EXPERIMENTAL STUDY ON BOILING FLOW AND HEAT TRANSFER IN PEBBLE BED CHANNELS

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Boiling flow and heat transfer in porous media composed of spherical fuel elements have a significant influence on a reactor’s efficiency and safety. In the present study, an experimental setup is designed and the boiling flow and heat transfer in porous media composed of regularly distributed spheres are investigated. The test sections mainly consisted of a polycarbonate plate for observation, an aluminum plate with densely distributed hemispheres, and a layer of glass spheres. The water flowing in the porous channels was heated by electrical power through the aluminum plate with hemispheres. Four types of spheres with diameters of 5, 6, 7, and 8 mm were involved. In the experimental parameter range, four flow regimes (bubbly, bubbly slug, slug, and slug annular) were observed to consider the effects of heat flux, mass flux, and particle diameter. The boiling heat transfer results showed that larger particles lead to a lower heat transfer coefficient and result in a higher wall superheat of the original nucleation boiling. Finally, a correlation between the boiling heat transfer coefficient and the measured data is proposed.

KEY WORDS: boiling, pebble bed, heat transfer coefficient, flow regime, porous channel

1. INTRODUCTION

The spherical fuel element nuclear reactor shows great promise because of its advantages in safety, fuel utilization, small volume, and economy (Kadak, 2005; Tsiklauri et al., 2005; Sefidvash, 2004). In the core of the reactor, there is porous media composed of small spherical fuel elements, which serve as the heating sources of the coolant. The boiling heat transfer in the porous media exerts significant influences on the reactor. However, research on the reactor’s thermal-hydraulic characteristics is still in its infancy and a deeper understanding of their impact on heat pipes is needed. Some different or even conflicting conclusions have been drawn because of the complex and special particle compositions of porous structures and the difficulty in conducting experiments on them (Yang et al., 2006).

According to the research of Jiang et al. (1999, 2004) and Jamialahmadi et al. (2005), the heat transfer increases with an increase in the particle diameter; however, this conclusion is contrary to that obtained in Jeigurnik et al. (1991) and Hwang and Chao (1994). There are few published experimental research studies on the heat transfer of porous media, which serve as the heating source. Such studies are of great significance, because in pebble bed reactors the spherical fuel elements serve as the heating sources. Naik and Dhir (1982) obtained the temperature distribution along the direction of flow for water flowing through layers of inductively heated steel particles; however, in their experiment the heating by the particles was non-uniform because of the skin effect (Schäfer and Lohnert, 2006; Atkhen and Berthoud, 2006). Therefore, uniform heating of a fluid by particles was one of the key issues in the experiment. Meanwhile, most research studies are qualitative (Liao and Zhao, 2000; Chen et al., 2000; Cao et al., 2004), while there are very few quantitative research studies.

In this study, an experimental investigation was conducted on the boiling flow and heat transfer characteristics in porous media composed of heating particles. In order to observe the flow patterns between the heating surface and the spheres, transparent glass spheres were employed to overcome the shortcomings induced by metal spheres, which
were always employed in previous studies. The effects of particle diameter, heat flux, and mass flux on the boiling flow regime and the heat transfer were investigated. A correlation between the heat transfer coefficient and the measured experimental data is proposed.

2. EXPERIMENTAL TECHNIQUES

2.1 Experimental Apparatus

The apparatus used to measure the heat transfer in channels packed with particles is shown in Fig. 1. Its major parts included a test section, temperature controller, preheater, flowmeter, liquid vessel, and condenser. The deionized water pumped from the liquid vessel and flowing through the filter and the flowmeter was heated to a desired temperature in the preheater by an electrically resistant heater, which was controlled by the temperature controller. The fluid flows through the test section and condenser, and finally back into the liquid vessel. A high-speed imaging system was employed to visualize the boiling process in the test section.

The test section, as shown in Fig. 2, was designed for the purpose of both heating the particles and visual observation of the two-phase flow pattern. It consisted of a polycarbonate plate for observation on one side, and a regular aluminum structure with densely distributed hemispheres on the other side, which was the heating source of the fluid (see Fig. 3). The hemispheres on the aluminum plate were carved by a numerically controlled machine. Therefore, the hemispheres and the aluminum plate were integrated and no contact thermal resistance existed between them. There was a layer of glass particles between the polycarbonate and aluminum plates, which improved the flow uniformity.
perpendicular to the flow direction. The diameters of the hemispheres on the regular aluminum structure and those of the glass particles were the same. An electrical resistant heater was used to heat the regular aluminum structure, mica was used for insulation, asbestos was used to prevent heat loss in the test section, and a metal plate was used to fasten the structure. Four different types of packing media composed of particles were employed, as shown in Table 1. The experimental parameters including particle diameter \(d_p\), fluid inlet temperature \(T_{in}\), outlet vapor quality \(x\), Reynolds number \(Re\), and heat flux \(q\) are given in Table 2. The particle diameter and the inlet superficial velocity of the liquid
FIG. 3: A regular aluminum structure

TABLE 1: Specifications of the test section

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Case Number</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>1</td>
</tr>
<tr>
<td>Particle diameter $d_p$ (mm)</td>
<td>5</td>
</tr>
<tr>
<td>Length (mm)</td>
<td>260.48</td>
</tr>
<tr>
<td>Width (mm)</td>
<td>45</td>
</tr>
<tr>
<td>Height (mm)</td>
<td>6.58</td>
</tr>
<tr>
<td>Porosity</td>
<td>0.3667</td>
</tr>
</tbody>
</table>

TABLE 2: Experimental parameters

<table>
<thead>
<tr>
<th>$d_p$ (mm)</th>
<th>$T_{in}$ (°C)</th>
<th>Outlet Vapor Quality, $x$</th>
<th>Re minimum</th>
<th>Re maximum</th>
<th>$q$ (kW·m$^{-2}$) minimum</th>
<th>$q$ (kW·m$^{-2}$) maximum</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>80</td>
<td>0.16</td>
<td>466</td>
<td>4897</td>
<td>2.6</td>
<td>287.1</td>
</tr>
<tr>
<td>6</td>
<td>80</td>
<td>0.21</td>
<td>289</td>
<td>4862</td>
<td>0.8</td>
<td>329.8</td>
</tr>
<tr>
<td>7</td>
<td>80</td>
<td>0.16</td>
<td>330</td>
<td>3987</td>
<td>0.2</td>
<td>164.0</td>
</tr>
<tr>
<td>8</td>
<td>80</td>
<td>0.10</td>
<td>464</td>
<td>4744</td>
<td>0.2</td>
<td>123.4</td>
</tr>
</tbody>
</table>

phase, $u_{in}$, was used to calculate the Re number. The average heat flux, $q$, was obtained based on the real heating surface area adjacent to the fluids.

2.2 Uncertainty Analysis and Data Reduction

Six K-type thermocouples with ±0.1°C uncertainty were used to measure the wall temperature, as shown in Fig. 4. Thermocouples 1 and 2 were used to measure the pore temperature and thermocouples 3–6 were used to measure the hemisphere temperature. The six thermocouples were mounted on the aluminum plate with a depth of 2 mm from the heating surface. The measurement region was in the central area of the test section along the main flow direction. Two T-type thermocouples were employed to measure the inlet and outlet fluid temperatures with ±0.1°C uncertainty. The water flow rate was measured by an orifice mass flux meter with 0.5% uncertainty, the outlet pressure of the test
section was measured by a static pressure transmitter with \( \pm 0.1\% \) uncertainty, and the pressure drop was measured by the transmitter (Rosemount, America) with \( \pm 0.5\% \) uncertainty. An alternating current voltage transmitter and a current transmitter were employed to measure the heating voltage and current of the test section, respectively, with \( \pm 0.2\% \) uncertainty.

The fluid pressure drop in the single-phase flow region in the test section was calculated using our previously proposed correlation (Bai et al., 2011). The fluid specific enthalpy was obtained based on the uniform heating condition. The position of the saturated boiling in the test section was determined based on the pressure and specific enthalpy. Then, the pressure in the two-phase flow region was calculated by assuming a linear distribution. The fluid temperature was calculated based on the pressure and specific enthalpy in the single-phase flow region and was equal to the saturation temperature in the boiling flow region. The fluid specific enthalpy, \( h \), at any cross section is determined by both the inlet specific enthalpy and effective heating power, and the vapor quality, \( x \), in saturated boiling is determined by

\[
x = \frac{(h - h_l)}{h_{lg}}
\]

where \( h_l \) is the enthalpy of the saturated liquid; and \( h_{lg} \) is the latent heat of evaporation. In sub-cooled boiling, the vapor quality is determined by the Bowring (1962) correlation:

\[
x = e_l \frac{(h - h_D)}{h_{lg}}
\]

where \( e_l \) is the fraction of the heat used for liquid evaporation; and \( h_D \) is the fluid enthalpy at the departure point of the bubbles. Here, \( e_l \) is obtained as follows (Bowring, 1962):

\[
e_l = \left(1 + 3.2 \frac{p \rho_l c_{p,l}}{\rho_g h_{lg}}\right)^{-1}, \quad p = 0.1 - 0.95 \text{ MPa}
\]

Su (2009) measured the local void fraction of air/water two-phase flow in porous media with a conductivity probe sensor and proposed that the void fraction, \( v_f \), can be calculated by the drift flux model

\[
v_f = \frac{J_g}{C_0 J + v_{gj}}
\]

where \( J \) is the superficial velocity of two-phase fluids; and \( J_g \) is the gas superficial velocity. The coefficients \( C_0 \) and \( v_{gj} \) are determined by the empirical correlation of Coddington and Macian (2002).

The average temperature of the heating surface is calculated based on the measured values of the six thermocouples. First, the local heating surface temperatures, \( T_{wi} \) (\( i = 1–6 \)), are determined by the measured temperature \( T_i \) (\( i = 1–6 \)) through the heat conduction equation.
\[ q = K \frac{(T_i - T_{wi})}{\Delta x_i} \]  
(5)

where \( K \) is the thermal conductivity of the aluminum structure; and \( \Delta x_i \) (i = 1–6) is the distance between the thermocouple and the heating surface, which is equal to 2 mm in the present experiment. It is assumed that each local temperature, \( T_{wi} \), characterizes the corresponding heating surface elements, \( S_i \), respectively, as shown in Fig. 5. Therefore, the following relationship is obtained according to the conservation law of energy:

\[ \alpha (T_w - T_i) \sum_{i=1}^{6} S_i = \sum_{i=1}^{6} \alpha_i (T_{wi} - T_i) S_i \]  
(6)

It is supposed that the heat transfer coefficients are equal, and thus the average heating surface temperature is calculated by

\[ T_w = T_i + \frac{\sum_{i=1}^{6} (T_{wi} - T_i) S_i}{\sum_{i=1}^{6} S_i} \]  
(7)

Finally, the heat transfer coefficient is determined as follows:

\[ \alpha = \frac{q}{T_w - T_i} \]  
(8)

3. BOILING FLOW BEHAVIOR VISUALIZATION

3.1 Flow Regime Classification

Based on the flow characteristics of the liquid and vapor phases, the boiling flow regimes in the porous channels generally fall into four groups within the experimental parameter range: bubbly, bubbly slug, slug, and slug-annular flow. These flow regimes are generally consistent with those observed in air–water two-phase flow by Bai et al. (2010).

3.1.1 Bubbly Flow

Bubbly flow is characterized by discrete bubbles dispersed in a continuous liquid phase. The sizes of the bubbles are generally smaller than the particles. The small bubbles are nearly spherical in shape. On the contrary, large bubbles of various shapes tend to deform constantly during their movement, as shown in Fig. 6. Moreover, they are easily blocked in the pores, except when large bubbles become deformed or breakup into smaller bubbles.

![FIG. 5: Local heating surface temperature and its corresponding area](image-url)
3.1.2 Bubbly Slug Flow

Bubbles with sizes larger than those of the particles are defined as slugs. Bubbly slug flow is the transition from bubbly flow to slug flow, in which some unstable slugs and a great number of bubbles coexist and occur intermittently in the channel. In addition to the greater vapor quantity, larger size, and more frequent breakup and coalescence of bubbles, the bubbles in the bubbly slug flow are assumed to behave similarly to those in the bubbly flow. The slugs result from the development and combination of the bubbles, as shown in Fig. 7.

3.1.3 Slug Flow

In slug flow, the slugs become larger and deform more frequently than those in bubbly slug flow. Once the neighboring slugs meet each other, they can coalesce immediately. In addition, the bubbles break up and coalesce more frequently, as shown in Fig. 8.

3.1.4 Slug-Annular Flow

In this flow regime, continuous vapor and slugs occur in turn. Once the neighboring slugs meet each other, they coalesce and form a continuous vapor regime. The continuous vapor deforms and breaks up frequently. After the departure of the continuous vapors, the slugs occupy the channels. Then, the slugs develop, coalesce, and finally form a continuous vapor again. Figure 9 illustrates the process of continuous vapor formation through the development and coalescence of bubbles and slugs.
3.2 Effects of Heat Flux, Mass Flux, and Particle Diameter

3.2.1 Heat Flux

At a higher heat flux, boiling is more intense and therefore the bubbles and slugs generate, coalesce, deform, and break up more frequently. The number and size of the bubbles and slugs increase with an increase in the heat flux. In slug-annular flow, with an increase in the heat flux, the coverage and endurance of continuous vapor increase while the endurance of the slugs decreases.

3.2.2 Mass Flux

In a given flow regime, the size and number of bubbles decrease with an increase in the mass flux. At high mass flux, greater heat flux is required to achieve the same flow regime as that at low mass flux. At low mass flux, the bubbles are blocked in the pores with very slow movement and by the fully developed shapes.

3.2.3 Particle Diameter

At the same mass flux, inlet temperature, and heat flux, the bubbles and slugs in 5- and 6-mm-diameter particle channels are fewer and smaller, and remain for a shorter period of time than those in a 8-mm-diameter particle channel. This means that a higher heat flux is required to achieve the same flow regime in a flow channel composed of bigger particles.

4. BOILING HEAT TRANSFER RESULTS

4.1 Boiling Curves

When boiling occurs in a porous channel, a characteristic boiling curve can be presented in terms of wall superheat $T$ versus heat flux $q$ at the wall. Figure 10 shows the boiling curves for porous channels composed of particles with diameters of 5, 6, 7, and 8 mm. In the single-phase flow region, the heat transfer coefficient is low and has an approximately linear dependence on the wall superheat; in the two-phase flow region, the heat transfer coefficient increases dramatically. As shown in Fig. 10, boiling occurs with low superheat at the wall. Moreover, the wall superheat of the original nucleation boiling increases with an increase in the particle diameter. This indicates that the heat transfer performance is deteriorated by the increase in particle diameter.

4.2 Effect of Void Fraction

Figure 11 shows the heat transfer coefficients versus the void fractions. The variations of the heat transfer coefficients can be divided into three zones according to the two-phase flow patterns and the void fractions, i.e., low, middle, and high void fraction zones. Bubbly flow appears in the low void fraction zone, and the boiling heat transfer coefficient increases with an increase in the void fraction; Slug-annular flow is present in the high void fraction zone, and heat transfer coefficient increases dramatically with an increase in the void fraction; In the middle void fraction zone the heat transfer coefficient increases slowly, where the bubbly slug and slug flows occur.
FIG. 10: Boiling curves ($T_{\text{in}} = 80^\circ\text{C}$): (A) $d_p = 5$ mm; (B) $d_p = 6$ mm; (C) $d_p = 7$ mm; (D) $d_p = 8$ mm

FIG. 11: Effect of the void fraction on the heat transfer coefficient ($T_{\text{in}} = 80^\circ\text{C}$): (A) $d_p = 5$ mm; (B) $d_p = 6$ mm; (C) $d_p = 7$ mm; (D) $d_p = 8$ mm
4.3 Effect of Heat Flux

Figure 12 shows the heat transfer coefficients versus heat flux for porous channels composed of particles with diameters of 5, 6, 7, and 8 mm, where it can be seen that the effect of the heat flux on the heat transfer coefficient is significant. The boiling is more intense at higher heat flux, leading to the increase in the heat transfer coefficients.

4.4 Effect of Mass Flux

Figures 11 and 12 also show the heat transfer coefficients at different mass fluxes of porous channels composed of particles with diameters of 5, 6, 7, and 8 mm. The flow disturbance becomes stronger with an increase in the mass flux, which leads to a slight increase in the heat transfer coefficient. Consequently, the heat transfer coefficient increases slightly with increasing mass flux, especially at higher heat flux.

4.5 Effect of Particle Diameter

Figure 13 shows the heat transfer coefficients at different particle diameters. The results show that a larger particle diameter leads to a lower heat transfer coefficient. The smaller the particle diameter, the smaller is the pore size. Thus, as the particle diameter decreases the intensity of the disturbances in the fluid flow and the frequency of the bubbles splitting and merging increase. These phenomena can all lead to an increase in the heat transfer coefficients. Consequently, the boiling heat transfer coefficient increases with the decrease in the particle diameter.

**FIG. 12:** Effect of the heat and mass flux on the heat transfer coefficient ($T_{in} = 80^\circ$C): (A) $d_p = 5$ mm; (B) $d_p = 6$ mm; (C) $d_p = 7$ mm; (D) $d_p = 8$ mm
4.6 Correlation between the Boiling Heat Transfer Coefficient and Experimental Data

In the present study, the porosity and the dimensionless boiling number Bo were employed in order to take into account the effects of particle diameter and heat flux. Thus, the heat transfer in porous media composed of particles is related to Re, Pr, Bo, and $\varepsilon$, and the experimental data can be correlated by the following equations:

$$\text{Nu} = a \cdot \varepsilon^{-b} \cdot \text{Bo}^c \cdot \text{Re}_l^d \cdot \text{Pr}_l^e$$

$$\text{Nu} = \frac{\alpha d_p}{\lambda}, \quad \text{Bo} = \frac{q}{G h_{fg}}, \quad \text{Re}_l = \frac{\rho_l J_l d_p}{\eta_l} = \frac{G (1 - x) d_p}{\eta_l}, \quad \text{Pr}_l = \frac{\nu_l}{\alpha_l}$$

where $a$, $b$, $c$, $d$, and $e$ are constants; $\lambda$ is the thermal conductivity of the fluid; $\rho_l$ is the density of the liquid; $J_l$ is the superficial velocity of the liquid; $\eta_l$ is the dynamic viscosity of the liquid; $\nu_l$ is the kinematic viscosity of the liquid; and $\alpha_l$ is the thermal diffusivity of the liquid. Based on the experimental data, the constants are fitted as follows: $a = 0.447$, $b = 0.425$, $c = 0.741$, $d = 0.697$, and $e = 0.330$. The mean relative deviation of the correlation is about $\pm 16\%$, as shown in Fig. 14.

FIG. 14: Comparison of the current model of the boiling heat transfer coefficient with the present experimental data: the right-hand side is an enlarged view of the data points indicated by the circle ring in the left-hand side.
5. CONCLUSIONS

This paper presents an experimental study on the boiling flow and heat transfer characteristics in a porous media composed of spherical heating particles. Boiling two-phase flow patterns were observed, and four flow regimes have been identified, i.e., bubbly flow, bubbly slug flow, slug flow, and slug-annular flow. It was shown that the heat flux, mass flux, and particle diameter have a great influence on these flow regimes. For example, a higher heat flux is required to produce the same flow regime at higher mass flux with particles of smaller diameter. Consequently, the boiling heat transfer in porous media is affected by the heat flux, mass flux, and particle diameter. The heat transfer coefficient increases with the increase in heat and mass flux. A larger particle diameter leads to a lower heat transfer coefficient, which results in a higher wall superheat of the original nucleation boiling. The variations in the heat transfer coefficient can be divided into three zones according to the flow patterns and void fraction values. Based on the experimental data, a correlation is proposed to calculate the boiling heat transfer coefficient by taking into consideration the effects of the pebble bed porosity and the Reynolds, Prandtl, and boiling numbers.

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