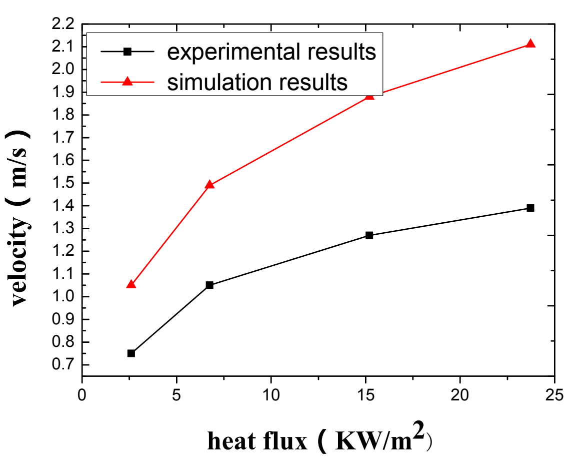


Figure 6: Comparison between simulation results and experimental results

4. CONCLUSIONS

It is studied that a numerical simulation and analysis on evaporation heat transfer outside a horizontal tube with the Mixture model in FLUENT software, and UDF programming. It is concluded as follow:

- During the boiling process of refrigerant CO_2 outside the horizontal tube, the quantity of bubbles increases along the flow direction. Bubbles appear on the wall and slowly flee until they devote into the main flow. Most of the bubbles gather in the upper space of the annular space due to gravity effect;
- Under the same condition of heat flux density, the gas volume rate outside the horizontal tube increases with the evaporation pressure. When pressure is fixed, it increases with heat flux density;
- The comparison of velocity on center location of pool boiling experimental tube between simulation results and experimental results under different heat flux density is analyzed. Integrally, simulation results of pool boiling heat transfer match the experimental results perfectly, which verified the reliability of this paper's model.

NOMENCLATURE

$\overline{v}_m$	average velocity	(m/s)
ρ_m	mixture density	(Kg/m$^{3)}$
k	phase	(-)
α_k	volume fraction of phase k	(-)
$\dot{m}$	mass transfer of user-defined mass	(kg)
$\overline{\mathrm{F}}$	volume force	(N)
μ_m	viscocity in Mixture model	(Pa·s)
$\overline{v}_{dr,k}$	drift velocity of phase k	(m/s)
k_{eff}	effective thermal conductivity	(W/(m^2·k))
P	evaporation pressure	(MPa)
q	heat flux density	(kW/m$^{2)}$

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