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International Journal of Heat and Mass Transfer 46 (2003) 4779–4788

International Journal of HEAT and MASS TRANSFER

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Convective boiling heat transfer and two-phase flow characteristics inside a small horizontal helically coiled tubing once-through steam generator

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Received 15 February 2003; received in revised form 8 June 2003

Abstract

The pressure drop and boiling heat transfer characteristics of steam-water two-phase flow were studied in a small horizontal helically coiled tubing once-through steam generator. The generator was constructed of a 9-mm ID 1Cr18Ni9Ti stainless steel tube with 292-mm coil diameter and 30-mm pitch. Experiments were performed in a range of steam qualities up to 0.95, system pressure 0.5–3.5 MPa, mass flux 236–943 kg/m^{2}s and heat flux 0–900 kW/m^2. A new two-phase frictional pressure drop correlation was obtained from the experimental data using Chisholm's B-coefficient method. The boiling heat transfer was found to be dependent on both of mass flux and heat flux. This implies that both the nucleation mechanism and the convection mechanism have the same importance to forced convective boiling heat transfer in a small horizontal helically coiled tube over the full range of steam qualities (pre-critical heat flux qualities of 0.1–0.9), which is different from the situations in larger helically coiled tube where the convection mechanism dominates at qualities typically >0.1. Traditional single parameter Lockhart–Martinelli type correlations failed to satisfactorily correlate present experimental data, and in this paper a new flow boiling heat transfer correlation was proposed to better correlate the experimental data.

Keywords: Steam-water two-phase flow; Boiling heat transfer; Frictional pressure drop; Helically coiled tube

1. Introduction

Helically coiled tubes are extensively used in steam generators, refrigerators, nuclear reactors and chemical plants, etc., due to their practical importance of high efficiency heat transfer, compactness in structure, ease of manufacture and arrangement. Horizontal helically coiled tubing once-through steam generator is favored for space, navigation and other specific techniques because of its lower gravitation center and higher efficiency both in heat transfer and steam generation [1,2]. For practical design and application of those steam generators, the prediction of two-phase flow and boiling heat transfer characteristics is extremely important. The investigation of two-phase flow and boiling heat transfer characteristics in helically coiled tubes is highly lacked, compared to the investigation that in straight channels.

Although different flow and heat transfer characteristics exist, the methods for analyzing the pressure drop and heat transfer for the straight tube are still used or modified to describe forced convective boiling two-phase flow and heat transfer in helically coiled tube. Owhadi et al. [3] carried out a pioneering research on forced convective boiling heat transfer to water at atmospheric pressure in two helically coiled tubes of 12.5 mm ID, and $d/D = 0.05$, 0.024, respectively. Their results show that over most of the quality region, the prevailing heat transfer mode is convection and a nucleate boiling component is present at low qualities. They found that the local average boiling heat transfer coefficient of coil could be predicted by Chen's correlation [4] with

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doi:10.1016/S0017-9310(03)00354-5

Nomenclature

A	flow channel section area (m^2)
Bo	boiling number (dimensionless, q/Gi_{lg})
c_P	specific heat ($J\,kg^{-1}\,K^{-1}$)
D	coil diameter (m)
d	tube inner diameter (m)
E	electric voltage input (V)
f	frictional factor (dimensionless)
G	mass flow rate ($kg\,m^{-2}\,s^{-1}$)
h	heat transfer coefficient ($W\,m^{-2}\,K^{-1}$)
I	electric current through test section (A)
i_{lg}	latent heat of evaporation ($J\,kg^{-1}$)
k	thermal conductivity ($W\,m^{-1}\,K^{-1}$)
L_H	heated length (m)
Nu	Nusselt number (dimensionless, hd/k)
P	pressure (Pa)
Pr	Prandtl number (dimensionless, $c_P\mu/k$)
q	heat flux ($W\,m^{-2}$)
q_E	electric power input ($W\,m^{-2}$)
q_T	input heat flux calculated from enthalpy increase of fluid over heated length ($W\,m^{-2}$)
Re	Reynolds number based on inside diameter of tube (dimensionless, Gd/μ)
T	temperature (K)
x	equilibrium mass quality (dimensionless)
z	distance along channel (m)

Greek symbols

ΔP	total pressure drop (Pa)
ΔP_a	acceleration component of total pressure drop (Pa)
ΔP_f	frictional component of total pressure drop (Pa)
ΔP_g	gravity component of total pressure drop (Pa)
ΔT_{sat}	difference between the wall temperature and the saturation temperature ($T_w - T_{sat}$) (K)
η	heat loss factor (dimensionless)
μ	dynamic viscosity ($Pa\,s^{-1}$)
ρ	density ($kg\,m^{-3}$)
σ	surface tension ($N\,m^{-1}$)
ϕ^2_{lo}	two-phase multiplier (dimensionless)
χ_{tt}	Martinelli parameter for the particular case where the liquid and vapour phases are turbulent (dimensionless)

Subscripts

c	coils
cr	critical
in	test section inlet
l	liquid properties, or corresponding to liquid portion flowing alone in the channel
lo	corresponding to entire flow as liquid
g	gas properties
sat	saturation
sc	subcooled
TP	two-phase condition
tt	corresponding to turbulent liquid and vapour flow
w	tube wall
out	test section outlet
z	at location z along channel length

Over bar

—	average

accuracy of ±15% over the range tested. Their frictional pressure drop data were correlated by Martinelli parameter and the results indicated that the Lockhart–Martinelli correlation could be used to estimating the frictional pressure drop in steam-water two-phase flow in coils. Kozeki et al. [5] conducted a test on heat transfer and pressure drop characteristics in helically coiled tube heated by high temperature water at steam pressures of 0.5–2.1 MPa. They found their two-phase frictional pressure drop data were larger than those predicted by Martinelli–Nelson's correlation for straight tube and the differences increase with the decrease of system pressure and the increase of the flow rate, and two-phase forced convection occupied the most portions due to the effect of centrifugal force and secondary flow. Kozeki and most of the later researchers ([1,6,7]) correlated their experimental results of heat transfer coefficients using Martinelli type relationship in the two-phase forced convective region. Nariai et al. [8] conducted an investigation of thermal-hydraulic behavior in an once-through steam generator used for integrated type nuclear reactor, in which the helically coiled tube was heated with liquid sodium. Their experimental result indicates that modified Kozeki's and Martinelli–Nelson's correlations agree with their experimental results of two-phase frictional pressure drop within 30% and the effect of coiled tube on average heat transfer coefficients is small, Schrock–Grossman's correlation could also be applied to coiled tube with good accuracy at the pressures of lower than 3.5 MPa. Schrock–Grossman's correlation covered the effect of both saturated nucleate boiling and forced convection and commonly used for straight tube. Unal et al. [9] conducted the same experiment in a sodium-heated helically coiled tube and found that the diameter ratio d/D has little influence on the two-phase frictional pressure drop. This conclusion was confirmed by later researches of Zhou [1] and Guo [6] employing electrically heated helically coiled tube test section. Based on their own experimental data of pressure drop, Unal [9], Zhou [1], Guo et al. [10] and Bi et al. [11] provided a series of empirical correlations developed from Lockhart–Martinelli turbulent relationship to calculate the steam-water two-phase frictional pressure drop inside vertical or horizontal coils at high pressure. Under the similar conditions to present work, the work of Guo et al. [12] indicated that the two-phase frictional pressure drop inside helically coiled tube could be calculated by Chen's correlation [13] for straight tube with a minor modification, which included the effect of the inclined angle of coil and the pressure.

Despite considerable progress was made on the prediction of two-phase flow and boiling heat transfer characteristics inside helically coiled tube in the past several decades, and many different correlations were proposed. So far, the empirical or semi-empirical correlations of different researchers can only be used in specific range and there are some conflicts among their conclusions. Many researches indicate that the two-phase flow and boiling heat transfer characteristics in small channel are different from that in large channel [14,15]. Their results indicate that the boiling heat transfer coefficient is sensitive to both heat flux and mass flux, and that convective boiling dominates at lower wall superheat values and nucleate boiling dominates at higher wall superheat values. Up to now, most research of two-phase flow and boiling heat transfer characteristics inside helically coiled tube were aimed at large diameter coils (inner diameter > 12 mm). It is the purpose of present investigations to experimentally investigate the effect of miniaturization on the two-phase flow and boiling heat transfer characteristics inside a horizontal helically coiled tubing once-through steam generator constructed of small diameter tube.

2. Experimental apparatus and test section

A closed-cycle test loop of steam-water two-phase flow for present investigation is schematically illustrated in Fig. 1. It consists of the following components, a centrifugal pump to supply power for the fluid flow, a surge tank connected to a high pressure nitrogen bottle to maintain and control the system pressure, a series of orifice meters to measure water mass flow rate, a test section and two pre-heat sections, a water cooled condenser and a water tank. The resistance of the tube wall of test section and pre-heater section was used to uniformly heat the working fluid with altering electrical current delivering total power of 200 kW. The test section was made of a 1380 mm long 1Cr18Ni9Ti stainless steel tube of $\varnothing 12 \times 1.5$ mm with the coil diameter of 292 mm and the pitch of 30 mm. The test section was well insulated thermally from the atmosphere to minimize heat loss to the environment. However, test-section heat loss calibration tests were performed, and the slight heat loss was subsequently incorporated into the data reduction.

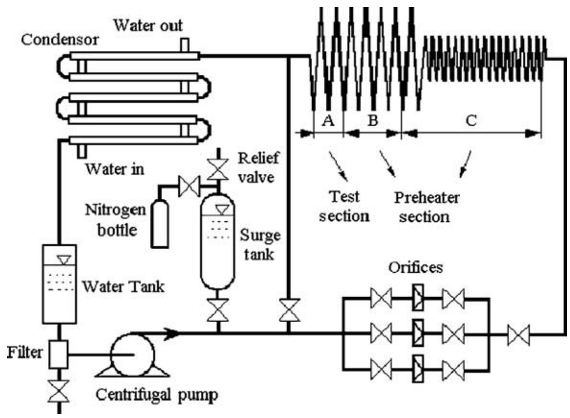

Fig. 1. Schematic diagram of the test loop.

The mass flux of working fluid through the test section was measured by using three orifice meters in different ranges (in-house construction with standard specification) appending to three 1151-DP type differential pressure transducers. These orifice flow meters were calibrated with the weighting method. The uncertainty was estimated to be less than 4% [16].

A manometer and a differential pressure transducer were used to measure the pressure at the outlet and the pressure drop of test section, respectively. The experimental uncertainty in pressure and pressure drop measurement was ±2.5% [16].

Four armored K-type thermocouples were installed into the core of tube to measure the bulk temperature of the fluid at inlet and outlet of test section and two preheaters. A total of 36 K-type thermocouples were welded to the outside surface of the tube at 9 thermocouples stations along the tube axis and electrically insulated so that the effect of heating electrical current on it was avoided. Thermocouples stations were arranged in 1/8 turns or 115 mm apart from between two stations along the flow direction, the first station was located at 240 mm from test section inlet. There were four thermocouples at each station. The location of thermocouples around the tube cross-section is shown in Fig. 2. It was estimated that the uncertainty in temperature measurement was ±3% [16].

Total electrical power supplied to test section and pre-heat sections was calculated from the measured voltage and current through the section respectively. The estimated uncertainty was ±4% [16].

All of the signals of the mass flow rate, pressure, temperature of the tube wall and the fluid, and the input

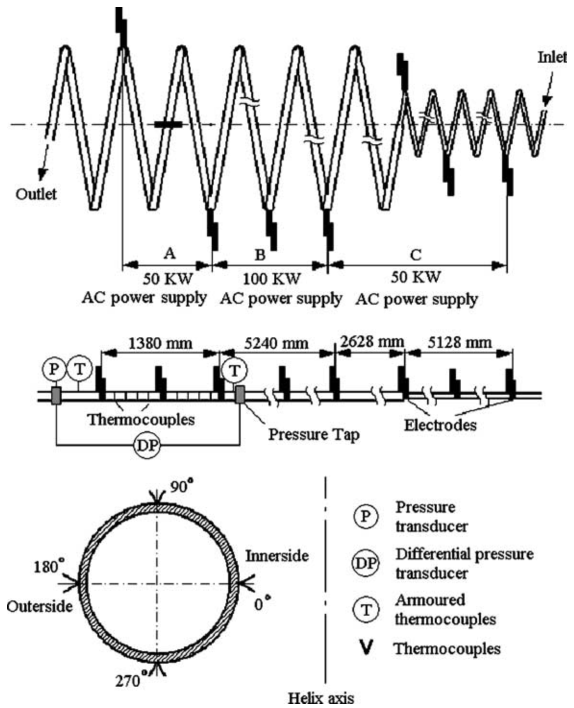

Fig. 2. Schematic diagram of the test section.

heating powers of test section and pre-heat sections were monitored and recorded via a IMP (isolated measurement pod) data acquisition system for future processing.

3. Data reduction

3.1. Single-phase heat transfer and frictional pressure drop

In order to validate the experimental apparatus, a series of tests was performed to compare the heat loss, single-phase heat transfer, and single-phase pressure drop at mass fluxes ranging from 390 to 1900 kg/m^{2}s and system pressures of 0.75 and 1.5 MPa, covering a Reynolds number range from 9700 to 86,500.

According to the transition criterion between laminar and turbulent flow in a curved pipe or helical coiled tube by Ito [17]

$$Re_{\mathrm{cr}} = 20,000\left(\frac{d}{D}\right)^{0.32} \tag{1}$$

and by Srinivansan et al. [18]

$$Re_{\mathrm{cr}} = 2100\left[1 + 12\left(\frac{d}{D}\right)^{0.5}\right] \tag{2}$$

the critical Reynolds number for present coils is 6568 and 6524, respectively. Thus the whole data of our single-phase tests fall into the turbulent region.

For the purpose of correlation, only the circumferential average heat transfer coefficients were considered. The local average heat transfer coefficient of cross-section with single-phase fluid at position z along the length of the tube is defined as

$$h_z = \frac{q_{\mathrm{T}}}{\overline{T}_{\mathrm{Wi}} - T_z} \tag{3}$$

where, $\overline{T}_{\mathrm{Wi}}$ is the cross-sectional average inner wall temperature of tube.

In Eq. (3), the input heat flux q_{T} is obtained from the enthalpy increase of the fluid over the heated length as

$$q_{\mathrm{T}} = \frac{(GAc_{\mathrm{p}})(T_{\mathrm{out}} - T_{\mathrm{in}})}{\pi d L_{\mathrm{H}}} \tag{4}$$

The corresponding bulk fluid temperature T_z at each axial position z is determined by interpolation, assuming a linear temperature gradient in the bulk fluid over the heated length.

For a given mass velocity, outside-wall temperatures were obtained at nine axial positions along the length of the test section, as indicated in Fig. 2. A numerical program to solve the two-dimensional inverse heat conduction problem developed by Bai et al. [19] with the least-squares method based on the following assumptions were introduced to calculate the inner wall temperature:

(1) The longitudinal heat conduction was small and can be neglected.
(2) The interior heat source was uniformly distributed.
(3) The thickness of tube wall was taken to be a constant.

Heat losses from the test section, including end losses, under flow conditions was determined as the ratio of the heat flux determined from the liquid enthalpy change (Eq. (4)) divided by the heat flux calculated from the electric power input,

$$\eta = \frac{q_{\mathrm{T}}}{q_{\mathrm{E}}} \tag{5}$$

where

$$q_{\mathrm{E}} = \frac{EI}{\pi D L_{\mathrm{H}}} \tag{6}$$

The heat loss factor η was determined as a function of wall temperature from single-phase tests, and it was subsequently used to determine the input heat flux for flow boiling experiments. (η was about 0.93 for most of the tests.)

For single-phase turbulent flow, Fanning frictional factor is calculated by following equation,

$$f_c = \frac{\Delta P_c \rho}{2G^2} \frac{d}{L_c} \tag{7}$$

3.2. Forced convective boiling heat transfer and two-phase frictional pressure drop

Flow boiling tests were performed at selected values of mass velocity of 400, 550 and 700 kg/m^{2}s. Pre-heater B and C were used to alter the inlet temperature and mass quality. The saturation pressure at the outlet ranged from 0.75 to 3.0 MPa. The electric power to the test section was set in a particular test to achieve a desired outlet quality or to maintain a prescribed heat flux. The experimental heat flux ranged from 0 to 900 kW/m^2.

In the data analysis, thermal equilibrium of the vapor and liquid phases was assumed along the entire length of the test section. The length of the subcooled inlet region was determined by iteration from the equation

$$L_{sc} = \frac{GAc_P(T_{sat} - T_{in})}{\pi d q_T} \tag{8}$$

and the saturation temperature T_{sat} at the start of bulk boiling was determined by the pressure calculated from the pressure-drop of single-phase subcooled liquid flow. The fluid exited the test section with a quality of <0.95 in all tests, and the saturation pressure at the exit from the heated length was calculated from the bulk fluid temperature measured there. Linear interpolation was used to determine the fluid saturation pressure at each measurement location along the test section, after which the fluid saturation temperatures were determined. The total test section pressure drop was small (generally) so that this pressure linearization introduces very little temperature error.

With knowledge of the input heat flux, inside wall temperatures and corresponding bulk fluid temperatures, local heat transfer coefficients were calculated from Eq. (3). The mass qualities x at measurement locations z were calculated from heat balances based on q_T as follows:

$$x(z) = \frac{\pi d q_T}{AGi_{fg}}(z - L_{SC}) \tag{9}$$

The following form is commonly used to correlate experimental results of forced convective boiling heat transfer coefficients in helically coiled tube,

$$\frac{h_{TP}}{h_{lo}} = f\left(\frac{1}{\chi_{tt}}\right) \tag{10}$$

where h_{TP} is the value of the two-phase heat transfer coefficient and h_{lo} is the heat transfer coefficient for the total flow, assumed to be liquid, and χ_{tt} is the Martinelli parameter, defined as

$$\chi_{tt} = \frac{(dp/dz)_L}{(dp/dz)_G} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_g}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_g}\right)^{0.1} \tag{11}$$

The experimental two-phase flow frictional pressure drop is calculated with the following formulae:

$$\Delta P_f = \Delta P - \Delta P_g - \Delta P_a \tag{12}$$

where, ΔP is the total pressure drop, ΔP_g and ΔP_a are, respectively, static and momentum pressure drop components, which can be calculated according to the method of Guo et al. [10,20]

Generally, the two-phase frictional multiplier is employed to correlate the frictional pressure drop of two-phase flow. Its definition is

$$\phi_{lo}^2 = \frac{\Delta P_{TP}}{\Delta P_{lo}} \tag{13}$$

where ΔP_{TP} is the two-phase flow frictional pressure drop and ΔP_{lo} is the single-phase frictional pressure drop at the same mass flux when the fluid is entirely liquid.

4. Experimental results and analysis

4.1. Check test

To check the suitability of the above experimental system and procedure for the present investigation, the single-phase pressure drop and single-phase heat transfer data were firstly compared with the well know correlations from White [22] and Seban and McLaughlin [23] in Figs. 3 and 4, respectively. White correlation and Seban–McLaughlin correlation are

$$f_c = 0.08Re^{-1/4} + 0.012(d/D)^{1/2} \tag{14}$$

and

$$Nu = 0.023Re^{0.8}Pr^{0.4}[Re(d/D)^2]^{0.05} \quad (6000 < Re < 65,600) \tag{15}$$

respectively. Where the liquid properties were evaluated at a film temperature defined as the average of the bulk temperature and the circumferential average temperature of the wall.

The comparison indicates that present experimental results of single-phase pressure drop are in good agreement (all data were in ±4%) with White's correlation and the results of single-phase heat transfer are in good agreement (all data were in ±10%) with Seban–McLaughlin's correlation.

These good agreements between single-phase pressure drop and single-phase heat transfer experimental

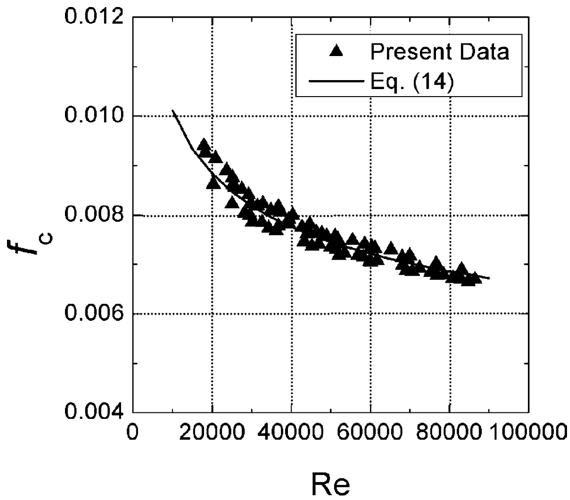

Fig. 3. Comparison of experimental single-phase frictional pressure drop results with White's correlation.

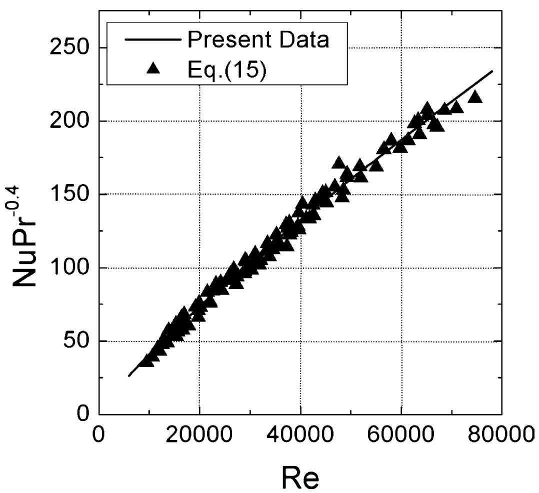

Fig. 4. Comparison of experimental single-phase heat transfer results with Seban–McLaughlin's correlation.

results with classical correlations served to establish the validity of the measurements and data-reduction method.

4.2. Boiling heat transfer coefficient

The Chen's correlation [4] widely used to predict flow boiling heat transfer coefficient in straight tube is the first to use the superposition principle of nucleate and convection dominated heat transfer. Owhadi et al. [3] indicated that Chen's correlation could also be used in helically coiled tubes with good accuracy. All the data of the present study are compared with this correlation as

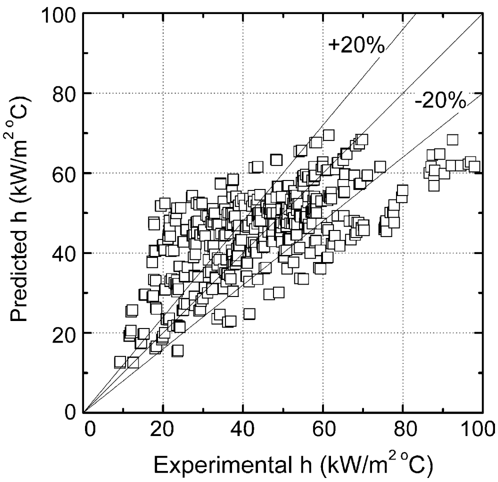

Fig. 5. Comparison of experimental boiling heat transfer results with Chen's correlation.

shown in Fig. 5. With a mean deviation of 23.4%, there is considerable scatter in the plot, but the predictions are well centered in the data. The Chen's correlation can be expressed as

$$h = 0.00122\frac{k_l^{0.79}C_{pl}^{0.45}\rho_l^{0.49}}{\sigma^{0.5}\mu_L^{0.29}i_{lg}^{0.24}\rho_g^{0.24}}\Delta T_{sat}^{0.24}\Delta p_{sat}^{0.75}S + 0.023Re_l^{0.8}Pr_l^{0.4}\frac{k_l}{d}F \tag{16}$$

S factor is calculated as

$$S = \frac{1}{1+2.53\times 10^{-6}F^{1.25}Re_l} \tag{17}$$

and F factor is calculated as

$$F = \begin{cases} 1.0 & 1/\chi_{tt} \leqslant 0.1 \\ 2.35(1/\chi_{tt}+0.213)^{0.736} & 1/\chi_{tt} > 0.1 \end{cases} \tag{18}$$

Forced convective boiling heat transfer coefficient in helically coiled tubes was correlated as a function of the Lockhart–Martinelli parameter by most of the researchers in this field [1,3,5–7], some of them adding a nucleate boiling term [8].

Fig. 6 show the comparisons of present experimental results of boiling heat transfer coefficients with the predicted values of Kozeki's correlation and Schrock–Grossman's correlation. Kozeki's test range closed to present study, but their results are similar to the Chen's correlation of Fig. 5, where the predictions are centered in the data; however, there is considerable scatter, the mean deviation is about 24%. Schrock–Grossman's correlation is one of the most famous correlation for predicting heat transfer coefficients at annular forced convective boiling flow regime in straight tubes and was suggested by Nariai et al. [8] for predicting boiling heat

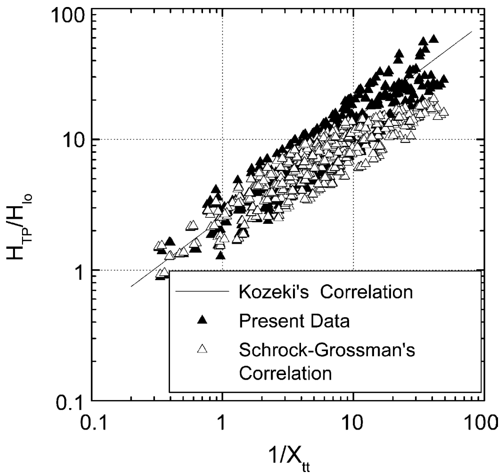

Fig. 6. Comparison of experimental boiling heat transfer results with Kozeki's and Schrock–Grossman's correlation.

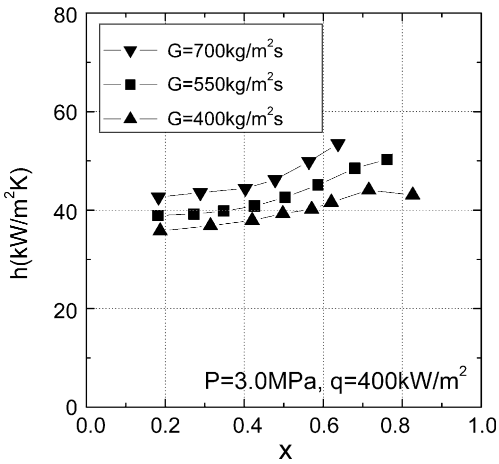

Fig. 7. The effect of mass flow rate on boiling heat transfer coefficient.

transfer coefficient in helically coiled tube. Although the mean deviation is good (about 20%), the Schrock–Grossman's correlation were mostly underpredicted the data. Kozeki's correlation and Schrock–Grossman's correlation are expressed as

$$\frac{h_{\mathrm{TP}}}{h_{\mathrm{lo}}} = 2.5\left(\frac{1}{\chi_{\mathrm{tt}}}\right)^{0.75} \tag{19}$$

and

$$\frac{h_{\mathrm{TP}}}{h_{\mathrm{lo}}} = 1.11\left(\frac{1}{\chi_{\mathrm{tt}}}\right)^{0.66} + 7400Bo \tag{20}$$

respectively.

In order to evaluate the applicability of existing correlations and/or develop a new correlation for boiling heat transfer, it is required to know whether nucleate boiling or convective boiling dominates in a particular quality range. Fig. 7 shows the variations of the measured heat transfer coefficient with the average mass quality x at three mass flow rates for $P_{\mathrm{out}} = 3.0$ MPa at $q = 400$ kW m^{-2}. It can be seen that the heat transfer coefficient increases with the increase of mass flow rate at a given mass quality. At a given mass flow rate the heat transfer coefficient increases with the increase of the mass quality and the raise in higher mass quality range is much quicker than that in lower mass quality range. Fig. 8 presents the variations of the measured heat transfer coefficients with the average mass quality x at seven different heat flux levels for $P_{\mathrm{out}} = 3.0$ MPa at $G = 400$ kg m^{-2} s^{-1}. The results indicate that at a given mass quality the heat transfer coefficient increases with the increase of heat flux. At a given heat flux the heat transfer coefficient increases with the increase of mass quality when the mass quality is low and decreases with the increase of mass quality in high mass quality range as the mass quality exceeds a critical value. Same trend can also be found under other system pressure and mass flow rates. Apparently both the nucleation mechanism and the convection mechanism have the same importance to forced convective boiling heat transfer in small horizontal helically coiled tube over the full range of qualities (pre-critical heat flux qualities of 0.1–0.9), it is different from the situations in larger helically coiled tube where the convection mechanism dominates at qualities typically >0.1. Many researchers have

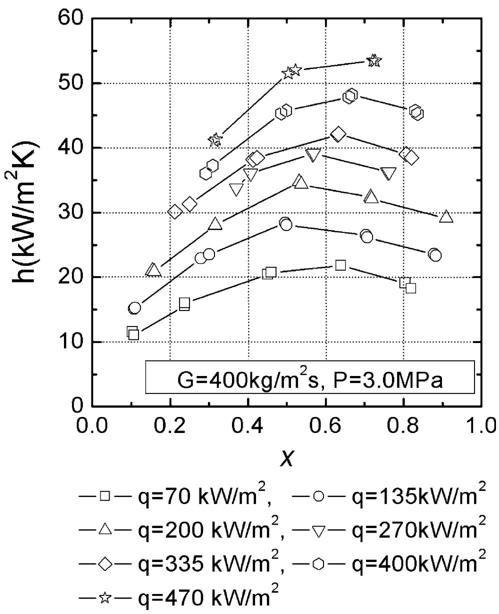

Fig. 8. The effect of heat flux on boiling heat transfer coefficient.

suggested that forced convective boiling heat transfer may be considered to be composed of a nucleate boiling component and a convective component, each of which may be correlated separately. Although this concept has been successfully applied to some problems, in general it is not clear how these two components should be combined, since the two mechanisms affect each other. Schrock–Grossman's correlation can better predict the present data, probably that is because it considers the effect of nucleate boiling mechanism properly through introducing the Boiling number, Bo. Following the approach of Schrock–Grossman, a new correlation for the boiling heat transfer coefficient was proposed as follows:

$$\frac{h_{\mathrm{TP}}}{h_{\mathrm{lo}}} = 1.6\left(\frac{1}{\chi_{\mathrm{tt}}}\right)^{0.74} + 183000Bo^{1.46} \tag{21}$$

where the Boiling number is given as:

$$Bo = \frac{q}{Gi_{\mathrm{lg}}} \tag{22}$$

Fig. 9 shows the experimental data and the predicted values obtained with Eq. (21). The predictions of Eq. (21) are in good agreement with the present experimental data, and the mean deviation is 12%.

4.3. Frictional pressure drop of steam-water two-phase flow

Fig. 10 shows the effect of system pressure on two-phase frictional multiplier ϕ^2_{lo}. The increase in system pressure remarkably decreases the frictional pressure drop, especially in low-pressure conditions. The two-phase frictional multiplier increases significantly with mass quality in the range of $x < 0.3$, and appears a slightly smoother relation in the range of $x > 0.3$.

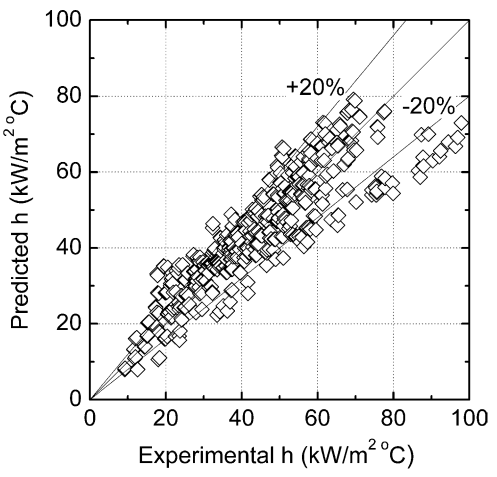

Fig. 9. Comparison of experimental boiling heat transfer results with present correlation.

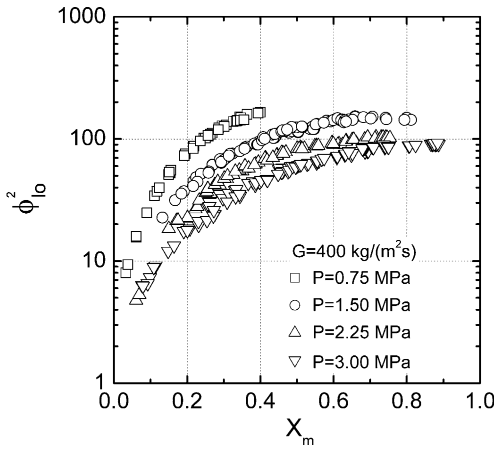

Fig. 10. The effect of system pressure on two-phase frictional multiplier ϕ^2_{lo}.

The two-phase frictional multiplier is independent of the mass flow rate in M–N method, but the effect of mass flow rate upon it cannot be ignored for present case. It can be seen from Fig. 11 that the two-phase frictional multiplier increase with the increase of mass flow rate at a system pressure of 3.0 MPa. The same phenomena also exist under other system pressures.

The effect of heat flux on two-phase frictional multiplier is shown in Fig. 12. It indicates that the effect of heat flux is not obviously and can be ignored in the present study.

Based on the Chisholm's B-coefficient method [21], a multivariate linear regression analysis was made using all of the present experimental data and a new correlation was obtained as follows:

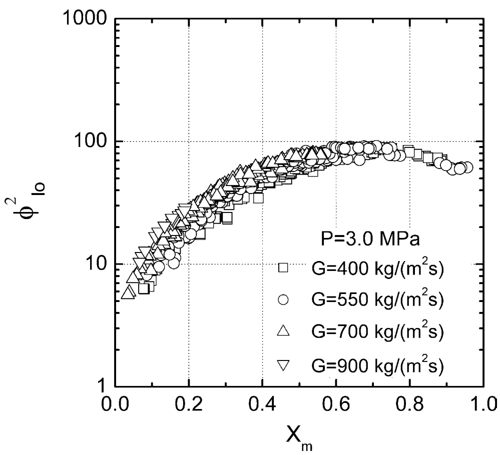

Fig. 11. The effect of mass flow rate on two-phase frictional multiplier ϕ^2_{lo}.

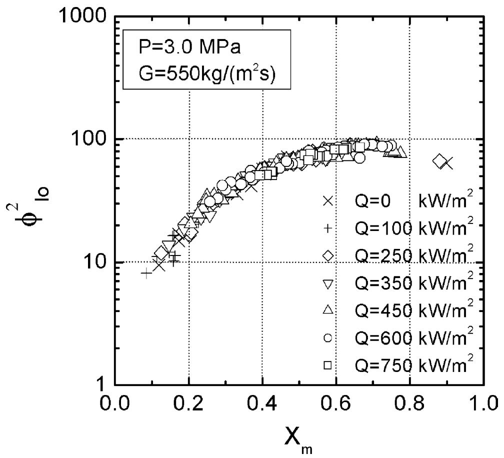

Fig. 12. The effect of heat flux on two-phase frictional multiplier ϕ_{lo}^2.

$$\phi_{lo}^2 = 1 + \left(\frac{\rho_l}{\rho_g} - 1\right)[0.303x^{1.63}(1-x)^{0.885}Re_{lo}^{0.282} + x^2] \tag{23}$$

The comparison between the experimental values of ϕ_{lo}^2 and the predicted results is shown in Fig. 13. Most of the experimental data are within a deviation of ±15%, so Eq. (23) can be used to calculate the frictional pressure drop with high accuracy in the present test range.

Three correlations are used to compare with Eq. (23) as shown in Fig. 14. Bi's correlation [11] is expressed as

$$\phi_{lo}^2 = 1 + \left(\frac{\rho_l}{\rho_g} - 1\right)(C + x^2) \tag{24}$$

$$C = 0.14691x^{1.3297}(1-x)^{0.59884}(d/D)^{-1.2864} \tag{25}$$

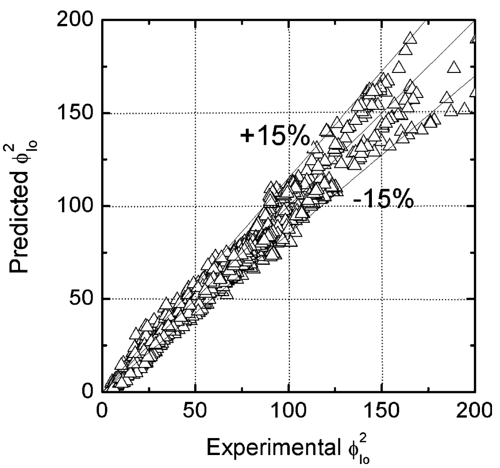

Fig. 13. Comparison of Eq. (23) with present experimental results.

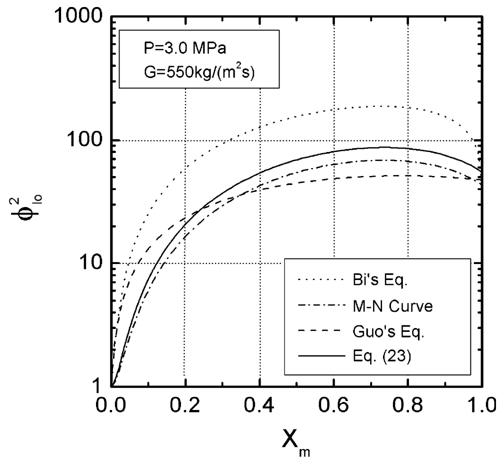

Fig. 14. Comparison of Eq. (23) with other correlations.

Guo's correlation is expressed as [12]

$$\phi_{lo}^2 = \psi_1\psi\left[1 + x\left(\frac{\rho_l}{\rho_g} - 1\right)\right] \tag{26}$$

$$\psi = 1 + \frac{x(1-x)(1000/G - 1)(\rho_l/\rho_g)}{1 + x(\rho_l/\rho_g - 1)} \quad \text{for } G \leqslant 1000 \tag{27a}$$

$$\psi = 1 + \frac{x(1-x)(1000/G - 1)(\rho_l/\rho_g)}{1 + (1-x)(\rho_l/\rho_g - 1)} \quad \text{for } G > 1000 \tag{27b}$$

$$\psi_1 = 142.2\left(\frac{P}{P_{cr}}\right)^{0.62}\left(\frac{d}{D}\right)^{1.04} \tag{28}$$

It indicates that the present correlation is closer to M–N curve and Guo's correlation (Eqs. (26)–(28)), the value predicted by Bi (Eqs. (24) and (25)) is higher than that by present correlation evidently.

5. Conclusions

In the present paper, a series of experiments of forced convective boiling heat transfer with water and the pressure drop of steam-water two-phase flow inside a small horizontal helically coiled tubing once-through steam generator were conducted.

(1) The results of single-phase turbulent flow and heat transfer show that present experimental setup is reliable, White's and Seban–McLaughlin's correlations can be used to predict single-phase frictional pressure drop and heat transfer coefficient inside small helically coiled tube with good accuracy.

(2) Both the nucleation and the convection have the same importance to forced convective boiling heat transfer in small horizontal helically coiled tube over the full range of qualities (pre-critical heat flux qualities of 0.1–0.9) which is different from the situations in larger helically coiled tube where the convection mechanism dominates at typically qualities > 0.1.
(3) The frictional pressure drop multiplier of two-phase flow is a function not only of the mass quality and the pressure, but also of the mass flow rate. The heat flux does not have obvious effect on the frictional pressure drop multiplier.
(4) Some new correlations for the forced convective boiling heat transfer coefficients and frictional pressure drop multiplier of two-phase flow inside small helically coiled tube were proposed, a better agreement with the present experimental data was achieved.

Acknowledgements

This work is supported by Major State Basic Research Development Program of China (Grant no.1999022308) and National Natural Science Foundation of China (Grant no. 59995462).

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