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Brief communication

Study of boiling incipience and heat transfer enhancement in forced flow through narrow channels

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1. Introduction

Flow boiling heat transfer in narrow channels had been investigated extensively. Among early works, Ishibashi and Nishikawa (1969) clarified the space restriction effect on the saturated boiling heat transfer. Aoki et al. (1982) discussed the influence of gap width on heat transfer coefficient within narrow gaps. Peng et al. (1998) argued that boiling nucleation in microchannels is much more difficult to achieve than in conventional scale channels. Hapke et al. (2000) found that intensive oscillations of pressure and mass flux in a 1.5 mm tube lead to temperature oscillations. Recently, Ghiaasiaan and Chedester (2002) suggested that boiling incipience in microchannels is controlled by thermocapillary forces that tend to suppress microbubbles formation on wall cavities. Qu and Mudawar (2002) presented a method of measuring boiling incipience. Basing on experimental data on water boiling, Yu et al. (2002) developed an empirical correlation for boiling incipience.

In subcooled forced flow, boiling incipience represents the initiation of nucleate boiling. Investigation of boiling incipience can reveal details about nucleate boiling heat transfer. The purpose of this paper is to analyze the main factors having influence on the boiling incipience heat flux q_{ONB} and develop an empirical correlation suitable for predicting q_{ONB} in narrow annular channels within the ranges: mass flux 45–180 kg/m^{2}s, system pressure 2–3.5 MPa, temperature of inlet water 50–180 °C, and heat flux 40–210 kW/m^2. Moreover, heat transfer enhancement in narrow channels is experimentally demonstrated.

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2. Experimental facility

2.1. Experimental system

The experimental system is shown schematically in Fig. 1. Pure water was driven from a water tank by a pump with variable frequency transducer which could regulate the system pressure. The mass flux in the system was adjusted by the valve and regulator. The standard hole-plate flowmeter, calibrated to an accuracy of $\pm 0.5\%$, was employed to measure mass flux. After passing the preheater, subcooled water was led into the test section in which it passed the states of saturated water, steam and superheated vapor. The temperature and pressure of inlet flow were measured by temperature and pressure gauges, respectively. The pressure drop in two-phase flow was monitored by a CECC-650G type capacitive pressure transducer calibrated to an accuracy of $\pm 0.5\%$. After passing through the test section, the vapor flowed into a condenser in which it was condensed to liquid water.

Measured data including mass flux, pressure and temperatures along the inner and outer tubes were acquired and processed by FLUKE-2645 data acquisition system and a computer. The system experimental parameters were in the following ranges: mass flux 45–180 kg/m^{2}s, system pressure 2–3.5 MPa, temperature of inlet water 50–180 °C, and heat flux 40–210 kW/m^2.

2.2. Test section

Two coaxial tubes formed a vertical annular channel in the test section (see Fig. 2). The inner and outer tubes were made of stainless steel and heated simultaneously by a high-current-regulated DC power supply. Two test sections were used whose tubes had the following diameters and wall thicknesses: 12×1 mm, 8×1.2 mm, and 13×1.5 mm, 7×1 mm, respectively. Both test sections had the

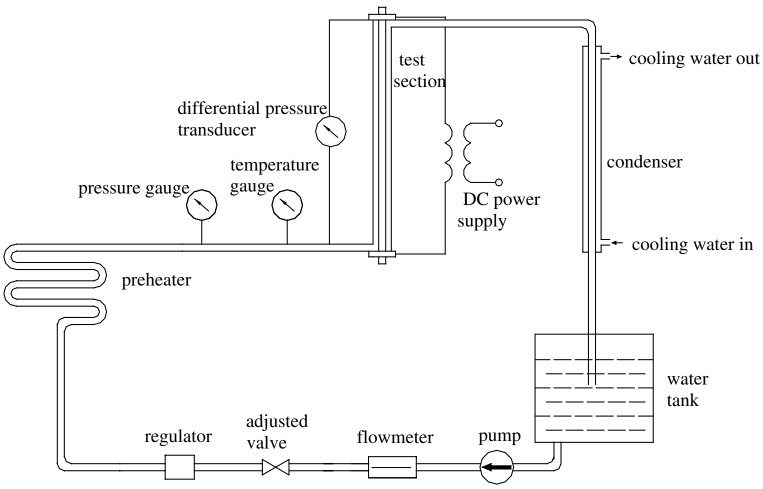

Fig. 1. Schematic diagram of the experimental system.

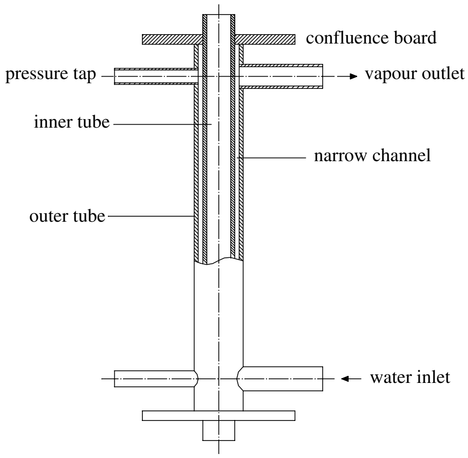

Fig. 2. Schematic diagram of the test section.

length of 1300 mm and the gaps 1 mm and 1.5 mm, respectively. Precautions were taken to achieve uniform distribution of heat flux between the two tubes in each test section. The temperature of inlet water was measured by a metal clad thermocouple (K-type, 1 mm). The inner wall temperatures of inner tube were measured by a metal clad thermocouple (K-type, 1 mm) which was drawn along the inner wall. The outer surface temperatures of the external tube were measured by chromel–alumel thermocouples (K-type, 0.3 mm) which were welded along the outer wall and reinforced by inorganic adhesives. Thermal insulating layers (30 mm thick) of slag wool with thermal conductivity 0.03–0.04 W/(m °C) were put on the outer tube of the test section, in order to minimize the heat loss. The accuracy of all thermocouples used in the experiment was ±0.2 °C.

3. Boiling incipience

3.1. The location of ONB point

The ONB point occurred at the first turning point in the curve of wall temperature that varied along the longitudinal direction of the tubes. The heat transfer coefficient increased dramatically when subcooled boiling occurred. It then resulted in a sharp drop in wall temperature for fixed heat flux as was described by Collier (1972). The typical wall temperature curve was shown in Fig. 3, where $(T_w)_{ONB}$ was the wall temperature at ONB point and q_{ONB} was the heat flux at ONB point. The ONB point could be identified as the point where the wall temperature changed noticeably before the saturated section of the wall temperature curve.

3.2. Previous correlations for onset of nucleate boiling

Some models have been proposed for boiling incipience. Han and Griffith (1965) investigated bubble initiation, growth and departure in nucleate pool boiling. The model of Davis and Anderson (1966) considered chopped-spherical bubbles and gave an expression for hemispherical

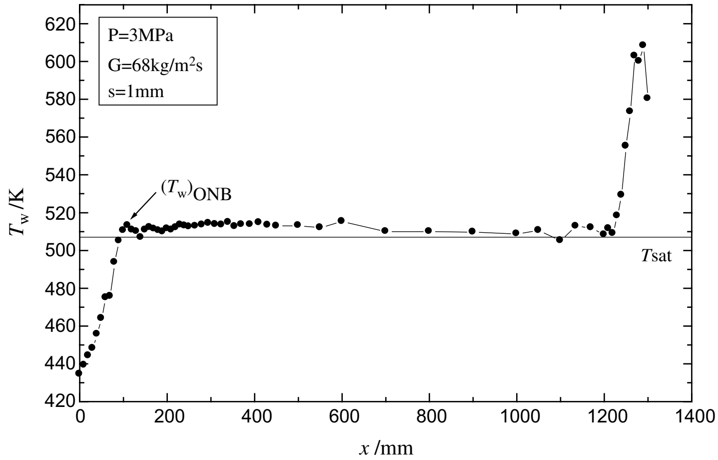

Fig. 3. Wall temperature along the tubes.

bubbles. The method of Bergles and Rohsenow (1964) was most widely used, and for water its predictions had been correlated as

$$q_{ONB} = 5.30P^{1.156}[1.8(T_w - T_{sat})_{ONB}]^{2.41/P^{0.0234}} \quad (1)$$

where q_{ONB} is the heat flux at the onset of nucleate boiling (W/m^2); P is the system pressure (kPa); T_w is the wall temperature (K); T_{sat} is the saturation temperature (K).

It should be mentioned that most of these models are based on data obtained from larger channels with diameters above 10 mm. They are not quite suitable for predicting flow boiling heat transfer in narrow channels, such as their hydraulic diameter is below 2 mm. Table 1 shows a comparison of q_{ONB} between the current experimental data and Bergles–Rohsenow correlation with $P = 2$ MPa, $s = 1.5$ mm.

3.3. Analysis of experimental data

In a heated tube, boiling does not occur when the temperature of heated surface is below the saturation temperature T_{sat} at system pressure. When the surface temperature is above T_{sat}, but the temperature of main fluid remains below T_{sat}, few bubbles develop on the wall surface and the bubbles condense when they encounter subcooled fluid. It means that the fluid flows into subcooled boiling region. According to the experimental data, at the beginning of subcooled boiling region, especially for the flow boiling heat transfer in narrow channels, the mass flux G and the gap size s have great influences on the heat flux at the onset of nucleate boiling q_{ONB} except for wall superheat ΔT_{sat}.

Table 1
Comparison of q_{ONB} (kW/m^2) between experimental data and Bergles–Rohsenow correlation

Experimental data	49.4	66.5	87.6	108.9	137.2	174.4
Bergles–Rohsenow correlation	109.1	142.7	189.8	260.8	512.2	703.8

3.4. Empirical correlation

A subcooled boiling region exists between the convection region of single-phase liquid and the saturated boiling region. The bubbles that develop from the heated wall are influenced by fluid flow, and then the departure diameter of the bubble decreases; the frequency of this process increases and the bubble's surface is deformed. Forced convection of single-phase liquid also has a strong influence on this process. Therefore, the influence of mass flux G on heat flux at the onset of nucleate boiling q_{ONB} cannot be neglected. Obviously the gap size s also has a strong influence on heat transfer of flow boiling in narrow channels. Basing on the gap size s as the characteristic dimension, Reynolds number Re can be defined as $Re = \frac{Gs}{\mu_{\mathrm{L}}}$.

The influence of mass flux G and gap size s on flow boiling heat transfer in narrow channels may be expressed by the Reynolds number. According to the above analysis, the heat flux at the onset of nucleate boiling q_{ONB} depends on the Reynolds number and the wall superheat ΔT_{sat} as

$$q_{\mathrm{ONB}} = f(Re, \Delta T_{\mathrm{sat}}) \tag{2}$$

By curve-fitting the experimental data, q_{ONB} can be expressed as

$$q_{\mathrm{ONB}} = 0.175Re^{0.92}(\Delta T_{\mathrm{sat}})^{0.11} \tag{3}$$

Fig. 4 shows a comparison between the predictions of this correlation and the experimental data with the deviation below 15%.

4. Heat transfer enhancement

Fig. 5 shows the influence of gap size s on wall superheat ΔT_{sat} for fixed pressure, mass flux and heat flux. At the ONB point, wall superheat ΔT_{sat} with $s = 1.5$ mm is smaller than that with

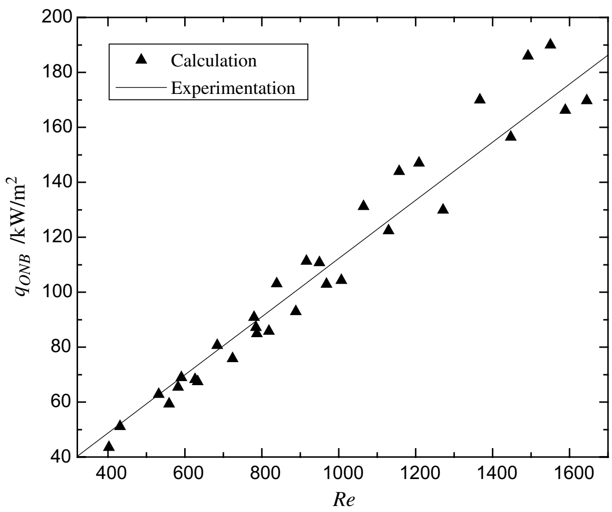

Fig. 4. Comparison of q_{ONB} between calculation and experimentation.

$s = 1$ mm, so ΔT_{sat} decreases with increasing gap size s. In other words, for the same ΔT_{sat}, the heat flux leading to subcooled boiling for $s = 1$ mm is less than that for $s = 1.5$ mm. Therefore, flow boiling in narrow channels enhances heat transfer.

The influence of gap size on local heat transfer coefficient along the axial direction of tube is shown in Fig. 6 for system pressure 3 MPa; mass flux 50 kg/m^{2}s; temperature of inlet water 50 °C and heat flux 70 kW/m^2. In the single-phase convective heat transfer region, for fixed heat flux, the temperature difference between the wall surface and fluid gradually increases. So the local heat transfer coefficient h decreases along the tubes. At ONB point, the wall temperature drops suddenly, and then the local heat transfer coefficient h increases gradually in the subcooled nucle-

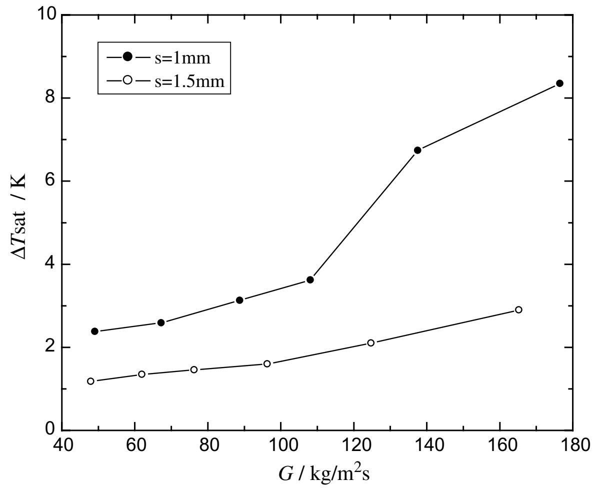

Fig. 5. The influence of gap size s on wall superheat ΔT_{sat}.

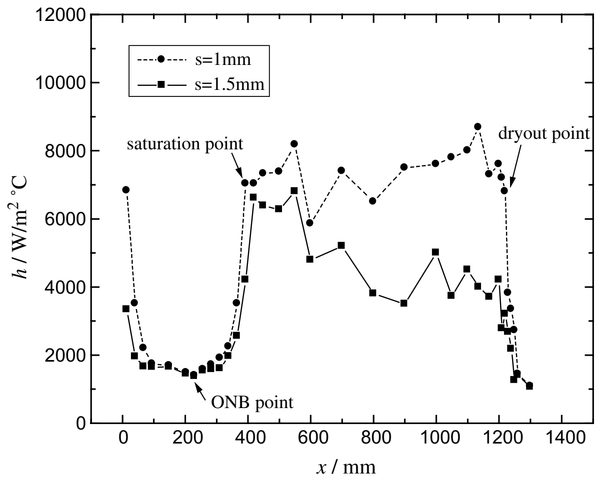

Fig. 6. The influence of gap size s on heat transfer coefficient h.

ate boiling region. In the saturated boiling region, fluid temperature remains constant, and the wall temperature only changes slightly, so the local heat transfer coefficient h only changes slightly with location as well. At the dryout point, the wall temperature increases, therefore, the local heat transfer coefficient h decreases. Fig. 6 shows that local heat transfer coefficient h, which changes along the axial direction of test section x, increases with decreasing gap size s. That means decreasing dimension of narrow channels enhances heat transfer. It may be due to the surface tension and dominant nucleate mechanism as was demonstrated by Tran et al. (1996).

5. Conclusions

The boiling incipience heat flux q_{ONB} is independent of system pressure, and increases with increasing mass flux. For fixed mass flux, q_{ONB} decreases with decreasing gap size. The empirical correlation Eq. (3) can be used to predict q_{ONB} in narrow annular channels within present experimental ranges. The deviation between the predictions of this correlation and the experimental data is below 15%.

Local heat transfer coefficient changes along axial direction of the test section, but for fixed heat flux, it increases with decreasing gap size in narrow channels. That leads to the conclusion that narrowing channels enhances flow boiling heat transfer.

Acknowledgments

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References

Aoki, S., Inoue, A., Aritomi, M., Sakamoto, Y., 1982. Experimental study on the boiling phenomena within a narrow gap. Int. J. Heat Mass Transfer 25, 985–990.

Bergles, A.E., Rohsenow, W.M., 1964. The determination of forced-convection surface boiling heat transfer. ASME J. Heat Transfer C 86, 365–372.

Collier, J.G., 1972. Convective boiling and condensation. McGraw-Hill, New York.

Davis, E.J., Anderson, G.H., 1966. The incipience of nucleate boiling in forced convection flow. AIChE J. 12, 774–780.

Ghiaasiaan, S.M., Chedester, R.C., 2002. Boiling incipience in microchannels. Int. J. Heat Mass Transfer 45, 4599–4606.

Han, C.V., Griffith, P., 1965. The mechanism of heat transfer in nucleate pool boiling-I. Bubble initiation, growth and departure. Int. J. Heat Mass Transfer 8, 887–904.

Hapke, I., Boye, H., Schmidt, J., 2000. Onset of nucleate boiling in minichannels. Int. J. Thermal Science 39, 505–513.

Ishibashi, E., Nishikawa, K., 1969. Saturated boiling heat transfer in narrow spaces. Int. J. Heat Mass Transfer 12, 863–894.

Peng, X.F., Hu, H.Y., Wang, B.X., 1998. Boiling nucleation during liquid flow in microchannels. Int. J. Heat Mass Transfer 41, 101–106.

Qu, W.L., Mudawar, I., 2002. Prediction and measurement of incipient boiling heat flux in micro-channel heat sinks. Int. J. Heat Mass Transfer 45, 3933–3945.

Tran, T.N., Wambsganss, M.W., France, D.M., 1996. Small circular and rectangular channel boiling with two refrigerants. Int. J. Multiphase Flow 22, 485–498.

Yu, W., France, D.M., Wambsganss, M.W., et al., 2002. Two-phase pressure drop, boiling heat transfer, and critical heat flux to water in a small-diameter horizontal tube. Int. J. Multiphase Flow 28, 927–941.