Available online at www.sciencedirect.com
ScienceDirect

NUCLEAR SCIENCE AND TECHNIQUES

Nuclear Science and Techniques, Vol.18, No.4 (2007) 247–251

Experimental research on flow instability in vertical narrow annuli

WU Geping[1] QIU Suizheng[2] SU Guanghui[2] JIA Dounan[2]

([1] School of Energy and Power Engineering, Jiangsu University, Zhenjiang 212013, China;

[2] School of Energy and Power Engineering, Xi'an Jiaotong University, Xi'an 710049, China)

Abstract A narrow annular test section of 1.5mm gap and 1800mm length was designed and manufactured, with good tightness and insulation. Experiments were carried out to investigate characteristics of flow instability of forced-convection in vertical narrow annuli. Using distilled water as work fluid, the experiments were conducted at pressures of 1.0~3.0MPa, mass flow rates of 3.0~25kg/h, heating power of 3.0~ 6.5kW and inlet fluid temperature of 20 °C, 40 °C or 60°C. It was found that flow instability occured with fixed inlet condition and heating power when mass flow rate was below a special value. Effects of inlet subcooling, system pressure and mass flow rate on the system behavior were studied and the instability region was given.

Key words Flow instability, Experimental research, Vertical narrow annuli, Forced-convection

CLC numbers TL33, O359

1 Introduction

Narrow annular channels have been widely used in many industries because of its compact configuration and high heat transfer ability per volume. However, two-phase flow instabilities may occur in some conditions. Oscillations of the flow rate and system pressure can cause mechanical vibrations, high pressures, and more requirements for system control. In extreme circumstances, they would disturb the heat transfer characteristics to such an extent that the heat transfer surface may burn out.

The flow instabilities have been studied experimentally and theoretically for four decades. The experimental studies of vertical or horizontal tube flow boiling systems were performed, with most of the researches [1-3] being concentrated on either a single tube channel or parallel tube channels. On the other hand, the theoretical researches used frequency-domain codes [4] or time-domain codes[5]. To the authors' knowledge, however, no investigation on flow instabilities in narrow annular channels in forced convection has been reported.

In this paper, we report an experimental investigation of flow instabilities in narrow annular channels of 1800 mm in length and 1.5mm in gap size. Using distilled water as work fluid, the experiment was conducted at pressure of 1.0~3.0MPa, in a mass flow rate of 3.0~25kg/h, with heating power of 3.0~6.5kW and at inlet temperature of 20 °C, 40 °C or 60°C. Steady state characteristics at different inlet temperatures, pressures and heating powers were observed in the pressure drop versus mass flow rate diagrams. Dynamic instabilities are found and the effects of inlet subcooling, system pressure and mass flow rate on flow instability were investigated and stability boundary for the appearance of oscillations is given. The experiment results agree well results of a calculation based on the assumption of homogenous two-phase flow and thermodynamic equilibrium of the phases.

2 Experimental arrangements

Fig.1 is a schematic diagram of the boiling up-flow system apparatus used in the experiments. It is a closed-loop system consisting of a shield pump, a

E-mail: gpwu@mailst.xjtu.edu.cn
Received date: 2007-01-20

pressurizer, a calibrated flowmeter, a pre-heater and a condenser. Distilled water is pumped through the loop. The pressure is controlled by high-pressure nitrogen via pressure regulator. The fluid flows through the pre-heater, in which it is raised to a desired level for a given test. A calibrated flowmeter is arranged to measure the mass flow rate. The fluid enters annuli gap of the test section, shown schematically in Fig.2. The outside and inside tubes of the test section are heated with AC independently by a power supply. Heating of the pre-heater and test section is evaluated by measuring the voltage drop and current. Electrical insulation between the pre-heater and test section is provided by glass fibers and polyvinyl gasket, designated as ISO in Fig.1. The outlet fluid of the test section is condensed in the cooler. Finally, the fluid leaves the condenser and returns to pump, closing the system.

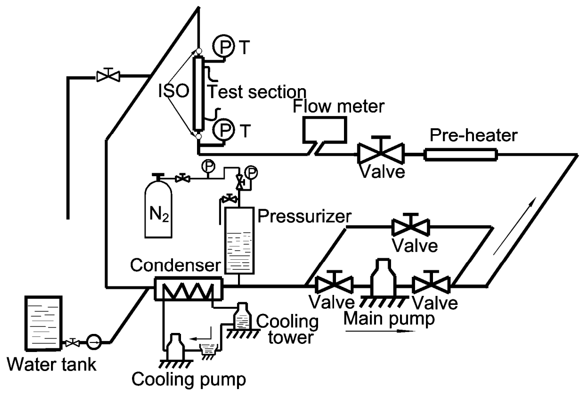

Fig.1 Schematic of the loop.

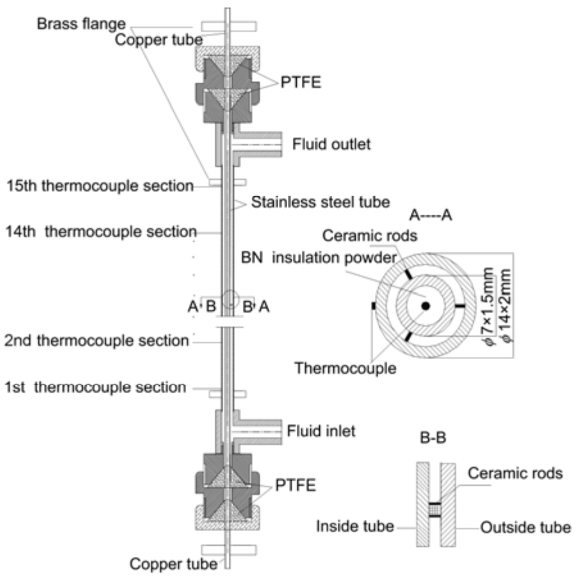

Fig.2 Schematic of the test section.

The test section is made of specially processed straight stainless steel tubes with a linearity error less than 0.01 % to form narrow concentric annuli. The test section is 1800 mm in length. Inner diameter of the outside tube is 10mm, and outer diameter of the inside tube is 7mm. Thus, the gap width is 1.5 mm. In order to ensure the concentricity and the electrical insulation between the outside and inside tubes, three Φ2.0 mm ×1.3 mm ceramic rods are arranged at the same cross section with equal angular interval at the middle of test section as shown in Fig.2. To minimize heat losses, the whole test section is covered by silicon-aluminium glass fibers of 120 mm in thickness and a guard heater is built inside the silicon-aluminium glass fiber.

Temperature of the outside tube was measured by Φ0.5mm thermocouples, which were spot-welded on the surface of the 15 cross-sections, in an interval of 120 mm along the axial direction. For measuring inner wall temperature of the inside tube, 15 thermocouples (Φ0.5mm) bundled up by thin flexible mica were inserted into the inside tube at the same arrangement as the outside tube. The space between the inside tube and thermocouples was filled with BN powder. The fluid temperature at the inlet and outlet was measured with K-type thermocouples. The system pressure was measured with pressure transducer.

A data acquisition system consisting of one PC and three Isolated Measurement Pod data loggers was assembled to record output data from pressure sensors and thermocouples.

As calibrated by authoritative institutes, uncertainties of the measurements were estimated at ±2% for the flow rate, ±0.25% for the absolute pressure, ±0.75% for the liquid temperature, ±1.5% for the wall temperatures (due to the spot-welding), and ±0.2% for the voltage and current. Dimensions of the channel were determined with an accuracy of ±1%. The heat loss from the test section is estimated at < 3%.

In performing the flow instabilities tests, the mass flow rate was gradually decreased, while the inlet fluid temperature, system pressure, and heating power to the test section were fixed. It was assumed that it would take 30 min for the system to reflect the change after the new mass flow rate was set. Then, temperatures of the walls and fluid, mass flow rate and pressure drop versus time were recorded. The system's stability was judged according to the parameters versus time. Mass flow rate was decreased until dryout occurred in the test section. The experiment was conducted by changing the parameters successively. The experimental parameter ranges are listed in Table 1.

Table 1 The range of experimental parameters

S.P./MPa	H. P./kW	I. T. /℃	M. F. R. /kg·h^{-1}
1.5~2.5	3.5~5.5	20,40,60	3.0~25.0

S.P. system pressure; H.P. heating power;
I.T. inlet temperature; M.F.R. mass flow rate

3 Experimental results

3.1 Steady state characteristics

Steady state characteristics are presented in pressure drop versus mass flow rate diagrams (Fig.3). The pressure drop decreased with increasing heating power, but increased with increasing pressure and inlet subcooling. The pressure drop of the test section decreased monotonously with the mass flow rate. Inflexion did not appear in the two phase region. This is because the ratio of gravity pressure drop to the whole pressure drop is big in vertical configuration of the test section, and the heating power is not high.

3.2 A typical flow instability

It was observed that the loop full of single phase fluid was stable. Flow instabilities occur with decreasing mass flow rate. There are the first type density-wave oscillations (low quality flow instability) and the second type density-wave oscillations (high quality flow instability). The former occurs when the exit quality of the test section slightly exceeds zero, and has higher frequency, shorter periods and smaller amplitude of vibration. The latter occurs at high exit quality with lower mass flow rate. Fig.4 shows a typical flow instability of the latter. With system pressure of 1.3 MPa, mass flow rate of 8.1 kg/h, inlet fluid subcooling at 53℃, and heating power of the inner and outer tubes of 1.3 kW and 3.3 kW respectively, the exit quality is 0.7676. This is due to a combined contribution of the pressure drop oscillations and density-wave oscillations at high exit quality. Amplitude of the vibration is variable. With increasing frequency, the phase of mass flow rate vibration is obviously different from that of wall temperature oscillations (pressure drop oscillations).

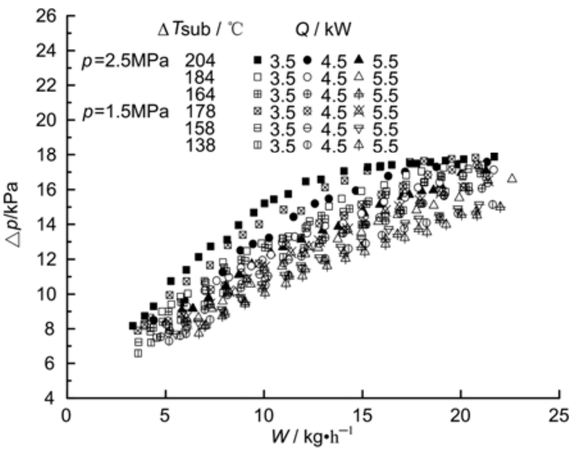

Fig.3 Curves of mass flux vs pressure drop.

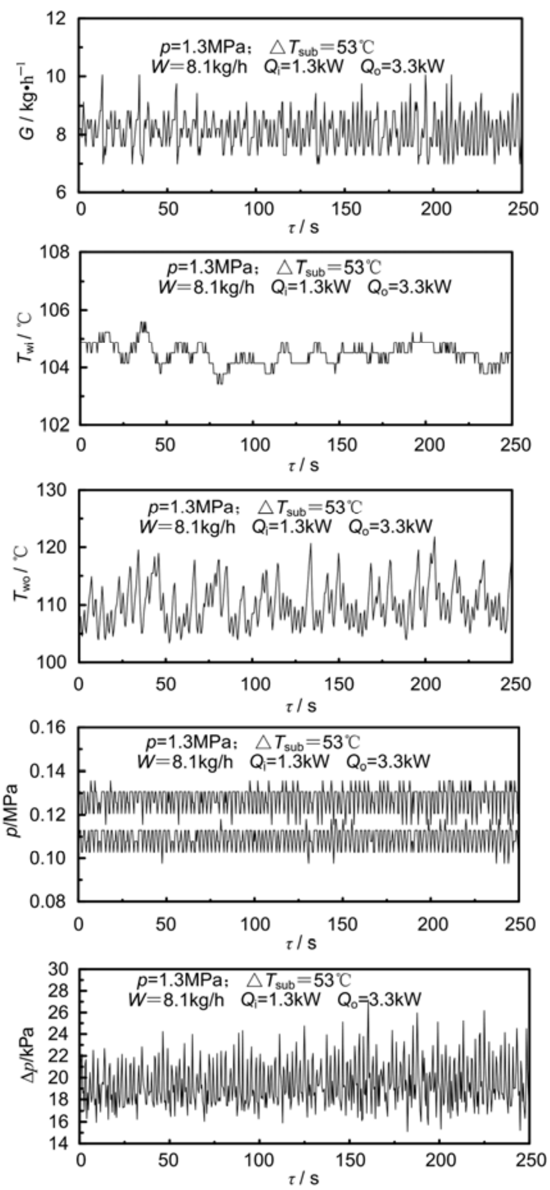

Fig.4 A typical flow instability.

3.3 Influence of various parameters on instability

3.3.1 Effect of inlet subcooling

Effect of inlet subcooling on flow instabilities is complicated [6-8]. The experiment data as a function of inlet subcooling is shown in Fig.5. With system pressure of 1.5MPa and heating power of 5.5kW, the critical mass flow rate was 6.00, 7.89 and 9.03 kg/h at 178℃, 158℃ and 138℃ of inlet subcooling, respectively. This means that the critical mass flow rate increased and system stability enhanced with decreasing inlet subcooling. However, with system pressure of 2.5MPa and heating power of 3.5kW, the critical mass flow rate was 5.27, 6.14 and 5.79 kg/h at 204℃, 184℃ and 164℃ of inlet subcooling, respectively. This indicates that the critical mass flow rate changed nonlinearly with inlet subcooling.

This may have two reasons. On one hand, the length of the single-phase increases with inlet subcooling, which is an additional inlet restriction to pressure drop, hence beneficial to system's stability. On the other hand, at the same heat input, average equality decreases and evaporation time becomes longer with increasing inlet subcooling. The time for inlet mass flow rate corresponding to pressure drop leading by evaporation becomes shorter, which enhances oscillations. The influence depends on the combined effects of the two aspects.

Generally speaking, the system stability reduces monotonously with increasing inlet subcooling in some conditions. However, its influence is nonlinear in other conditions.

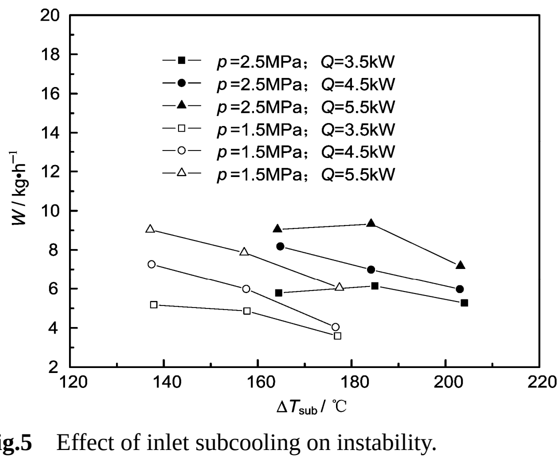

Fig.5 Effect of inlet subcooling on instability.

3.3.2 Effect of mass flow rate

Fig.6(a) shows critical heating power against mass flow rate. The critical heating power increased with the mass flow rate, which is benefit to system's stability. Apparently, with increasing mass flow rate, it becomes more difficult for bubbles to gather, and cooling capacity of the pipe wall is thus enhanced. Another conclusion can be drawn from Fig.6(b) that critical exit quality decreased with increasing mass flow rate.

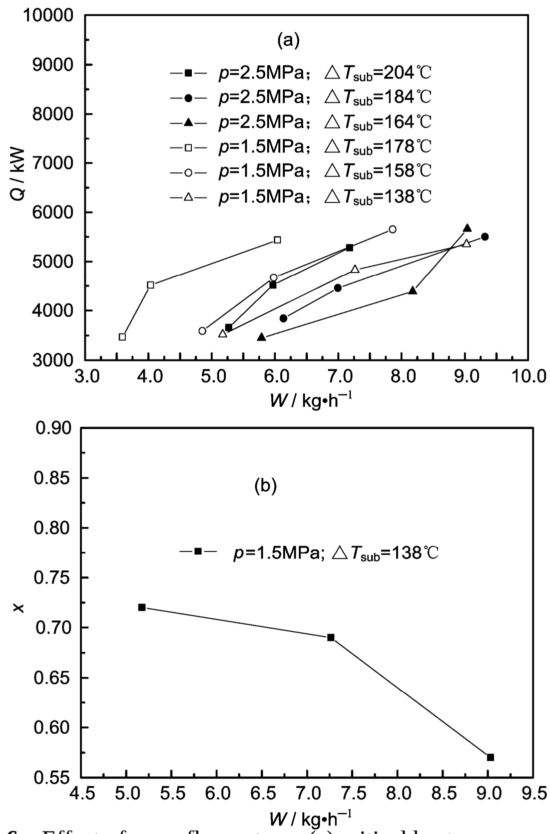

Fig.6 Effect of mass flow rate on (a) critical heat power and (b) critical exit quality.

3.3.3 Effect of pressure

It was observed from the experiment that pressure affected the flow instability, and increasing pressure improved the system's stability. The density difference between the single phase and two phase decreased with increasing pressure. At the same conditions, gravity drop perturbation in high pressure is smaller than that in low pressure with equivalent heat input. It is more difficult to generate sustained oscillations in high pressure than in low pressure. For maintaining sustained oscillations, the heating power must be higher.

3.4 Instability region

The thermal-hydraulic instability of a two-phase forced-convection circulation in narrow annular channels is a very complex multivariable and non-linear system. Many factors, such as mass flow rate, heat power, and loop structure, interact on each other. In this paper, dimensionless phase-change (N_{pch}) and subcooling number (N_{sub}) that synthetically consider influencing factors are adopted to describe instability region. It is obvious in Fig.7 that instability region is a slender region. It can be given as follows:

$$\begin{cases} N_{sub} = 0.4375 N_{pch} - 2.25 \\ N_{sub} = 0.1389 N_{pch} + 13.33 \end{cases} \tag{1}$$

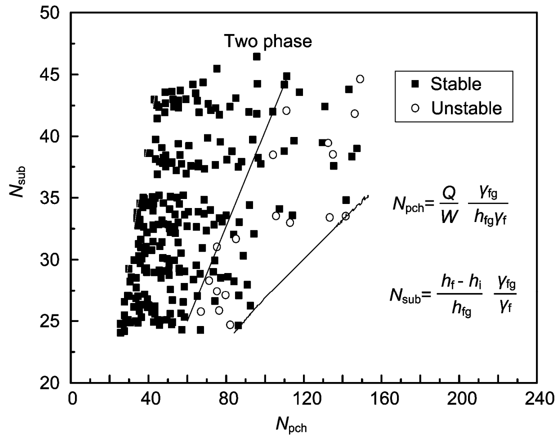

Fig.7 Instability region described with subcooling number (N_{sub}) and phase-change (N_{pch}).

It is essential, therefore, that the system be avoided to operate in this region.

3.5 Verification of calculation results

Fig.8 shows results of the calculation based on the assumption of homogenous two-phase flow and thermodynamic equilibrium of the phases against experimental data.

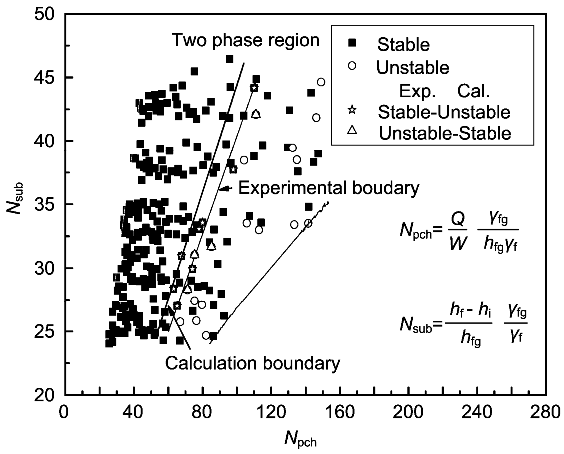

Fig.8 Comparison of experiment data with calculation results.

In general, the agreement is very good. The calculation boundary shifts to the left and there is a difference comparison against experimental boundary. We used general tube formula instead, as there is no formula of exact heat transfer and resistance coefficient for narrow annular channels.

4 Conclusions

Experimental investigation of two phase flow instability in narrow annular channels of 1800 mm in length and 1.5 mm in gap size was conducted and effects of inlet subcooling, system pressure and mass flow rate were studied. The following conclusions can be drawn.

(1) Inlet subcooling has a complicated effect on the system stability. In some conditions, the system stability reduces monotonously with increasing inlet subcooling. In other conditions, its influence is nonlinear.

(2) High mass flow rate is accompanied by high critical heat power and low critical exit quality.

(3) An increase in pressure is of benefit to the system stability.

(4) Flow instability region described with subcooling number (N_{sub}) and phase-change (N_{pch}) is a slender region.

References

1 Kakac S, Vezirglu T N, Padki M M, *et al*. Exp Thermal Fluid Science, 1990, **3**: 191-201.

2 Xu B, Chen X J. Int J Heat Mass Transfer, 1993, **3**: 497-503.

3 Comakli O, Karsli S, Yilmaz M. Energy Conversion and Management, 2002, **43**: 249-268.

4 Podowaki M Z, Rosa M P. Nuclear Engineering and Design.1997, **193**: 179-188.

5 Su Guanghui, Jia Dounan, Kenji Fukuda, *et al*. Nuclear Engineering and Design, 2002, **215**: 187-198.

6 Neal L G, Zivi S M. The mechanisms of hydrodynamic instabilities in boiling systems. Two-Phase Flow Dynamics. Symp, The Netherlands. 1968, 61-73.

7 Zuber N, Findlay J A. J Heat Transfer, Trans, ASME, 1965, **85**: 453-568.

8 Veziroglu T N, Lee S S, Kakac S. Two-Phase Flows and Heat Transfer, 1997, **1**: 423-466.