Experimental study on convective boiling heat transfer in narrow-gap annulus tubes

LI Bin,[1] ZHAO Jian-Fu,[1] ZHOU Fang-De,[2] TANG Ze-Mei,[1] HU Wen-Rui[1]

([1]*National Microgravity Laboratory, Institute of Mechanics, the Chinese Academy of Sciences, Beijing 100080;*

[2]*The National Laboratory of Multiphase Flow in Power Engineering, Xi'an Jiaotong University, Xi'an 710049)*

Abstract Since convective boiling or highly subcooled single-phase forced convection in micro-channels is an effective cooling mechanism with a wide range of applications, more experimental and theoretical studies are required to explain and verify the forced convection heat transfer phenomenon in narrow channels. In this experimental study, we model the convective boiling behavior of water with low latent heat substance Freon 113 (R-113), with the purpose of saving power consumption and visualizing experiments. Both heat transfer and pressure drop characteristics were measured in subcooled and saturated concentric narrow gap forced convection boiling. Data were obtained to qualitatively identify the effects of gap size, pressure, flow rate and wall superheat on boiling regimes and the transition between various regimes. Some significant differences from unconfined forced convection boiling were found, and also, the flow patterns in narrow vertical annulus tubes have been studied quantitatively.

Keywords Narrow gap annulus tube, Single phase flow, Boiling heat transfer, Two-phase flow pattern

CLC number TK124

1 Introduction

Convective boiling or highly subcooled single-phase forced convection in micro-channels is an effective cooling mechanism with a wide range of applications. Among these are the cooling of such diverse systems as accelerator targets, high power resistive magnets, compact fission reactor cores, fusion reactor blankets, advanced space thermal management systems, manufacturing and materials processing operations, and high-density multi-chip modules in supercomputers and other modular electronics. These devices involve micro-heat pipes, mini heat exchangers, micro-scale thermosyphon, oscillating capillary heat pipe, porous heat exchanger, etc. In order to improve the performance of such a heat dissipation device, the information of the heat transfer, two-phase flow and its mechanisms in the narrow flow passage is required. The fundamental knowledge, such as the flow pattern, heat transfer coefficient, pressure drop etc. is quite limited so far, although it is critical for the engineering design and practical performance. So more experimental and theoretical studies are required to explain and verify the forced convective heat transfer phenomenon in narrow channels. Several studies about boiling in narrow space have been reported, but they were not what we aimed at. A new type of heater tube, named narrow gap annulus tube(gap size is 1~2.5 mm) and used in a steam generator, is studied in this paper. Heat transfer characteristics and two-phase flow regime are also studied experimentally.

2 Experimental apparatus

A schematic diagram of the test loop is shown in Fig.1. R113 is provided by an anti-corrosion pump. A downward flowing water stream in the primary loop heated the tube wall of the annular gap. The inlet quality of R113 loop was controlled by a preheater and its rate was manually controlled by a valve between the pump and preheater. The two loops were entirely separate, their interaction is only to exchange heat in the test section. Each was heated, cooled,

Supported by the 973 Project of China (G1999022308) and the Knowledge Innovation Program of the Chinese Academy of Sciences (KJCX2-SW-L05)

Received date: 2002-12-02

pumped and metered independently.

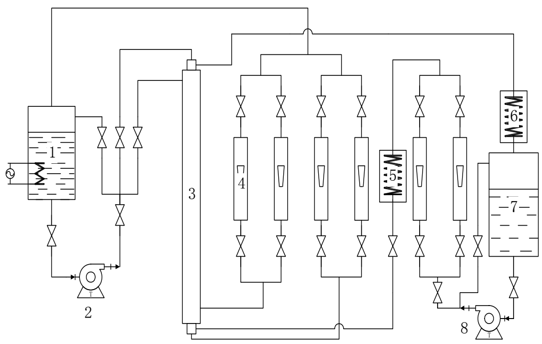


Fig.1 Schematic diagram of experimental apparatus.
1.Water tank, 2. Water pump, 3. Test section, 4. Rotary flowmeter, 5. Preheater, 6. Condensor, 7. R113 tank, 8. Anti-corrosion pump

The test section consists of outer and inner red copper tubes (Fig.2). To assure concentricity of two tubes, metal spots were welded on the outer wall of the inner tube. The same sized tube (10 mm OD and 8 mm ID) is used for all test sections. We can get different gap size through changing the ID of outer tube. For visualization test, quartz glass tubes are adopted to observe the boiling and flow phenomena. There are 8 measured sections and 16 thermocouples (0.1 mm OD) on each tube along the tube length direction, along the circumferential direction the angle between two thermocouples on each cross-section is 90 ^{o}. All thermocouples are K-type, and had a calibration accuracy of $\pm$0.2 ^{o}C. Both the inlet pressure and pressure drop between inlet and outlet of the test channel are measured by 1151 pressure transducers, which maximum uncertainty of measurement is 2.5%. Flow rates are measured by rotating flowmeters. All these flowmeters were calibrated before the experiment and their maximum uncertainty of measurement is 0.5%. Effective length of all test sections is 900 mm. Information about the heater tubes is presented in Table 1.

For a particular test section, an experimental run involved setting the R113 flow rate, pressure and inlet enthalpy, and maintaining these while varying the primary water temperature and flow rate. The ranges of the experimental parameters are:

Pressure: 1.14×10^5~1.49×10^5 Pa

R113 mass velocity: 20~1.0×10^3 kg·(m^2·s)$^{-1}$

Primary loop temperature: 45~75 ^{o}C

Primary loop mass flux: 5.6×10^{-3}~5.6×10^{-2} kg·s^{-1}

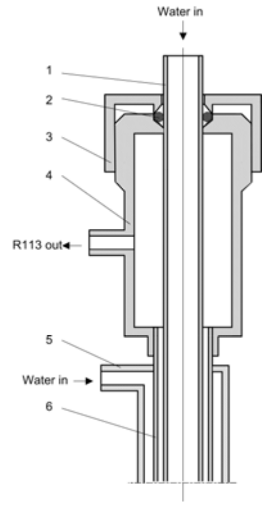


Fig.2 Schematic diagram of test section.
1. Inner tube, 2. O-shape joint ring, 3. Compressive screw cap, 4. R113 outlet cavity, 5. Primary loop passage, 6. Outer tube

Table 1 Description of the heater tubes.

Tube No.	Description of the tube	D_e (mm)
1	Visual experiment, only the inner tube heated	2.00
2	Visual experiment, only the inner tube heated	4.00
3	Heat transfer experiment, both the inner and outer tubes heated	2.00
4	Heat transfer experiment, both the inner and outer tubes heated	3.00
5	Heat transfer experiment, both the inner and outer tubes heated	4.67

The IMP-3595 Data Acquisition System was used to get all the electric data from sensors of pressure and temperature. A high speed camera was used to get the images of two-phase flow behavior pattern. The shutter speed was 1/1000 s.

3 Experimental results and discussion

The average heat transfer coefficient of annular tube is calculated as follows:

$$h_{\exp} = \frac{q}{\overline{T}_w - T_f}$$

where q is the average heat flux of inner and outer tube; T_f is the average fluid bulk temperature; $\overline{T_w}$ is the average value of two thermocouples along the circumferential direction at the same measured section:

$$\overline{T}_w = (T_{w1} + T_{w2}) / 2$$

3.1 Single-phase convection

It was desired to compare the experimentally obtained Nusselt number data with the best available correlation for the given range of Reynolds and Prandtl numbers. One of the most accurate correlations for single-phase forced convection in annular tube is the following equation (for Re<2300):[1]

$$Nu_0 = \left(Pr/Pr_w\right)^{0.11}\left\{Nu_\infty + 1 + 0.14\left(\frac{D_{oi}}{D_{io}}\right)^{0.4}\cdot\frac{0.19\left(RePr\frac{D_e}{L}\right)^{0.8}}{1+0.117\left(RePr\frac{D_e}{L}\right)^{0.467}}\right\} \quad (1)$$

where

$$Nu_\infty = 3.66 + \left[4 - \frac{0.102}{\left(D_{oi}/D_{io}\right)+0.02}\right]\left(\frac{D_{oi}}{D_{io}}\right)^{0.04}$$

As shown in Fig.3, fraction of our data falls below the lower Reynolds number limit (Re<1600) for Equation (1). When Re>1600, even though in the range of laminar flow defined for traditional circular tube, the experimental Nusselt numbers in the narrow annular tubes are generally higher than those predicted by Equation (1). Within the examined range, all experimental Nusselt numbers are smaller than those predicted by Dittus-Boelter formula:

$$Nu = 0.023Re^{0.8}Pr^{0.4}$$

The experimental data also show that inside-wall's Nusselt numbers of annular tube are higher than those of outside-wall. Compared with Equation (1), heat transfer is enhanced in the annular gap, and the extent of enhancement increases as the diameter decreases and Reynolds number increases. Maximum enhancement ratio of nearly 2.5 was obtained at Reynolds numbers of 1.6×10^3~3.5×10^3 for channel diameter of 1 mm.

A small number of investigators have looked at forced convection in small passages, starting with a study by Levy *et al*.[2] of turbulent forced convection in rectangular channels. The channels measured 2.54 mm×63.5 mm, the corresponding hydraulic diameter is 4.88 mm. Over the range of Reynolds numbers investigated (10^4~10^5), the data were nearly 30%~50% below values of the heat transfer coefficient predicted by the Seider-Tate equation. Gambill and Bundy[3] studied both momentum and heat transfer in thin rectangular channels with gap sizes ranging from 1.09 to 1.45 mm corresponding to hydraulic diameters of 1.91~2.67 mm. For Reynolds number ranging from 9×10^3 to 2.7×10^5, they found the heat transfer coefficients were slightly higher than those predicted by the Seider-Tate equation, thus calling the earlier results of Levy in question. Wang and Peng[4] investigated single-phase forced convection of water and methanol in rectangular microchannels with hydraulic diameters ranging from 0.311 to 0.747 mm. They found that the transition to turbulent flow was initiated at Reynolds numbers of 1000~1500, and their heat transfer data could be well predicted by a modification of the Dittus-Boelter equation in which the constant has been changed from 0.023 to 0.00805.

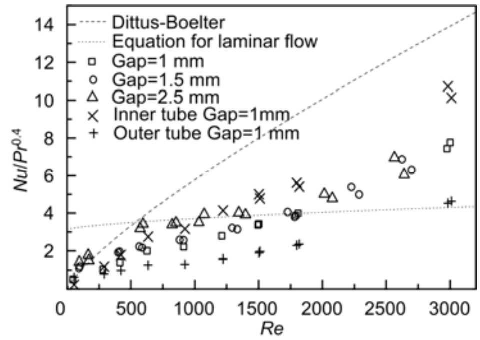


Fig.3 Single-phase forced convection characteristics in vertical narrow gap annular tubes with different gap sizes.

In the above cited literature, there is considerable disagreement as to the forced convection heat transfer characteristics in small size channels, and before this study there is no published paper focusing on fluid and heat transfer in the narrow gap annular tube. To meet the needs of design, modification of the Dittus-Boelter equations to accommodate the enhancement caused by the small diameters for different gap size annular tubes was developed from a least squares fit of the experimental data respectively:

For δ=1 mm: $Nu = 0.0031Re^{0.95}Pr^{0.4}$ (2)

For δ=1.5 mm: $Nu = 0.0187Re^{0.73}Pr^{0.4}$ (3)

For δ=2.5 mm: $Nu = 0.137Re^{0.48}Pr^{0.4}$ (4)

The range of validity of Equations (2)～(4) is Re<3500, and the mean value of the difference between the data and predicted values are less than ±9.3%.

3.2 Flow patterns of convective boiling

Even though considerable differences exist in the various researcher's definitions of two-phase flow pattern in channels having a larger size gap, it is generally agreed that flow regimes consist of bubbly flow, slug flow, churn flow and annular flow. Collier[5] gave the description of such typical flow regimes in vertical channels having a medium size gap. Photographs of flow boiling structures in the present vertical annular tube having mini-gaps of 1 mm and 2 mm have similar feature of such characteristics (Fig.4), except that a new flow pattern, coalesced bubble regime, had been observed. And also other flow regimes have little differences compared with those in traditional diameter tubes. Five different boiling regimes were identified in this study:

(a) Isolated bubble regime. Different from the bubble flow of larger tube, due to the limit of the narrow space the spherical bubble is compressed into flat crescent or elliptic one (Fig.4a). This also enhances the chaos of the two-phase flow. In this case the wall temperature is steady and close to the R113 saturation temperature.

(b) Slug flow. In slug flows, slug bubbles are relatively long and have a beautiful spherical nose (Fig.4b). Bridges of very thin liquid film are observed in a long slug bubble. In liquid slugs, very little bubbles are observed. When gap size was 2 mm slug bubbles were more easier to be observed compared with annular gap in size of 1 mm.

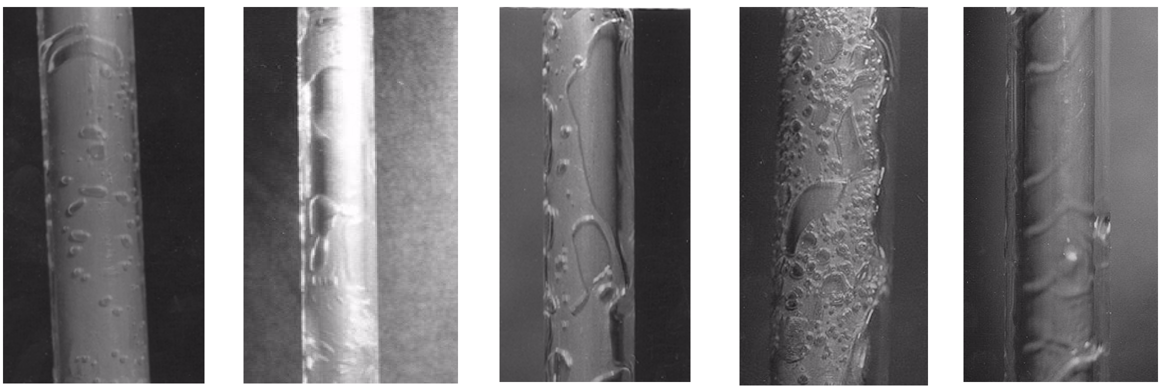
(c) Churn flow. Generally this flow pattern appeared alternately with coalesced bubble regime.

(d) Coalesced bubble regime. As the wall temperature increased a new kind of flow pattern (Fig.4d), coalesced bubble regime was observed, with large number of bubbles gathering which were swept away intermittently by the vapor.

(e) Annular flow. Fig.4e shows the fully developed annular flow pattern. Most of the liquid phase attached the tube wall formed film flowing steady to the outlet. In this case a very interesting phenomenon was observed, the liquid film flow in the term of ripples like a series of symmetrically distributed liquid rings surrounding the steam core. As the heat flux increased unsteady dry patches formed which remained on the tube surface for a relatively short time (1 sec), and then were swept away, leading to tube rewetting.

The transition to the coalesced-bubble regime depends on mass velocity, gap size, wall superheat and inlet quality. Smaller mass velocity and gap size, and larger wall superheat and inlet quality favor this transition.

Frankum[6] compared lots of experimental data with other flow-pattern maps obtained by many other researchers, and found that the empirical flow-pattern map of Fair[7] using the Lockhart-Martinelli parameter had the best performance in these types of maps. In this study we also gave the boiling two-phase flow map by using the Lockhart-Martinelli parameter, as shown in Fig.5. Compared with Fair's map, in narrow gap annulus channels, the transition lines from bubbly flow to intermittent flow and from intermittent flow to annular flow shifted to the left. The reason that transition line shifted to the left is mainly the increased friction shear stress due to the squeezed bubbles in the gap.



(a) (b) (c) (d) (e)

Fig.4 Flow patterns in narrow gap annular gap: (a) isolated bubble regime; (b) slug flow; (c) churn flow; (d) coalesced bubble regime; (e) annular flow.

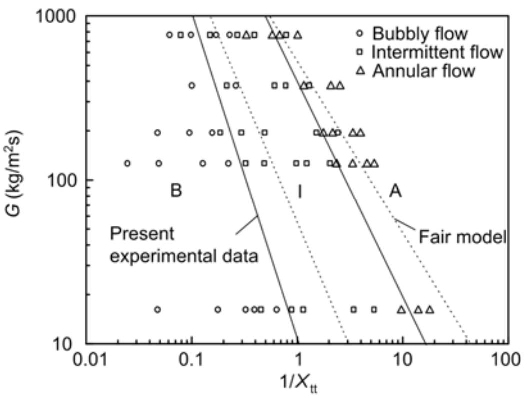


Fig.5 Flow regime transition comparison with Fair model for narrow annulus gap (gap size is 1mm).

3.3 Convective boiling heat transfer

Having compared the experimental data with the equation: $h = 0.7574q^{0.75}$, which was developed by Kutepov for calculating the heat transfer coefficient of R113 flowing in circular tube, we found that heat transfer coefficients were generally higher than those in traditional larger tube. Fig.6 shows the relation between heat flux and heat transfer coefficient for R113 boiling in annular tube of different gap size, the range of heat enhanced ratio reached from 1.7 to 2.0, and when the gap size decreased, the heat transfer coefficient increased.

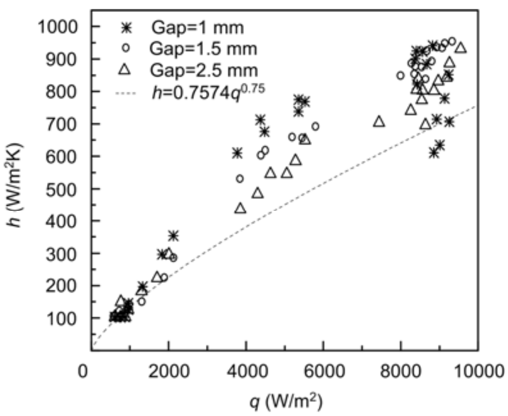


Fig.6 Boiling heat transfer characteristics in annular gap.

From the visualization test we considered the thinning of liquid film due to the limit of narrow space between compressed steam bubble and tube wall was the main reason of heat transfer enhancement in annular gap. We know that the liquid film is the dominated heat conduction resistance during heat exchange between primary loop and secondary loop. Otherwise, the density of vaporization points per volume was increased when heating both sides of the annular gap, this also enhanced the boiling heat transfer process.

In order to meet the engineering design needs, we expressed our boiling heat transfer coefficient in the following form:[5]

$$\frac{h_{tp}}{h_{fo}} = f\left(\frac{1}{X_{tt}}\right)^n$$

where X_{tt} is Martinelli number:

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_g}\right)^{0.1}$$

Thus the equations for computing boiling heat transfer coefficient were developed, and the mean value of the difference between the experimental data and predicted values are less than ±11.2%.

For δ =1 mm, $h_{tp}/h_{fo} = 3.257(1/X_{tt})^{0.37}$

For δ =1.5 mm, $h_{tp}/h_{fo} = 8.154(1/X_{tt})^{0.39}$

For δ =2.5 mm, $h_{tp}/h_{fo} = 7.277(1/X_{tt})^{0.19}$

4 Conclusions

We can conclude this paper as follows.

(1) For single-phase forced convection, the enhanced heat transfer may be caused by turbulent flow, although the range of Reynolds number is less than 2300. Experimental data show that Nusselt number of inner tube wall of the annular gap is higher than that of the outer tube wall and the difference between the inside tube wall and outside tube wall becomes larger as the gap size is smaller.

(2) We observed five flow patterns of R113 boiling in the annulus gap, and two flow regimes, coalesced bubble regime and annulus flow with ripples, are different from the flow pattern in traditional larger diameter tubes. And also because of the increased friction shear stress due to the squeezed bubbles in the gap, the transition lines from bubbly flow to intermittent flow and from intermittent flow to annular flow shifted to the left.

(3) Boiling heat transfer is enhanced in narrow gap annular tube. We consider that the reason may be the deformed bubble, which decreases the thickness of the liquid film between the steam bubble and heated tube wall, thus the resistance of heat transfer is decreased. And also the disturbance of two-phase flow is enhanced in the annular gap.

(4) From the experimental data, we developed

equations both for single-phase flow and flow boiling in the narrow gap annular tube. This can be the reference for heat exchanger design.

Nomenclature and subscript

C_p: specific heat, J/kg·K
D_{oi}: outer diameter of inner tube, m
D_e: equivalent diameter, m
D_{io}: inner diameter of outer tube, m
h: heat transfer coefficient, W/m^2·K
N: number of thermocouples location
Nu: Nusselt number
Pr: Prandtl number
q_I: heat flux of inner tube, W/m^2
q_o: heat flux of outer tube, W/m^2
Re: Reynolds number
$\overline{T}_w$: mean temperature of the tube wall, K
T_f: average fluid bulk temperature, K
W_{wi}: water mass flow rate of inner tube, kg/s
W_{wo}: water mass flow rate of outer tube, kg/s
X_{tt}: Martinelli number
ρ: density, kg/m^3
μ: viscosity, Pa·s
δ: gap size, m
cal: computational value
exp: experimental value
g: gas
l: liquid
sp: single phase
fo: liquid only flow
tp: two phase

References

1 Bohn, Th., Knoche, Thermal power engineering computational atlas. Translated by Xu Z Y, Li Y Q, (in Chinese) WCEP Pub, Beijing, 1991

2 Levy S, Fuller R, Niemi R. J Heat Transfer, 1959, **1**: 129-143

3 Gambill W, Bundy R. HFIR heat transfer studies of turbulent water flow in thin rectangular channels. ORNL-3079 UC-80-Reactor Technology. Oak Ridge National Laboratory, Oak Ridge, TN, 1961

4 Wang B, Peng X. Int J Heat Mass Transfer, 1994, **37**: 73-82

5 Collier. Convective boiling and condensation, McGraw-Hill Inc, England, 1981

6 Frankum D P, Wadekar V V, Azzopardi B J. Exp Thern Fluid Sci, 1997, **15**: 183-192

7 Fair J R. Petroleum Refiner, 1960, **39**(2): 105-123