EXPERIMENTAL STUDY ON BOILING HEAT TRANSFER IN NEGATIVE-PRESSURE FLOWING WATER IN A VERTICAL ANNULAR TUBE

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To study energy recovery from low-temperature wastewater during industrial production, a water-boiling heat-transfer experimental device was built. A casing evaporator with a length of 1450 mm, an inner tube diameter of 30 mm, and an annular space gap of 14.2 mm is taken as the main research object. The boiling heat transfer characteristics of water in the annular tube area of the casing evaporator were experimentally studied by adjusting the inlet temperature (60–85 ℃) of hot water and the pressure in the annular tube (12.1–57.6 kPa). The results show that, as the pressure of the system decreases, the boiling phenomenon in the annular tube becomes more intense and the convective heat transfer coefficient increases. The boiling flow and average surface convective heat transfer enhancement rates (β) were 2.2 and 1.5 when an initial pressure of 1 kPa was used. When the flow rate of the working medium in the annular tube was 1.69 kg·m^{-2}*·s*$^{-1}$*, the convective heat transfer coefficient gradually increased with the temperature and then stabilized when the inlet temperature reached between 80 ℃ and 85 ℃. These results reveal the characteristics of boiling-water heat transfer in an annular tube and aid in the design of heat pipe systems for waste-heat recovery.*

Key words*: annular tube, boiling heat transfer, convective heat transfer coefficient, experimental study*

1. Introduction

Waste heat utilization at low temperature levels is difficult to resolve in the global energy industry. At least 50% of the industrial energy consumption is abandoned via various forms of waste heat. In some projects with large industrial water consumption rates, such as the printing and dyeing industries, the typical wastewater discharge temperature is 45–80 ℃ [1]. Recovery of this low-grade waste heat could become an effective way to solve the energy problem. Low-grade heat recovery is considered to be the generation of the various energy demands, such as cooling, heat, and power [2]. As an efficient heat exchanger, the heat pipe plays an important role in waste heat recovery [3–6]. The boiling heat transfer efficiency of an evaporator directly affects the operational efficiency of the system.

Enhancement of the boiling heat transfer performance is also critical for improving the energy efficiency among various industrial applications. Additionally, boiling heat transfer is among the important technologies used by researchers [7]. Many factors affect boiling heat transfer in a tube, including the heat flux density, working medium flow rate, saturation pressure, heating surface properties, pipe diameter, and tube inclination angle [8]. In their research on boiling heat transfer in tubes, Chen *et al.* [9] summarized the research on the heat transfer characteristics and dynamic characteristics of fluids in annular tubes. Xu [10] studied heat fluxes, coolants, and their application fields using heat transfer, critical heat flux, and bubble behavior visualization experimental platforms. The authors of Refs. [11,12] experimentally studied the changes of flow patterns, heat-transfer characteristics, dynamic temperature changes, boiling heat transfer, and pressure drop in microchannel evaporators. Ma [13] experimentally studied variations in flowing, boiling heat transfer coefficients in three-dimensional, enhanced pipes. They considered the mass flow rate, dryness, pipe diameter, and saturation temperature. With regard to two-phase flow patterns, Mishima *et al.* [14] conducted an experimental study on the characteristics of a two-phase gas–liquid flow in three narrow rectangular channels with a fixed width of 40 mm and narrow slit widths of 1.07 mm, 2.45 mm, and 5 mm. Bubble movement in the narrow rectangular channel was limited by the wall and was deformed. Bubble, slug, stirred, and annular flows were the main types of flow observed. Situ *et al.* [15] studied the bubble behaviors of subcooled flows boiling in a vertical annular channel. Their experimental results showed that the bubble separation frequency first increases and then decreases with increasing heat flux and that the bubble behavior interacts with the temperature and flow fields. Pan *et al.* [16] studied bubble morphologies and heat transfer in a vertical, rectangular, narrow slot channel via high-speed photography and identified the reason for boiling heat transfer enhancement in a narrow slot.

The application of heat-pipe heat exchangers to the industrial wastewater waste heat recovery field is of great significance. Although heat transfer in an annular pipe has been widely studied, many unknown factors remain [8]. Therefore, to better understand waste heat recovery from industrial wastewater, we built a boiling heat-transfer experimental device with water as the working medium. Then, we performed a quantitative study of how the temperature distribution within the working medium and the convective heat transfer coefficient in an annular tube change with the hot-water inlet temperature and pressure in the annular tube.

2. Experimental apparatus and method

2.1. Experimental apparatus

To study the boiling heat transfer of water in the annular tube, the experimental device shown in Fig. 1 was built.

The test device comprises hot-water circulation, heat pipe, and data acquisition systems. The hot-water circulation system controls the water temperature by adjusting the electric heating power delivered to the water heating tank and uses the water pump and rotameter to control the flow. The heat pipe system is the main test system. Its evaporator is a stainless-steel casing heat exchanger with a length of 1450 mm, an inner diameter of 30 mm, an annular space gap of 14.2 mm, and a wall thickness of 0.8 mm. The working medium on the tube side is hot water from the hot-water circulation system. The working medium on the shell side (annular tube) is distilled water from the heat-pipe system. The circulation process is as follows: distilled water enters the casing heat exchanger under pressure from the magnetic pump, absorbs heat from the hot water circulating in the inner pipe, and undergoes negative-pressure boiling. The generated two-phase gas–liquid flow enters the condenser for cooling and then returns to the liquid reservoir to complete circulation. The data acquisition system comprises pressure sensors, PT1000 platinum resistance temperature sensors, and a data acquisition module. The function of the latter item is to acquire experimental data, such as the temperature, pressure, and flow data.

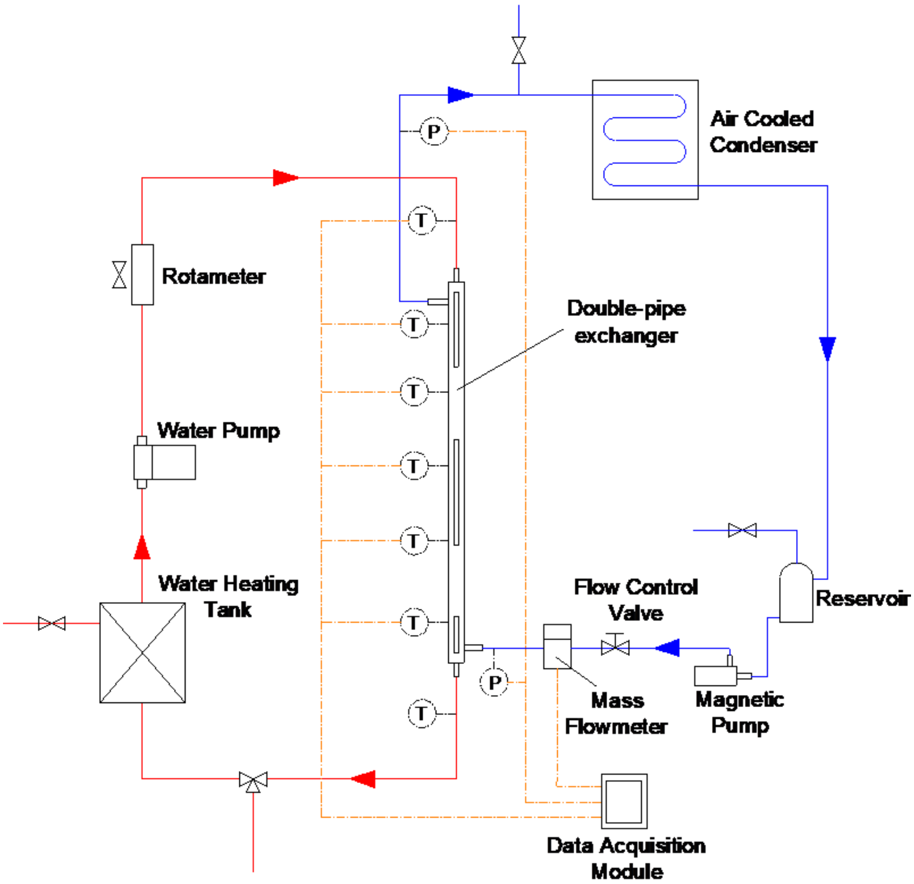

Fig. 1. Experimental device for boiling heat transfer of water.

In addition, to facilitate observation and recording of flow pattern changes within the two-phase flow in the annular pipe, three transparent windows with visual widths of 25 mm are installed on the outer wall of the annular pipe. The entire test device is insulated with insulation cotton to reduce heat loss.

2.2. Data measurement method

Fig. 2 shows the temperature and pressure measurement schemes used at each position in the casing heat exchanger. Temperature sensors were placed at the inlets and outlets of the annular pipe and the inner pipe. Five groups of temperature measurement points are located at 100, 500, 780, 1160, and 1360 mm away from the bottom of the annular tube. Each temperature measurement point comprises three temperature sensors that measure the temperatures of the outer wall of the inner tube, the fluid in the annular tube, and the outer wall of the outer tube, respectively. Pressure sensors are located at the inlet and outlet of the annular pipe.

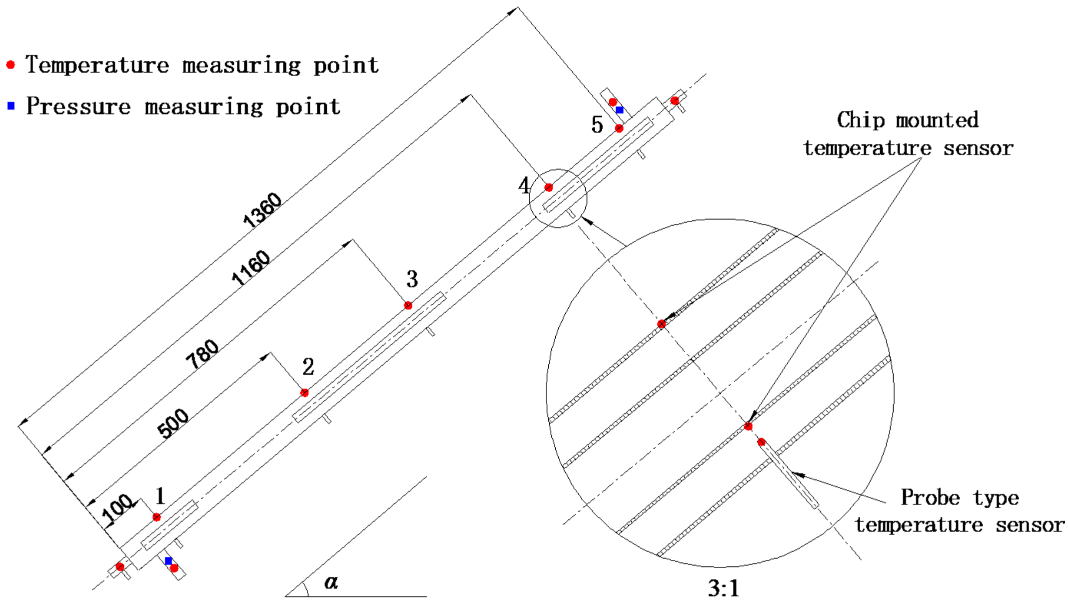

Fig. 2. Temperature and pressure measurement scheme.

For flow measurement, an LZB-10 rotameter and RHE 14 mass flowmeter are employed to measure the flow of the working medium in the hot-water circulation and heat pipe systems, respectively. The data measured by the temperature sensor, pressure sensor, and RHE 14 mass flowmeter are collected by the Agilent 34972a data acquisition instrument.

3. Data reduction

Heat exchange between the working medium in the annular pipe and the circulating hot water is described by Eq. (1).

$$Q_{\mathrm{hp}} = Q_{\mathrm{hw}} - Q_{\mathrm{s}} \tag{1}$$

$$Q_{\mathrm{hw}} = q_{m,\mathrm{hw}} c_p \Delta T_{\mathrm{hw}} \tag{2}$$

$$Q_{\mathrm{s}} = \pi d_{\mathrm{ins}} L_{\mathrm{an}} K_{\mathrm{s}} \Delta T_{\mathrm{w}} \tag{3}$$

where

$$K_{\mathrm{s}} = \frac{1}{\dfrac{d_{\mathrm{ins}}}{2\lambda_{\mathrm{ins}}} \ln \dfrac{d_{\mathrm{ins}}}{d_{\mathrm{o,an}}} + \dfrac{1}{h_{\mathrm{ins,air}}}} \tag{4}$$

Based on the measured outlet pressure and the density and liquid level height of the hydraulic medium, the pressure at each height in the pipe is calculated using Eq. (5) [17].

$$p_H = p_{\mathrm{in,an}} - \rho_1 g H \tag{5}$$

According to thermodynamic theory, the water in the annular pipe is considered to be saturated at this position when the fluid temperature is equal to the saturation temperature that corresponds to the referenced pressure. The temperature of the inner tube surface and working medium are calculated using a linear interpolation algorithm. Then, the position of the saturation point is determined according to the corresponding relation between fluid pressure and temperature.

The phase-change heat transfer from water is calculated using Eq. (6).

$$Q_{\mathrm{pt}} = Q_{\mathrm{hp}} - Q_{\mathrm{sp}} \tag{6}$$

$$Q_{\mathrm{sp}} = q_{m,\mathrm{hp}} c_p \left(T_{\mathrm{sat,hp}} - T_{\mathrm{in,hp}}\right) \tag{7}$$

The average boiling convective heat transfer coefficient h is calculated from the definition in Eq. (8). $T_{\mathrm{w,ave}}$ and $T_{\mathrm{f,\ ave}}$ are the average values of the temperature of the inner tube surface and fluid at the saturation point and measuring points above the saturation point, respectively.

$$h_{\mathrm{b}} = \frac{Q_{\mathrm{pt}}}{A_{\mathrm{b}}\left(T_{\mathrm{w,ave}} - T_{\mathrm{f,ave}}\right)} \tag{8}$$

$$A_{\mathrm{b}} = \pi d_{\mathrm{o}}\left(1.45 - H_{\mathrm{sat}}\right) \tag{9}$$

To ensure the accuracy of the measured data, the uncertainty of the relevant experimental parameters is calculated using Eqs. (10) and (11) [18]. The results are shown in Table 1.

$$\eta = \frac{\sigma}{m} \tag{10}$$

$$\eta_i^2 = \sum_{i=1}^{n} \left(\frac{\sigma_i}{m_i}\right)^2 \tag{11}$$

Tab. 1. Uncertainty of experimental parameters.

Parameter	Uncertainty
Length (L)	0.1%
Pipe diameter (d)	0.31%
Temperature (T)	1.2％
Pressure (p)	2.1%
Mass flow (q_m)	2.92%
Mass flow rate (G)	3.1%
Heat exchange (Q)	7.38%
Heat transfer coefficient (h)	12.58%

4. Experimental results and analysis

4.1. Effect of pressure on boiling heat transfer

Experiments are performed using a heat-pipe working medium mass flow rate of 1.69 kg·m^{-2}·s^{-1} and initial control system pressures of 1, 5, 10, 15, 20, 30, and 50 kPa. The saturation pressure and convective heat transfer coefficient at the heat-pipe outlet are analyzed under various experimental conditions.

Fig. 3 shows the pressure (saturation) variation law at the heat pipe outlet as a function of the initial system pressure. The pressures and the fitting curve produced at 80 ℃ are above those produced at 70 ℃, signifying that the water vapor partial pressure of the working medium in the system is large and that boiling is more intense at 80 ℃. The gray, dotted line denotes that the pressure value at the outlet changes with increasing initial pressure of the system in the single-phase convective heat transfer process. The vertical distance between the experimental data and the gray, dotted line can also qualitatively reflect the boiling of water in an annular tube. The smaller the distance, the closer the system is to single-phase convective heat transfer.

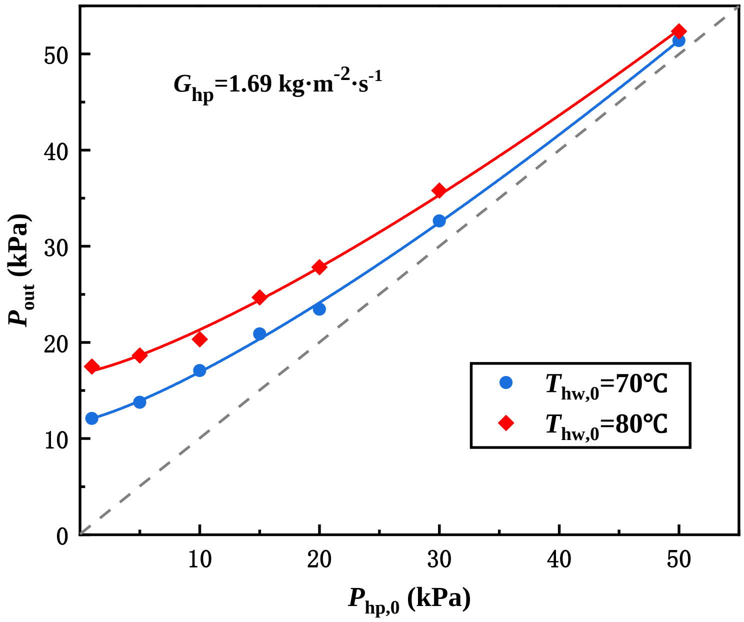

Fig. 3. Variation trend of pressure at the outlet of heat pipe.

To accurately reflect the boiling heat transfer characteristics, experimental data are recorded and analyzed. The law describing the variation in the convective heat transfer coefficient with the system pressure is summarized in Fig. 4.

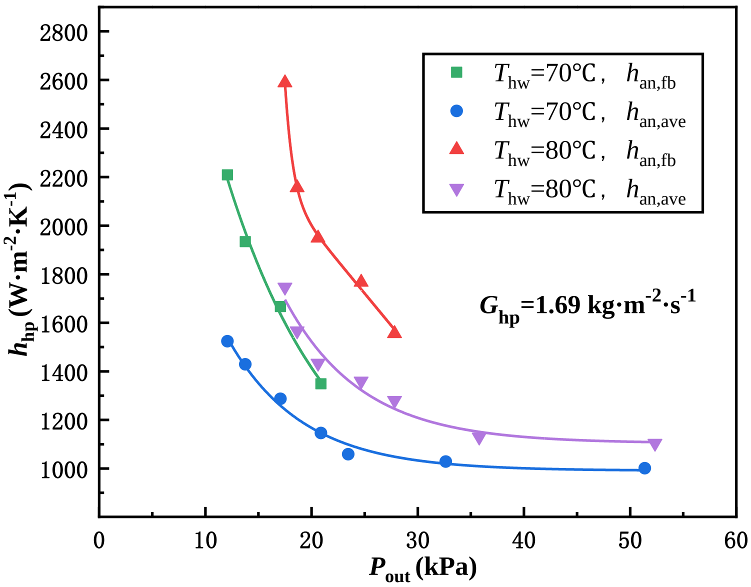

Fig. 4. Variation in the convective heat transfer coefficient.

The curves in the figure show the boiling and average convective heat transfer coefficient variation trends at 70℃ and 80℃, respectively. The lower the system pressure is, the greater the convective heat transfer coefficient is when the inlet temperature of hot water is the same. This trend is particularly obvious in the boiling section. The two working conditions of 70 ℃ and 80 ℃ produce a state of single-phase convective heat transfer when the initial pressure is at least 20 or 30 kPa, respectively. Therefore, the boiling h is not calculated under these working conditions. The blue and purple lines represent the variation law of the average h. Its value gradually decreases, and tends to 1000 and 1100 $W \cdot m^{-2} \cdot k^{-1}$. These two values are the single-phase convective heat transfer coefficients.

The enhancement ratio β is introduced to highlight the effect of boiling on the heat transfer process; β is the ratio of the boiling and single-phase convective heat transfer coefficients. When the initial pressure is 1 kPa, the flow boiling and average surface convective heat transfer enhancement ratios are 2.2 and 1.5, respectively. As the initial pressure increases, the enhancement ratio decreases and gradually tends to 1.

4.2. Effect of the hot-water temperature on boiling heat transfer

The hot-water inlet temperature has a large influence on the wall temperature, heat transfer coefficient, and extent of boiling. Fig. 5 shows the initial temperature distribution in the annular tube when the initial pressure is 1 kPa and the heat-pipe working medium flow is 12 kg/h.

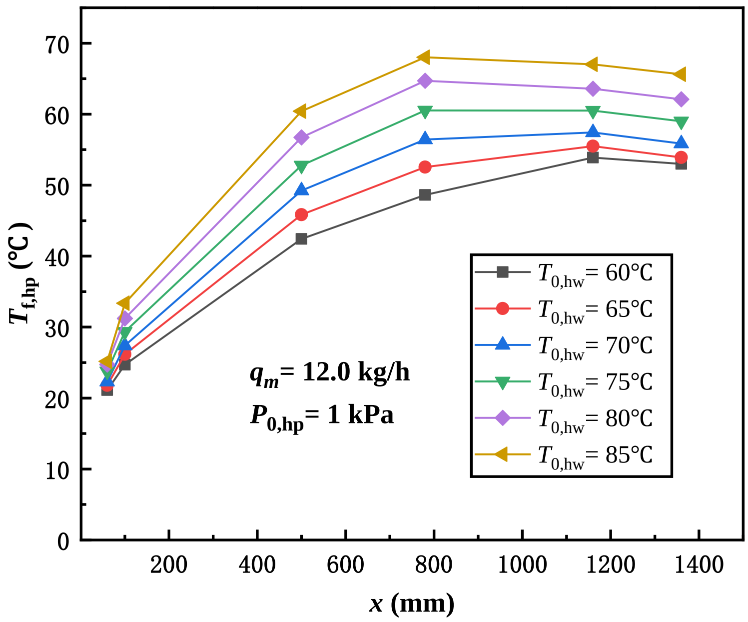

Fig. 5. Fluid temperature distribution in the pipe.

The temperature of the fluid in the annular pipe first rapidly increases and then slowly decreases. When the condensate enters the annular tube, it undergoes single-phase convective heat exchange with the outer wall of the inner tube. The large temperature difference results in a large rate of temperature increase within the hydraulic medium; consequently, the temperature rapidly increases. The test tube is in a vertical state, and the working medium in the annular tube is obviously affected by gravity. The closer it is to the top of the vertical annular tube, the lower the pressure is and the lower the corresponding saturation temperature is. Therefore, the temperature decreases slowly after saturation is reached.

Fig. 6 shows the effect of the hot-water inlet temperature on boiling heat transfer. The ordinate in this figure represents the heat exchange quantity Q_{hp} and the convective heat exchange coefficient h_{an}, and the abscissa represents the hot-water inlet temperature.

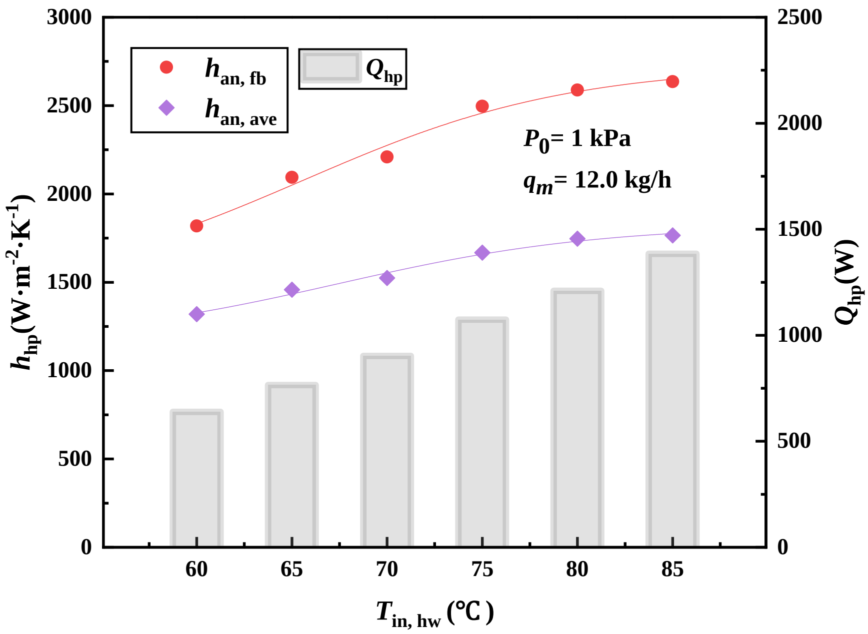

Fig. 6. Variation in Q_{hp} and h_{an} with the hot-water inlet temperature.

Heat transfer from the annular tube gradually increases with the inlet temperature. The boiling convective heat transfer coefficient $h_{an,fb}$ and the average convective heat transfer coefficient $h_{an,ave}$ also gradually increase and then tend to stabilize when the temperature reaches 80 ℃. The value of $h_{an,fb}$ gradually increases from 1800 to 2700 W·m^{-2}·k^{-1}. This occurs because the distilled water in the annular pipe is affected by the saturation pressure and its ability to become hotter is limited when the hot-water inlet temperature is too high. Thus, the temperature difference between the heating wall and boiling working medium increases, but the heat flux density increase is small. Thus, the heat transfer coefficient no longer increases significantly. This shows that the practical application process does not improve with increasing hot-water temperature at the inlet.

4.3. Boiling heat transfer flow patterns

Different wall temperatures afford different bubble positions and rates when the working medium boils in the annular tube. Thus, the flow states of the two-phase flows vary. Fig. 7 shows the physical diagrams of the boiling of the working medium during the experiment. It reflects how the two-phase working medium flow patterns change when the hot-water temperatures at the inlet are 60 ℃, 65℃, 70 ℃, 75 ℃, 80 ℃, and 85 ℃, respectively.

a) 60℃ b) 65℃ c) 70℃ d) 75℃ e) 80℃ f) 85℃

Fig. 7. Two-phase flow patterns in annular tubes.

Clearly, the extent of boiling of the working medium in the annular tube increases with the inlet temperature. Small bubbles are generated in the lower part of the annular tube and they rapidly move upward to form bubble flow at temperatures of 60 ℃ and 65 ℃. As the temperature continues to increase, a bubbly flow is generated and sometimes a slug flow can be observed. Finally, a large number of bubbles gather together, the flow irregularity increases, and a churn flow forms at the outlet. At temperatures of 80 ℃ and 85 ℃, only the churn flow is seen in the figure, but bubble and slug flows are observed at lower heights.

5. Conclusion

Based on the background of low-grade energy recovery in the production process, we identified deficiencies in research on boiling heat transfer of water in an annular tube by collecting and analyzing the relevant literature and data. Thus, heat transfer characteristics and flow patterns associated with negative-pressure water boiling in annular tubes were studied. The primary results are as follows.

a) When the initial pressure of the system increases from 1 to 50 kPa, the extent of boiling of the working medium in the annular tube decreases and the convective heat transfer coefficient gradually decreases during stable operation under various working conditions. Moreover, the boiling and average surface convective heat transfer enhancement ratios gradually tend to 1 from 2.2 and 1.5, respectively.

b) The heat transfer and convective heat transfer coefficient increase with the hot- water temperature. However, when the temperature reaches 80 ℃, the convective heat transfer coefficient varies little and tends to be stable.

c) The extent of boiling of the working medium in the loop increases with the inlet hot-water temperature. When the temperature is low, the boiling flow mainly comprises bubble and slug flows.

When the temperature is high, the water boils fully and the flow pattern in the boiling region is mainly a churn flow.

Nomenclature

Nomenclature		ρ	density, [kg·m^{-3}]
A	heat exchange area, [m^2]	σ	maximum error
c_p	specific heat capacity, [kJ·kg^{-1}·K^{-1}]	η	uncertainty, [%]
d_i	inner diameter of tube, [m]	Subscripts	
d_o	outer diameter of tube, [m]	an	annular tube
g	gravitational acceleration, [m·s^{-2}]	ave	average
H	height, [m]	b	boiling
h	convective heat transfer coefficient, [W·m^{-2}·K^{-1}]	f	fluid
K	heat transfer coefficient, [W·m^{-2}·K^{-1}]	hp	heat pipe
L	length, [m]	hw	hot water
p	pressure, [Pa]	in	inlet
Q	heat exchange capacity, [W]	ins	thermal insulation material
q	heat flux, [W·m^{-2}]	l	liquid
q_m	mass flow, [kg·s^{-1}]	out	outlet
T	temperature, [K]	pt	phase transition
$\triangle T$	temperature difference, [K]	s	loss
Greek symbol		sat	saturation
α	angle, [°]	sp	single-phase
λ	thermal conductivity, [W·m^{-1}·K^{-1}]	w	wall

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