Effect of Flow Directions on Multiphase Flow Boiling Heat Transfer Enhanced by Suspending Particles in a Circulating Evaporation System

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Abstract

A circulating fluidized bed evaporator (including down-flow, horizontal, and up-flow beds) was constructed to study the effect of flow directions on multiphase flow boiling heat transfer. A range of experimental investigations were carried out by varying amount of added particles (0–2%), circulation flow rate (2.15–5.16 m³/h) and heat flux (8–16 kW/m²). The comparison of heat transfer performance in different vertical heights of the horizontal bed was also discussed. Results reveal that the glass bead particle can enhance heat transfer compared with vapor–liquid two-phase flow for all beds. At a low heat flux ($q = 8 \text{ kW/m}^2$), the heat-transfer-enhancing factor of the horizontal bed is obviously greater than those of the up-flow and down-flow beds. With the increase in the amount of added particles, the heat-transfer-enhancing factors of the up-flow and down-flow beds increase, whereas that of the horizontal bed initially increases and then decreases. However, at a high heat flux ($q = 16 \text{ kW/m}^2$), the heat-transfer-enhancing factors of the three beds show an increasing tendency with the increase in the amount of added particles and become closer than those at a low heat flux. For all beds, the heat-transfer-enhancing factor generally increases with the circulation flow rate but decreases with the increase in heat flux.

Keywords Heat transfer enhancement · Circulating fluidized bed evaporator · Flow direction · Down-flow bed · Up-flow bed · Horizontal bed

Introduction

Fouling in industrial heat exchangers is a complex, serious problem that occurs when a fouling substance builds along the walls of a heat exchanger, thereby limiting the efficiency of the heat transfer within the system. It has a significant effect on equipment performance, thereby increasing projected overhead and maintenance costs [1–5]. As a promising alternative to the conventional fouling control technology, circulating fluidized bed heat exchanger can reduce investment and maintenance costs [6] and has been used for a wide range of industrial processes, such as catalytic cracking [7], paper making [8], desalination [9, 10], wastewater evaporation [6, 11], mitigation of ice crystallization [12], and pharmaceuticals [13]. Numerous researchers

have made extensive investigation on the heat transfer and hydrodynamic performance of circulating fluidized bed heat exchanger [14–28].

On the basis of the combination flow boiling heat transfer with multiphase fluidization, the performance of circulating fluidized bed heat exchanger is affected by several key factors, such as type of particles, amount of added particles, circulation flow rate, heat flux, and flow directions.

In a circulating fluidized bed exchanger with different flow directions, particle distribution and collision frequency between particles and wall may be affected by gravity, which results in the difference in hydrodynamics and heat transfer performance. However, experiments and numerical simulation in current studies have mainly concentrated on up-flow bed. Reports on the characteristics of heat transfer in down-flow and horizontal beds are limited. Wen et al. [14] reported that the heat transfer coefficient of vapor–liquid (V–L) two-phase flow boiling declined significantly without the addition of polytetrafluoroethylene particles. Meanwhile, the vapor–liquid–solid (V–L–S) three-phase flow boiling could maintain a slightly higher heat transfer coefficient for more than 1500 h because the solid particles sheared the heat

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transfer wall and kept it clean. Liu et al. [15] investigated the hydrodynamic and heat transfer in a V–L–S flow boiling system with a charge-coupled device (CCD) measuring technique to illustrate the presence of solid particles enhancing the boiling evaporation process, and found that the heat transfer coefficient of the system increased with increases in the heat flux and amount of added solid particles. Li et al. [16] reported that the boiling heat transfer coefficient of a V–L–S three-phase fluidized bed was about twice that of the V–L two-phase flow. Sefiane et al. [17, 18] performed visualization studies and proposed the “bubble push” mechanism to account for the effect of reduced particle size and high heat flux having a reduced enhancement. They also developed a correlation to describe heat transfer during boiling in a three-phase circulating fluidized bed. Andersson [19] investigated the effect of bed particle size on heat transfer in a circulating fluidized bed and found that particle diameter in the range of 0.22–0.44 mm was independent on the wall-to-bed heat transfer coefficient. Wang et al. [20] studied the solid particle distribution in the heating tube bundle of a

transparent multitube V–L–S circulating fluidized bed evaporator by using the CCD measuring system and found that the heterogeneity of particle distribution decreases with the increases in circulation flow rate, heat flux, and amount of added particles but increases with the increase in the settling velocity of particles. Pronk et al. [21] attributed the deposit removal from heat exchanger walls and the enhancement of heat transfer to particle collisions with the wall. Kim and Lee [22] investigated the glass bead–water flow and found that pressure loss and heat transfer characteristics were closely related with the wall-hitting frequency and particle slip velocity. Ma et al. [23] experimentally investigated the collision behavior of fluidized solid particles on the tube wall of a graphite evaporator via vibration signal analysis. The hydrodynamic characteristics in the bottom region of circulating fluidized bed risers with fluid catalytic cracking particles were studied by Zhu and Zhu [24] over a wide range of operating conditions. The results showed the radial solid concentration and corresponding radial profiles of standard deviation, particle velocity profiles, and probability density

Fig. 1 V–L–S three-phase circulating fluidized bed evaporator. 1—centrifugal pump; 2—water inlet; 3—down-flow bed; 4—horizontal bed; 5—up-flow bed; 6—particle inlet; 7—particle collector; 8—

electromagnetic flowmeter; 9—evaporation chamber; 10—content gauge; 11—sight glass; 12—condenser; 13–23—ball valves

Fig. 2 Distribution of RTDs on the wall of bed in different directions. **a** Down-flow bed, **b** up-flow bed, **c** horizontal bed

distributions. Zheng and Zhu [25] performed a pressure balance analysis and derived a dimensionless empirical correlation for the solid holdup and a semiempirical equation for the pressure drop to predict the stable operating conditions in a liquid–solid (L–S) circulating fluidized bed successfully.

Han et al. [26] presented a new approach to calculating the onset velocity using CFD–DEM simulation of the particle residence time distribution in L–S two-phase system, and their calculation results were in reasonable agreement with experimental data. Wang et al. [27] simulated the flow

Fig. 3 Positions of RTDs relative to the resistance wire on the horizontal bed

Table 1 Specifications of experimental apparatuses

Apparatus	Type	Measuring range	Accuracy class	Maximum permissible error
RTD	Pt100	− 100 to 300 °C	A	$\pm (0.15 + 0.002 t)$
Electromagnetic flowmeter	LDG-32-1.6 MPa	0.8 to 25 m ³ /h	0.5	$\pm 0.2\%$
Pressure transducer	MIK-P300G	− 0.1 to 0.1 MPa	0.5 (%FS)	± 0.5 kPa

$|t|$ is the absolute value of temperature

Table 2 Physical properties of glass beads

Item	Glass bead
Shape	Spherical
Mean size (mm)	$\Phi 0.9$
Density (kg/m ³)	2405
Sedimentation velocity (m/s)	0.124
Thermal conductivity [W/(m °C)]	1.1

The sedimentation velocity in water is measured at 20 °C under atmospheric pressure

behavior of solid phase using Eulerian–Eulerian method in an L–S fluidized bed with modified drag model on the basis of energy-minimization multiscale method. Qi and Jiang [28] adopted the Eulerian multiphase fluid model to simulate the particle distribution in a vertical fluidized bed heat exchanger incorporating tube bundle arranged in parallel. The results showed good agreement with experimental ones. They concluded that particle distribution became uniform at high velocity, and that particle maldistribution occurred with the increase in density or diameter of particles.

With the development of petrochemical industry and fluid transportation recently, the investigation on hydrodynamics and heat transfer performance in horizontal and down-flow bed exchangers has attracted considerable attention. Moreover, the systematic comparison of performance of beds with different directions is necessary. Zhang et al. [29] studied the particle distribution in a horizontal tube bundle after reforming the front header of an L–S horizontal circulating bed evaporator by using two inlets and adding a baffle plate. They found that the angle of the baffle plate and particle diameter affect the performance of particle distribution. Liu et al. [30] studied the particle distribution of a single-pipe horizontal circulating fluidized bed with a spiral flow generator under different operating conditions. Li et al. [31] investigated the particle distribution in up-flow and down-flow beds and found that the up-flow bed showed an even particle distribution compared to the down-flow one and determined the optimal range of operating parameters under experimental conditions.

In this work, a V–L–S three-phase circulating fluidized bed evaporator was designed and built to compare the

difference of thermal performance in different flow directions. The influence of operating parameters, such as the amount of added particles, circulation flow rate, and heat flux, is systematically investigated. The characteristics of heat transfer at different vertical heights of the horizontal bed were also discussed to further investigate the effect of gravity. The research findings may serve as a valuable reference for industries that apply circulating fluidized bed heat exchanger technology.

Experiments

Experimental Apparatus

A circulating fluidized bed evaporator was designed and constructed to study the effect of flow directions on multiphase flow boiling heat transfer, as shown in Fig. 1. The evaporator consists of a heating section, evaporation chamber, condenser, and particle collector.

The apparatus was made of stainless steel. The heating section included down-flow, horizontal, and up-flow beds. The tube length and diameter of each bed were 1200 mm and $\Phi 38$ mm $\times$ 3 mm, respectively. The heating method involved direct heating via the resistance wire around the heating pipe. In order to reduce heat loss to the surroundings, the heating tube was covered by insulation cotton. The external wall temperatures of three beds were measured by resistance temperature detectors (RTDs), and the distribution of RTDs is illustrated in Fig. 2. For the down-flow and up-flow beds (Fig. 2a, b), three RTDs were mounted along one side of each bed to measure the external wall temperature, respectively. For the horizontal bed (Fig. 2c), particle distribution may be uneven at different vertical heights of the same cross section due to gravity; thus, nine RTDs were divided into three groups and mounted to measure the external wall temperature. Figure 2a shows the installation method of RTDs in detail. The probes of RTDs were coated with silicon rubber not only for fixation, but also for weakening the effect of thermal radiation from the resistance wire [32]. Figure 3 displays the positions of RTDs relative to the resistance wire in detail, and the exact vertical heights of three groups of RTDs on the horizontal bed. The inlet

and outlet temperatures of the fluid in each bed were also surveyed by RTDs. Pressure transducers were amounted to measure the pressure of fluid at the inlet and outlet of each bed (the range was from -0.1 to 0.1 MPa). A particle collector was used to collect the existing particles in the system when the particles need to be changed.

The experiment was initiated by adding a certain amount of working fluid and solid particles to the system. The frequency of the circulation pump was adjusted to the specified circulation flow rate. The heat power was

controlled by a solid-state voltage regulator. The temperature and pressure were recorded when the V–L–S three-phase flow boiling reached a steady state.

The data collection device consisted of digital display instruments (AI-501MF, Xiamen Yudian Automation Technology Limited Company, China) and a computer with KingView software. The pressure transducers and RTDs were combined with the digital display instruments. The specifications of the relevant apparatuses are listed in Table 1.

Fig. 4 Comparison of convective heat transfer coefficients of beds in different directions: **a** two-phase, $\varphi = 0\%$, $q = 8 \text{ kW/m}^2$; **b** three-phase, $\varphi = 1\%$, $q = 8 \text{ kW/m}^2$; **c** two-phase, $\varphi = 0\%$, $q = 12 \text{ kW/m}^2$; **d**

three-phase, $\varphi = 1\%$, $q = 8 \text{ kW/m}^2$; **e** two-phase, $\varphi = 0\%$, $q = 16 \text{ kW/m}^2$; **f** three-phase, $\varphi = 1\%$, $q = 8 \text{ kW/m}^2$

Experimental Materials and Parameters

Glass bead particles and pure water were used as the working medium in the experiment. The physical properties of glass beads are listed in Table 2.

The adjustable operating variables used in the experiment included the amount of added particles, circulation flow rate, and heat flux.

The amount of added particles φ ($\varphi = 0, 0.5\%, 1\%, 1.5\%$, and 2%) is the ratio of the bulk volume of solid particles to the total volume of the working medium in the system. The circulation flow rate V was set to 2.15, 2.88, 3.61, 4.32, and $5.16 \text{ m}^3/\text{h}$. The input electric power was controlled at 1200, 1800, and 2400 W when heat was supplied. The heat flux ($q = 8, 12$, and 16 kW/m^2) was calculated by dividing the input power by the external surface area of each bed.

Data Processing

The convective heat transfer coefficient α is used as an evaluation parameter for the thermal performance of beds in different directions. It can be obtained as follows:

$$\alpha = \frac{q}{t_{iw} - t_f} = \frac{Q}{S_o(t_{iw} - t_f)}, \quad (1)$$

where Q is the heating power; S_o is the external surface area of each bed; and t_{iw} and t_f are the average internal wall temperature and temperature of fluid, respectively. t_{iw} and t_f can be obtained as follows:

$$t_{iw} = t_{ow} - \frac{Q \ln\left(\frac{d_o}{d_i}\right)}{2\pi L \lambda}, \quad (2)$$

$$t_f = \frac{t_{fi} + t_{fo}}{2}, \quad (3)$$

where d_o and d_i are the outer and inner diameters of the bed, respectively; λ is the thermal conductivity of the wall; L is the length of the bed; t_{fi} and t_{fo} are the inlet and outlet temperatures of fluid, respectively; and t_{ow} is the average external wall temperature of the bed.

For the down-flow bed,

$$t_{ow} = \frac{t_2 + t_3 + t_4}{3}; \quad (4)$$

Fig. 5 Effect of the amount of added particles on convective heat transfer coefficient: **a** $V = 2.15 \text{ m}^3/\text{h}$, $q = 8 \text{ kW/m}^2$; **b** $V = 4.32 \text{ m}^3/\text{h}$, $q = 8 \text{ kW/m}^2$; **c** $V = 2.15 \text{ m}^3/\text{h}$, $q = 16 \text{ kW/m}^2$; **d** $V = 4.32 \text{ m}^3/\text{h}$, $q = 16 \text{ kW/m}^2$

for the horizontal bed,

$$t_{ow} = \frac{t_7 + t_8 + t_9 + t_{10} + t_{11} + t_{12} + t_{13} + t_{14} + t_{15}}{3}; \quad (5)$$

and for the up-flow bed,

$$t_{ow} = \frac{t_{18} + t_{19} + t_{20}}{3}. \quad (6)$$

To further investigate the effect of gravity on the fluidization of particles and thermal performance in horizontal bed, the convective heat transfer coefficients (i.e., α_1 , α_2 , and α_3) of three different vertical heights (i.e., h_1 , h_2 , and h_3) in the horizontal bed are investigated and discussed. Taking the horizontal plane through the axis of horizontal bed as the zero point of the height, then $h_1 = \frac{d_i}{2} = 0.016$ m, $h_2 = 0$, and $h_3 = -\frac{d_i}{2} = -0.016$ m, respectively.

α_1 , α_2 , and α_3 can also be calculated by using Eqs. (1)–(3), repeatedly, where t_{ow} is the average of T7, T10, and T13 for α_1 , of T8, T11, and T14 for α_2 , and of T9, T12, and T15 for α_3 , respectively.

The enhancing factor E describes the enhancing effect of the heat transfer performance. It is expressed as follows:

$$E = \frac{\alpha_{th} - \alpha_t}{\alpha_t} \times 100, \quad (7)$$

where α_t and α_{th} are the experimentally determined two-phase and three-phase convective heat transfer coefficients, respectively.

The uncertainty of the experimental results is specified on the basis of deviations in different measured parameters involved in the experiment. The uncertainty of the parameter in this experiment is calculated by the following equations [32]:

$$\frac{u(\alpha)}{\alpha} = \sqrt{\left[\frac{u(Q)}{Q} \right]^2 + \frac{u^2(t_{iw}) + u^2(t_1)}{(t_{iw} - t_1)^2}}, \quad (8)$$

$$\frac{u(E)}{E} = \sqrt{\left[\frac{u(\alpha_{th})}{\alpha_{th}} \right]^2 + \left[\frac{u(\alpha_t)}{\alpha_t} \right]^2}. \quad (9)$$

Fig. 6 Effect of the amount of added particles on heat-transfer-enhancing factor: **a** $V = 2.15$ m³/h, $q = 8$ kW/m²; **b** $V = 4.32$ m³/h, $q = 8$ kW/m²; **c** $V = 2.15$ m³/h, $q = 16$ kW/m²; **d** $V = 4.32$ m³/h, $q = 16$ kW/m²

The maximum uncertainties for the convective heat transfer coefficient and heat-transfer-enhancing factor are calculated within 9.5% and 12.6%, respectively.

Experimental Results and Discussion

Convective Heat Transfer Coefficient

Figure 4 compares the convective heat transfer coefficients of beds in different directions. From Fig. 4a, c, e, the two-phase convective heat transfer coefficient is larger in the up-flow bed than those in the horizontal and down-flow beds. The difference of convective heat transfer coefficients among the three beds decreases with the increase in heat flux. Figure 4b, d, f shows the comparison of the three-phase convective heat transfer coefficients of beds in different directions. The result is similar to that of two-phase flow.

In the up-flow bed, the direction of buoyance is the same as that of the shear force exerted by the fluid. This condition makes it easy for the generated bubbles to detach from the wall during boiling, thereby increasing the frequency

of bubble generation and the disturbance to the boundary layer. Hence, the convective heat transfer coefficient is high.

Figure 4 also shows that the presence of particles can enhance the heat transfer for all the beds. The fluidization of particles can destroy the heat transfer boundary layer near the tube wall, cause additional convection, and improve the turbulent extent. Meanwhile, the fluidized particles can also increase the active nucleation sites [15], thereby enhancing the convective boiling heat transfer.

Effect of the Amount of Added Particles on Heat Transfer Enhancement

Figure 5 shows the relationship between convective heat transfer coefficient and the amount of added particles. It can be seen that the convective heat transfer coefficient is larger in up-flow bed than those in down-flow and horizontal beds with different amount of added particles. Figure 6 shows the relationship between enhancing factor and the amount of added particles. At a low heat flux ($q = 8 \text{ kW/m}^2$), the heat-transfer-enhancing factor is significantly greater in the horizontal bed than those in the up-flow and down-flow

Fig. 7 Effect of circulation flow rate on heat-transfer-enhancing factor: **a** $\phi = 1\%$, $q = 8 \text{ kW/m}^2$; **b** $\phi = 2\%$, $q = 8 \text{ kW/m}^2$; **c** $\phi = 1\%$, $q = 16 \text{ kW/m}^2$; **d** $\phi = 2\%$, $q = 16 \text{ kW/m}^2$

beds. With the increase in the amount of added particles, the heat-transfer-enhancing factors increase in the up-flow and down-flow beds, whereas that in the horizontal bed, it initially increases and then decreases, as shown in Fig. 6a, b.

Numerous particles deposit on the bottom and move forward along the tube wall surface due to the gravity in the horizontal bed. The holdup of the particles on the bottom of the bed is high, which shears and destroys the laminar sublayer near the wall and increases the boiling nucleation sites. Thus, the heat-transfer-enhancing factor of the horizontal bed is higher than those of the up-flow and down-flow beds. However, the accumulation of particles on the bottom of the tube can also hinder the bubbles from breaking away from the wall and moving into the fluid, which impedes the formation of bubble and perturbation of liquid, weakening the heat transfer enhancement [33]. Thus, the heat-transfer-enhancing factor of the horizontal bed increases initially and then decreases under the combined effect of the aforementioned factors.

The heat-transfer-enhancing factors of all the beds show an increasing tendency with the increase in the amount of added particles at a high heat flux ($q = 16 \text{ kW/m}^2$), as shown in Fig. 6c, d. Meanwhile, the heat-transfer-enhancing factors of all the beds obviously decrease and become closer

compared with Fig. 6a, b. This is due to the increases in the contact area and frequency among the particles, and those between the particle and the tube wall with the increase in the amount of added particles, which further enhances the heat transfer [15]. However, the superheating degree of the wall and the boiling nucleation sites increases with the heat flux [34], which weakens the effect of particles on heat transfer enhancement.

Therefore, the added particles can evidently enhance the thermal performance when the heat flux is low (8 kW/m^2). With the increase in heat flux, the effect of added particles on heater transfer enhancement can be reduced.

Effect of Circulation Flow Rate on Heat Transfer Enhancement

Figure 7 shows the effect of circulation flow rate on heat-transfer-enhancing factor. From the figure, the heat-transfer-enhancing factors of the three beds tend to increase. At a low heat flux and a small amount of added particles, the heat-transfer-enhancing factor of the horizontal bed is obviously higher than those of up-flow and down-flow beds. With the increase in the amount of added particles, heat flux and circulation flow rate, the difference of

Fig. 8 Effect of heat flux on convective heat transfer coefficient: **a** $\phi = 0.5\%$, $V = 2.15 \text{ m}^3/\text{h}$; **b** $\phi = 0.5\%$, $V = 4.32 \text{ m}^3/\text{h}$; **c** $\phi = 2\%$, $V = 2.15 \text{ m}^3/\text{h}$; **d** $\phi = 2\%$, $V = 4.32 \text{ m}^3/\text{h}$

the heat-transfer-enhancing factors among the three beds decreases.

The increase in the circulation flow rate results in the enhancement of the turbulent extent, which thins the thickness of laminar sublayer and enhances convective heat transfer [16]. Meanwhile, the impact of particles on the wall is more intense [20], which is beneficial to breaking the heat transfer boundary layer and increasing heat transfer. However, the increase in velocity also restrains the nucleate boiling heat transfer to some extent [16]. Thus, the heat-transfer-enhancing factor slightly increases with the circulation flow rate under the combined effect of the factors above.

Effect of Heat Flux on Heat Transfer Enhancement

Variations of convective heat transfer coefficient and heat-transfer-enhancing factor with the heat flux under different amounts of added particles and circulation flow rates are shown in Figs. 8 and 9. From the figures, the convective heat transfer coefficient and heat-transfer-enhancing factor generally decreases with the increase in heat flux. According to Sefiane et al. [17], with the increase in heat flux, the

generated bubbles coalesce to form large vapor attaching on the heating tube wall. This prevents the fluidized particles from contacting the wall and restricts the collision frequency and strength between the particles and the wall, thereby degrading the heat transfer enhancement. Meanwhile, the boiling nucleation sites increases with the increase in heat flux, which weakens the effect of the heat transfer enhancement of fluidization bed. Hence, with the increase in heat flux, the heat-transfer-enhancing factor decreases, and the differences of convective heat transfer coefficients and heat-transfer-enhancing factors of the three beds decrease.

Thermal Performance of Horizontal Bed

Figure 10 presents the heat transfer coefficients of different vertical heights in the horizontal bed. From Fig. 10a, d, it can be seen that the two-phase convective heat transfer coefficients of different vertical heights are almost the same in the horizontal bed. However, the case is different for the three-phase flow. At a low heat flux ($q = 8 \text{ kW/m}^2$), $\alpha_3 > \alpha_2 > \alpha_1$, whereas at high heat flux (16 kW/m^2), $\alpha_3 < \alpha_2 < \alpha_1$, as shown Fig. 10b, c, e, f.

Fig. 9 Effect of heat flux on heat-transfer-enhancing factor: **a** $\phi = 0.5\%$, $V = 2.15 \text{ m}^3/\text{h}$; **b** $\phi = 0.5\%$, $V = 4.32 \text{ m}^3/\text{h}$; **c** $\phi = 2\%$, $V = 2.15 \text{ m}^3/\text{h}$; **d** $\phi = 2\%$, $V = 4.32 \text{ m}^3/\text{h}$

The nucleation boiling sites are less at a low heat flux. Further accumulation of particles on the bottom can increase the nucleation sites, which also indicates a high convective heat transfer coefficient. However, the number of boiling nucleation sites greatly increases at a high heat flux, and the effect of particles is reduced. Moreover, the accumulation of particles on the bottom hinders the detachment of the generated bubbles, which will coalesce to form large vapors attaching on the heating tube wall and deteriorate the heat transfer enhancement.

Figure 11 shows the comparison of heat-transfer-enhancing factors at different vertical heights in the horizontal bed.

Conclusions

For two- and three-phase flows, the convective heat transfer coefficient of the up-flow bed was the largest, whereas that of the horizontal bed was the smallest. The difference

Fig. 10 Comparison of the convective heat transfer coefficients at different vertical heights in the horizontal bed: **a** $\phi = 0\%$, $q = 8 \text{ kW/m}^2$; **b** $\phi = 0.5\%$, $q = 8 \text{ kW/m}^2$; **c** $\phi = 2\%$, $q = 8 \text{ kW/m}^2$; **d** $\phi = 0\%$, $q = 16 \text{ kW/m}^2$; **e** $\phi = 0.5\%$, $q = 16 \text{ kW/m}^2$; **f** $\phi = 2\%$, $q = 16 \text{ kW/m}^2$

Fig. 11 Comparison of heat-transfer-enhancing factors at different vertical heights in the horizontal bed: **a** $\varphi = 0.5\%$, $q = 8 \text{ kW/m}^2$; **b** $\varphi = 0.5\%$, $q = 16 \text{ kW/m}^2$; **c** $\varphi = 2\%$, $q = 8 \text{ kW/m}^2$; **d** $\varphi = 2\%$, $q = 16 \text{ kW/m}^2$

of convective heat transfer coefficient among different beds decreased with the increase in heat flux.

The presence of particles enhanced the boiling heat transfer for all beds. The maximum improvement of 39.8% was found at $q = 8 \text{ kW/m}^2$, $\varphi = 1\%$ and $V = 4.32 \text{ m}^3/\text{h}$. When the heat flux was 8 kW/m^2 , the heat-transfer-enhancing factor of the horizontal bed was obviously higher than those of the up-flow and down-flow beds. With the increase in the amount of added particles (0.5%–2.0%), the heat-transfer-enhancing factors of the up-flow and down-flow beds increase, whereas that of the horizontal bed initially increased and then decreased.

With the increase in the circulation flow rate, the heat-transfer-enhancing factors of the three beds tended to increase. However, the heat-transfer-enhancing factors decreased for three beds with the increase in heat flux.

The heat transfer performance at different vertical heights in the horizontal bed was also investigated. When the heat flux was 8 kW/m^2 , $\alpha_3 > \alpha_2 > \alpha_1$, when heat flux was 16 kW/m^2 , $\alpha_3 < \alpha_2 < \alpha_1$.

These results may serve as a valuable reference for the academic investigation and industrial applications of

circulating fluidized bed evaporators, which can be used to prevent fouling and enhance heat transfer.

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