PERGAMON

International Journal of Multiphase Flow 27 (2001) 2043–2062

International Journal of Multiphase Flow

www.elsevier.com/locate/ijmulflow

Heat transfer correlation for boiling flows in small rectangular horizontal channels with low aspect ratios

Han Ju Lee, Sang Yong Lee *

Department of Mechanical Engineering, Korea Advanced Institute of Science and Technology, 373-1, Kusong-Dong, Yusong-Gu, Taejon 305-701, Republic of Korea

Received 23 March 2001; received in revised form 15 August 2001

Abstract

In the present experimental study, a correlation is proposed to represent the heat transfer coefficients of the boiling flows through horizontal rectangular channels with low aspect ratios. The gap between the upper and the lower plates of each channel ranges from 0.4 to 2 mm while the channel width being fixed to 20 mm. Refrigerant 113 was used as the test fluid. The mass flux ranges from 50 to 200 $\mathrm{kg/m^2\ s}$ and the channel walls were uniformly heated up to 15 $\mathrm{kW/m^2}$. The quality range covers from 0.15 to 0.75 and the flow pattern appeared to be annular. The modified Lockhart–Martinelli correlation for the frictional pressure drop was confirmed to be within an accuracy of $\pm 20\%$. The heat transfer coefficients increase with the mass flux and the local quality; however the effect of the heat flux appears to be minor. At the low mass flux condition, which is more likely to be with the smaller gap size, the heat transfer rate is primarily controlled by the liquid film thickness. A modified form of the enhancement factor F for the heat transfer coefficient in the range of $Re_{\mathrm{LF}} \leqslant 200$ well correlates the experimental data within the deviation of $\pm 20\%$. The Kandlikar's flow boiling correlation covers the higher mass flux range ($Re_{\mathrm{LF}} > 200$) with 10.7% mean deviation. © 2001 Published by Elsevier Science Ltd.

Keywords: Two-phase flow; Horizontal rectangular channel; Frictional pressure drop; Two-phase frictional multiplier; Boiling heat transfer

1. Introduction

Compact heat exchangers are widely adopted in refrigeration and air-conditioning industries because of their high heat exchanging performances. Mostly, a compact heat exchanger is

* Corresponding author. Tel.: +82-42-869-3026; fax: +82-42-869-8207.
E-mail address: e_hyunny@cais.kaist.ac.kr (S.Y. Lee).

PII: S0301-9322(01)00054-4

composed of an array of rectangular channels with the small gap size (typically in the range of 0.5–2 mm). However, only a limited number of works have been reported on the frictional pressure drop and boiling heat transfer within the small channels especially at the low flow rate condition.

In boiling channels, heat is transferred either by nucleate boiling or two-phase forced convection. Nucleate boiling mode is the dominant heat transfer mechanism in the high heat flux and the low quality region. The corresponding flow patterns are the bubbly and the slug flows. Heat transfer rate is enhanced by the flow mixing through repetition of bubble formation and departure from the heated wall. Two-phase forced convection dominant region, however, is most likely to be in an annular flow. Heat is transferred by the conduction and the convection modes through the liquid film and the evaporation takes place continuously at the interface. Heat transfer rate increases with the increases in quality and mass velocity.

Cornwell and Kew (1993) and Kew and Cornwell (1995) have examined the relationships between flow patterns and heat transfer coefficients for small rectangular vertical channels with the cross section of 1.2 mm × 0.9 mm and 3.25 mm × 1.1 mm. R-113 was used as the working fluid. They identified three flow patterns through flow visualization experiments. They are the isolated bubbles (IB), confined bubbles (CB) and the annular-slug flow (ASF) regimes. The isolated-bubbles regime is found in a very low quality region and the bubble sizes are small compared to the channel gap size. In the confined-bubbles regime, however, the bubbles are as large as the gap size and their motions are restricted by the channel walls. They proposed Confinement number as

$$N_{\mathrm{CONF}} = \left(\frac{\sigma}{g(\rho_{\mathrm{L}} - \rho_{\mathrm{G}})} \right)^{0.5} / D_{\mathrm{h}}. \tag{1}$$

Here, $\rho_{\mathrm{L}}, \rho_{\mathrm{G}}, D_{\mathrm{h}}, \sigma$ and g denote the liquid and gas densities, hydraulic diameter of the channel, surface tension and the gravitational constant, respectively. The Confinement number was used as an important dimensionless parameter in the confined-bubbles regime. The annular-slug flow pattern was observed at the quality less than 0.2, and the heat transfer rate increased as the quality increased. They proposed the heat transfer coefficient correlations for each region as follows:

$$\text{IB region}: \quad Nu = C_1 Re_{\mathrm{Lo}}^{0.8} Bo^{0.7}, \tag{2}$$

$$\text{CB region}: \quad Nu = C_2 Re_{\mathrm{Lo}}^{0.8} Bo^{0.3} N_{\mathrm{CONF}}^{0.5} Pr_{\mathrm{L}}^{0.4}, \tag{3}$$

$$\text{ASF region}: \quad Nu = C_3 F Nu_{\mathrm{Lo}}. \tag{4}$$

Here, C_1, C_2 and C_3 are the experimental constants and F is the enhancement factor. Dimensionless parameters in Eqs. (2)–(4) (i.e., Nusselt number, Boiling number, all-liquid Reynolds number, and Prandtl number) are defined as

$$Nu = \frac{hD_{\mathrm{h}}}{k_{\mathrm{L}}}, \quad Bo = \frac{q''}{Gi_{\mathrm{fg}}}, \quad Re_{\mathrm{Lo}} = \frac{GD_{\mathrm{h}}}{\mu_{\mathrm{L}}}, \quad Pr_{\mathrm{L}} = \frac{\mu_{\mathrm{L}} c_p}{k_{\mathrm{L}}}. \tag{5}$$

Here, h, k_{L}, q'', G, i_{fg}, μ_{L} and c_p denote the heat transfer coefficient, liquid thermal conductivity, heat flux, mass flux, latent heat of vaporization, liquid viscosity and the specific heat, respectively.

Lazarek and Black (1982) examined the boiling heat transfer and the pressure drop of a R-113 flow in a vertical tube with 3.1 mm internal diameter. They reported that the conventional Lockhart–Martinelli correlation can be used with the value of C being 30 for the turbulent (liquid)–turbulent (gas) regime. They also found that the quality had no influence upon the heat transfer rate and proposed a correlation similar to Eq. (2) as

$$Nu = 30Re_{\mathrm{Lo}}^{0.857}Bo^{0.714}. \tag{6}$$

Tran et al. (2000) performed a series of experiments on the frictional pressure drops of R-134a, R-12 and R-113 flows using 2.46 and 2.92 mm circular tubes and a rectangular channel with 4.06 mm × 1.7 mm. They correlated the experimental pressure drop data using the modified B-coefficient method suggested by Chisolm (1983) in terms of the Confinement number (Eq. (1)) for the turbulent (liquid)–turbulent (gas) regime.

Tran et al. (1993, 1996) measured boiling heat transfer coefficients for a R-12 flow in a horizontal rectangular channel with 4.06 mm × 1.7 mm and in a horizontal circular tube of 2.46 mm in diameter, respectively. Wambsganss et al. (1993) obtained the heat transfer coefficients for a boiling flow of R-113 within a horizontal circular tube of 2.92 mm in diameter. Tran (1998) pointed out that the boiling heat transfer results of Tran et al. (1993, 1996) and Wambsganss et al. (1993) correspond to the nucleate-boiling-dominant region where quality ranges up to 0.6–0.8 and proposed the correlation having the similar form of Eq. (3) as

$$Nu = 770(Re_{\mathrm{Lo}}N_{\mathrm{CONF}}Bo)^{0.62}\left(\frac{\rho_{\mathrm{G}}}{\rho_{\mathrm{L}}}\right)^{0.297}. \tag{7}$$

Cornwell and Kew (1993), Kew and Cornwell (1995), Kuznetsov and Shamirzaev (1999), Oh et al. (1998), Mandrusiak and Carey (1989), Robertson (1982, 1983) and Robertson and Lovegrove (1983) found that the dominant heat transfer mechanism was the two-phase forced convection.

Kuznetsov and Shamirzaev (1999) studied the boiling heat transfer phenomena of a R-318C flow in a horizontal small annulus with 0.9 mm gap. They showed that Eq. (7) represented experimental data well where quality ranges below 0.3 but the error increased as the film thickness decreased (i.e., as the quality increased). They also reported that the corresponding flow pattern was annular when the quality was greater than 0.3.

Oh et al. (1998) performed experiments on the boiling heat transfer of R-134a flowing through horizontal tubes with 0.75, 1 and 2 mm in diameters. They showed that the boiling heat transfer characteristics belonged to the two-phase forced convection in the quality ranges above 0.1.

Mandrusiak and Carey (1989) examined boiling heat transfer in vertical offset finned channels with 1.91, 3.81 and 9.52 mm in fin heights using water, methanol, butanol and R-113. They reported that the boiling heat transfer phenomenon was characterized by the two-phase forced convection and the nucleate boiling was fully suppressed in most quality ranges, and suggested that the enhancement factor, F, could be expressed as a function of Martinelli parameter.

Robertson (1982, 1983) and Robertson and Lovegrove (1983) examined the boiling flows of R-11 and N_2 in vertical rectangular channels. The channel cross section was divided by plate fins and was composed of small sub-channels with the hydraulic diameter of 2.4 mm. They emphasized that the operating conditions of compact heat exchangers with the fins correspond to the low flow rates and the low heat fluxes. The mass flux and heat flux conditions in their boiling experiments were up to 150 kg/m^2 s and 4 kW/m^2, respectively. They showed that the heat transfer mode was the two-phase forced convection. They also represented the experimental data based on the film-flow model of Hewitt and Hall-Taylor (1970), applicable to low flow rate conditions with small passages, provided that the local two-phase pressure gradient is known. This implies that the boiling heat transfer coefficient is somehow related to the two-phase frictional multiplier.

Nevertheless, for the two-phase forced convection regime, no reliable correlation for the heat transfer coefficient based on the film flow model has been reported. Therefore, in the present study, a heat transfer correlation was proposed for boiling flows within narrow gap channels with their aspect ratios ranging from 0.1 to 0.02, much smaller than the previous works. The heat transfer coefficient was correlated to the two-phase frictional multiplier and the channel aspect ratio.

2. Experiments

Fig. 1 shows an experimental setup for the boiling heat transfer and pressure drop measurements. The experimental setup is composed of a R-113 loop and a water-cooling loop to control the R-113 temperature. The R-113 loop consists of a gear pump, rotameters, test section, cooler, pre-heater, condenser and an after-condenser. The gear pump and rotameters with three different measurable ranges (20, 100 and 350 cm^3/min) are used to control the flow rate. The pre-heater for the adjustment of the temperature and the quality at the test section inlet is made of 3 m-long

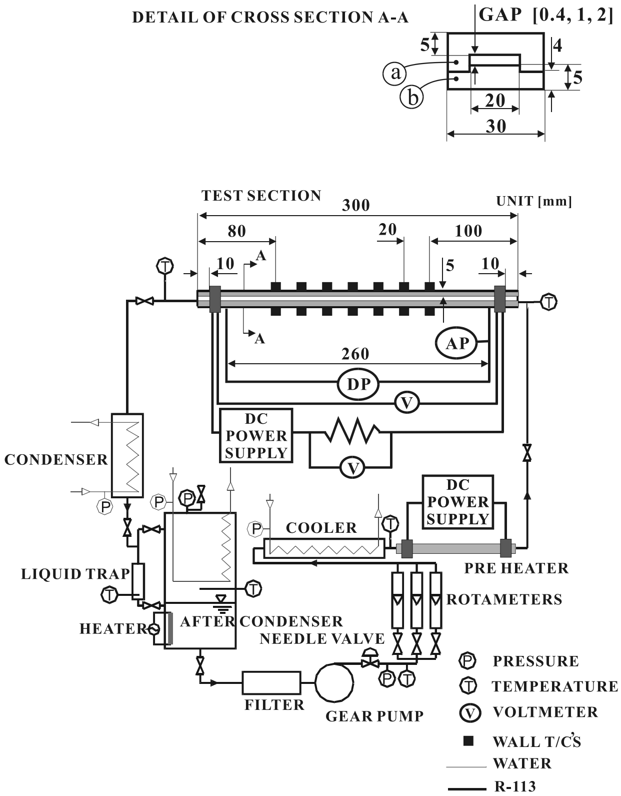

Fig. 1. Schematic diagram of the experimental system.

stainless steel tube with 6.35 mm outer diameter. The pre-heater is the direct-resistance heating type using the high-current-regulated DC power supply and the inlet quality was controlled from 0.15 to 0.75. The cooler and the condenser are the concentric counter-flow type heat exchangers made of brass tubes with their outer diameters being 12.7 and 19.05 mm. Within the after-condenser, a coil-type heat exchanger and a heater with 5 kW were installed at the upper and the lower parts, respectively, to control the system pressure; the after-condenser is also used as the R-113 storage tank. The test section has the rectangular cross section with the height (gap) much smaller than the width. As shown in Fig. 1, test section is composed of the upper and the lower parts $(<)$ and (∇). The center portion of part ∇ is protruded by 1 mm from its edge while that of part $<$ being grooved with its depth larger than 1 mm to form a rectangular channel. Three different gap sizes were tested; the gap sizes are 0.4, 1 and 2 mm while the width of the channel is fixed to 20 mm. Thus the channel aspect ratio ranges from 0.02 to 0.1. The tolerances of the gaps of the channels were maintained within 0.01 mm. The test sections are made of stainless steel plates as the current-carrying medium. A high-current-regulated DC power supply with the maximum capacity of 660 A/10 V was used to heat the test section; the heat flux could be raised up to 15 kW/m^2. The entire length of the test section is 300 mm and two pressure taps located 20 mm inwards from each end were drilled to measure the absolute pressure and the differential pressure drop through the test section. Self-adhesive K-type thermocouples (Omega) were used to measure the wall temperature distribution. Signals from the pressure transducers and the thermocouples were recorded using a DT3001PGL board and a 32-channel DAS TC/B board, respectively.

Local boiling heat transfer coefficients were obtained as

$$h = \frac{q''}{(T_w - T_{sat})}. \tag{8}$$

Here, saturation temperature, T_{sat}, was obtained indirectly considering two-phase flow pressure drop along the flow direction. The inner wall temperature T_w can be obtained by measuring the outer wall temperature $T_{w,0}$ assuming steady, uniform volumetric heat generation within the test section wall as (Wambsganss et al. (1993))

$$T_w = T_{w,0} - \left(\frac{q'''t^2}{2k_w}\right). \tag{9}$$

Here, t is the thickness of the stainless steel walls of the test section (5 mm in the present case) and q''' stands for the volumetric heat generation within the wall material. The qualities at the inlet and the outlet of the test section were calculated from the following energy balance relations.

$$x_{in} = \left[\frac{q_{pre}}{W} - c_{p,L}\Delta T\right] \Big/ i_{fg}, \tag{10}$$

$$x_{out} = x_{in} + \frac{q_{test}}{W i_{fg}}. \tag{11}$$

Here, x, W and ΔT denote quality, mass flow rate and temperature rise between the inlet and the outlet of the pre-heater, respectively. The local qualities along the test section could also be derived from the qualities at the inlet and the outlet of the test section assuming a linear quality distribution.

An uncertainty analysis has been performed according to the method proposed by Kline (1985). The estimated uncertainties of R-113 flow rates, temperature and pressure measurements are ±2%, ±0.17 °C and ±2%, respectively. The uncertainties of the qualities at the inlet of and within the test section, frictional and accelerational pressure drops, and supply powers to the pre-heater and the test section are ±9.85%, ±13.2%, ±2.4%, ±21.3%, ±4.95% and ±4.81%, respectively. Also the uncertainties of the heat transfer coefficient ranges from ±5.22% to ±18.1%.

3. Single-phase flow pressure drop and heat transfer

Prior to performing boiling heat transfer experiments, friction factor and Nusselt number for single-phase flow were obtained to check the reliability of the experimental system. The friction factors for laminar flow through rectangular channels were obtained by using the simplified (polynomial type) equation that fits the exact analytical solution within an accuracy of ±0.05% by Hartnett and Kostic (1989).

$$fRe_{D_h} = h(\alpha), \tag{12}$$

$$h(\alpha) = 24(1 - 1.3553\alpha + 1.9467\alpha^2 - 1.7012\alpha^3 + 0.9564\alpha^4 - 0.2537\alpha^5). \tag{13}$$

Here,

$$f\left(= \frac{D_h}{2\rho V^2}\left(\frac{dp}{dz}F\right)\right)$$

stands for the friction factor. Also V and $(dp/dz)F$ denote the mean velocity and the frictional pressure gradient, respectively, and α represents the aspect ratio defined as the value of the height divided by the width of the channel cross-section. As seen in Fig. 2, the experimental results with R-113 agree with Eq. (12) for the laminar flows. Hartnett and Kostic (1989) proposed that hydraulic entrance length, L_{HY}, in rectangular channel is a function of the aspect ratio as

$$L_{HY} = fn(\alpha)Re_{D_h}D_h. \tag{14}$$

The hydraulic entrance length is reduced as the aspect ratio decreases and the relationship between L_{HY} and $fn(\alpha)$ is shown in a graphical form by Hartnett and Kostic (1989). However, the entrance effect in the present experimental conditions is negligible as evidenced in Fig. 2.

Nusselt numbers for the laminar flow through rectangular channels in the uniform heat flux conditions were obtained by using the simplified equation that fits the exact analytical solution within an accuracy of ±0.03% by Hartnett and Kostic (1989) also.

$$Nu = f(\alpha), \tag{15}$$

$$f(\alpha) = 8.235(1 - 2.0421\alpha + 3.0853\alpha^2 - 2.4765\alpha^3 + 1.0578\alpha^4 - 0.1861\alpha^5). \tag{16}$$

Fig. 3 shows the present experimental results on the dimensionless thermal entrance length $x^+(= 2(x/D_h)/Re_{D_h}Pr)$. For the laminar friction factors, Eq. (12) could be used without considering the entrance effect. On the other hand, the thermal entrance length must be considered as shown in Fig. 3. This is because the thermal entrance length, L_{TH}, depends on Prandtl number which is greater than 1 as

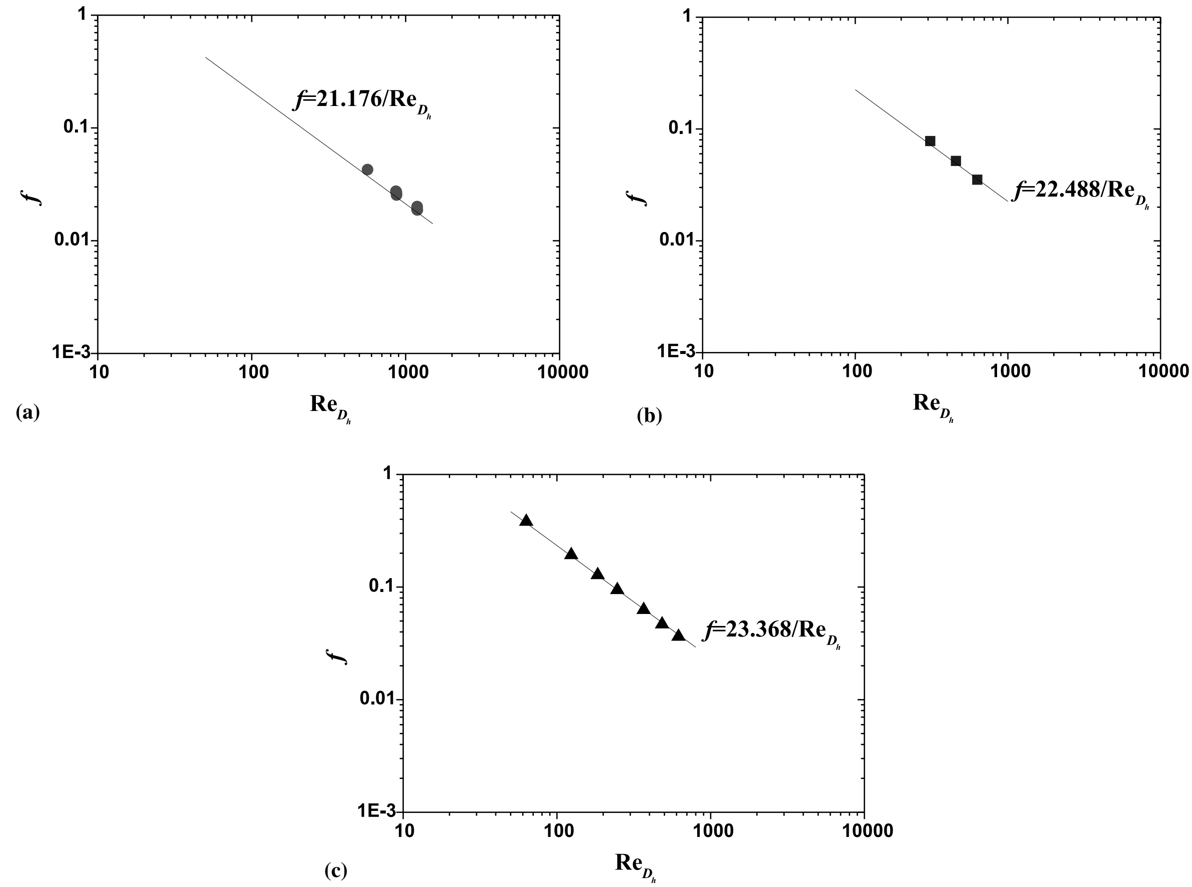

Fig. 2. Friction factors for single-phase flows in rectangular channels: (a) gap 2 mm; (b) gap 1 mm; (c) gap 0.4 mm.

$$L_{\mathrm{TH}} = fn'(\alpha)Re_{D_{\mathrm{h}}}PrD_{\mathrm{h}}. \tag{17}$$

Here, the relationship between L_{TH} and $fn'(\alpha)$ is shown graphically by Hartnett and Kostic (1989). In other words, the thermal entrance length is greater than the hydraulic entrance length, and the entrance effect for the larger gap size, especially for the 2 mm-gap case, should not be neglected. Kays and Perkins (1985) suggested the relationship between aspect ratio and Nusselt number in the thermally developing region as a tabular form and the fitted lines from the numerical table represent the experimental data well as shown in Fig. 3.

4. Frictional pressure drop in boiling flows

Pressure drop in boiling flows consists of two parts; accelerational and frictional pressure drops. The accelerational pressure drop is expressed by Tran (1998) as

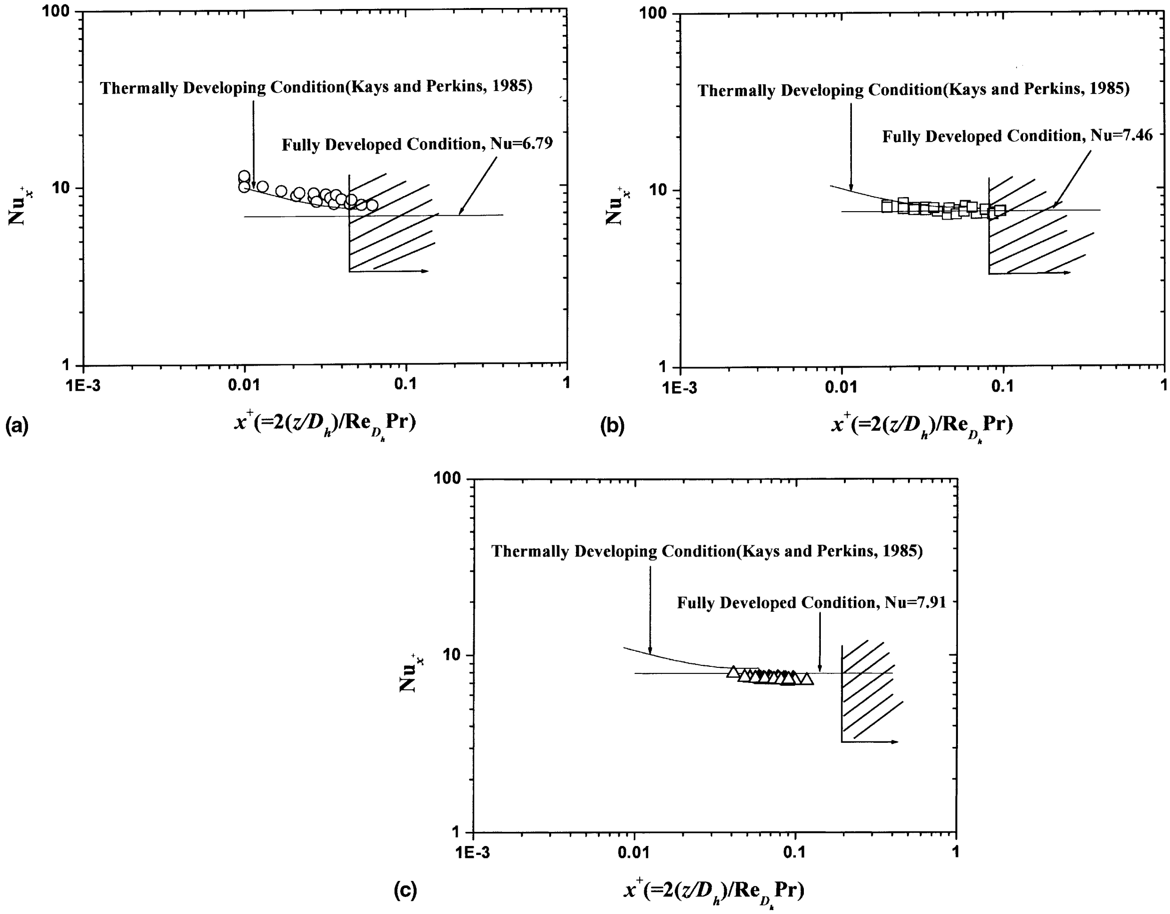

Fig. 3. Laminar Nusselt values for single-phase flows in rectangular channels: (a) gap 2 mm; (b) gap 1 mm; (c) gap 0.4 mm (The hatched area denotes the experimental conditions for boiling flow with $Re_{\mathrm{LF}} < 200$.)

$$\Delta p_{\mathrm{A}} = G^2 \left\{ \left[\frac{x_{\mathrm{out}}^2}{\rho_{\mathrm{G}} \varepsilon_{\mathrm{out}}} + \frac{(1 - x_{\mathrm{out}})^2}{\rho_{\mathrm{L}} (1 - \varepsilon_{\mathrm{out}})} \right] - \left[\frac{x_{\mathrm{in}}^2}{\rho_{\mathrm{G}} \varepsilon_{\mathrm{in}}} + \frac{(1 - x_{\mathrm{in}})^2}{\rho_{\mathrm{L}} (1 - \varepsilon_{\mathrm{in}})} \right] \right\}. \tag{18}$$

Here, ε denotes the void fraction and was expressed by Zivi's correlation (1964) as

$$\varepsilon = \left[1 + \left(\frac{1 - x}{x} \right) \left(\frac{\rho_{\mathrm{G}}}{\rho_{\mathrm{L}}} \right)^{0.67} \right]^{-1}. \tag{19}$$

Then the frictional pressure drop can be obtained by subtracting the accelerational pressure drop (Eq. (18)) from the measured two-phase pressure drop as

$$\Delta p_{\mathrm{F}} = \Delta p_{\mathrm{TP}} - \Delta p_{\mathrm{A}}. \tag{20}$$

In the adiabatic flow condition, there will be only the frictional pressure drop. Fig. 4 shows typical variations of the pressure drops with the mass flux and the inlet quality for the gap size of 0.4 mm.

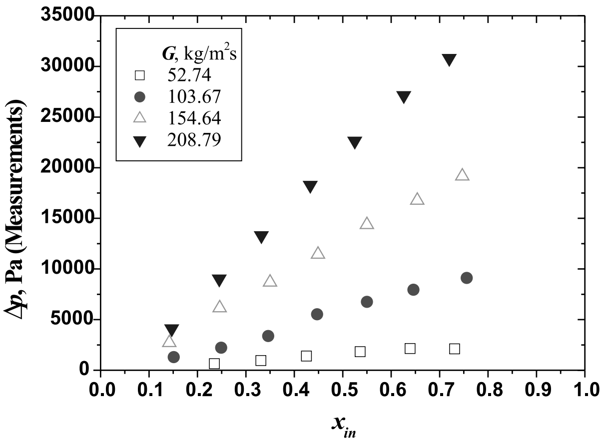

Fig. 4. Typical variations of the pressure drop (gap 0.4 mm, adiabatic condition).

As imagined, the pressure drop increases with the increases in the mass flux and the inlet quality. From the measured results, the two-phase frictional multiplier, ϕ_L was obtained as a function of the Martinelli parameter, X as shown in Fig. 5. The two-phase frictional multiplier and the Martinelli parameter are defined as

$$\phi_L^2 = \left(\frac{dp}{dz}F\right)_{TP} \bigg/ \left(\frac{dp}{dz}F\right)_L, \tag{21}$$

$$X^2 = \left(\frac{dp}{dz}F\right)_L \bigg/ \left(\frac{dp}{dz}F\right)_G. \tag{22}$$

The Lockhart–Martinelli (1949) correlation is expressed as

$$\phi_L^2 = 1 + \frac{C}{X} + \frac{1}{X^2}. \tag{23}$$

In the classical literature (Chisolm, 1967), the constant value of 12 has been proposed for C for the flow regime with the liquid and the gas phases being laminar and turbulent, respectively, which corresponds to the present case. However, as seen in Fig. 5, Eq. (23) with $C = 12$ cannot represent the experimental data well, especially with the smaller gap size. The two-phase frictional multiplier is obviously smaller with the smaller gap size as shown in Fig. 5. Besides, the two-phase frictional multiplier depends on the mass flux. Lowry and Kawaji (1988) and Wambsganss et al. (1992) have reported the effect of the flow rate on the two-phase frictional multiplier. On the other hand, concerned with the effect of the gap size, Mishima et al. (1993) and Mishima and Hibiki (1996) proposed a correlation for parameter C as a function of the hydraulic diameter. Later, to examine the effects of the gap size and the flow rate simultaneously, Lee and Lee (2001) performed a series of experiments on an air–water two-phase flow using the rectangular channels with $(0.4, 1, 2, 4)$ mm $\times$ 20 mm. They proposed a correlation for parameter C for the laminar(liquid)–turbulent(gas) regime as follows:

$$C = (6.185 \times 10^{-2}) Re_{Lo}^{0.726}. \tag{24}$$

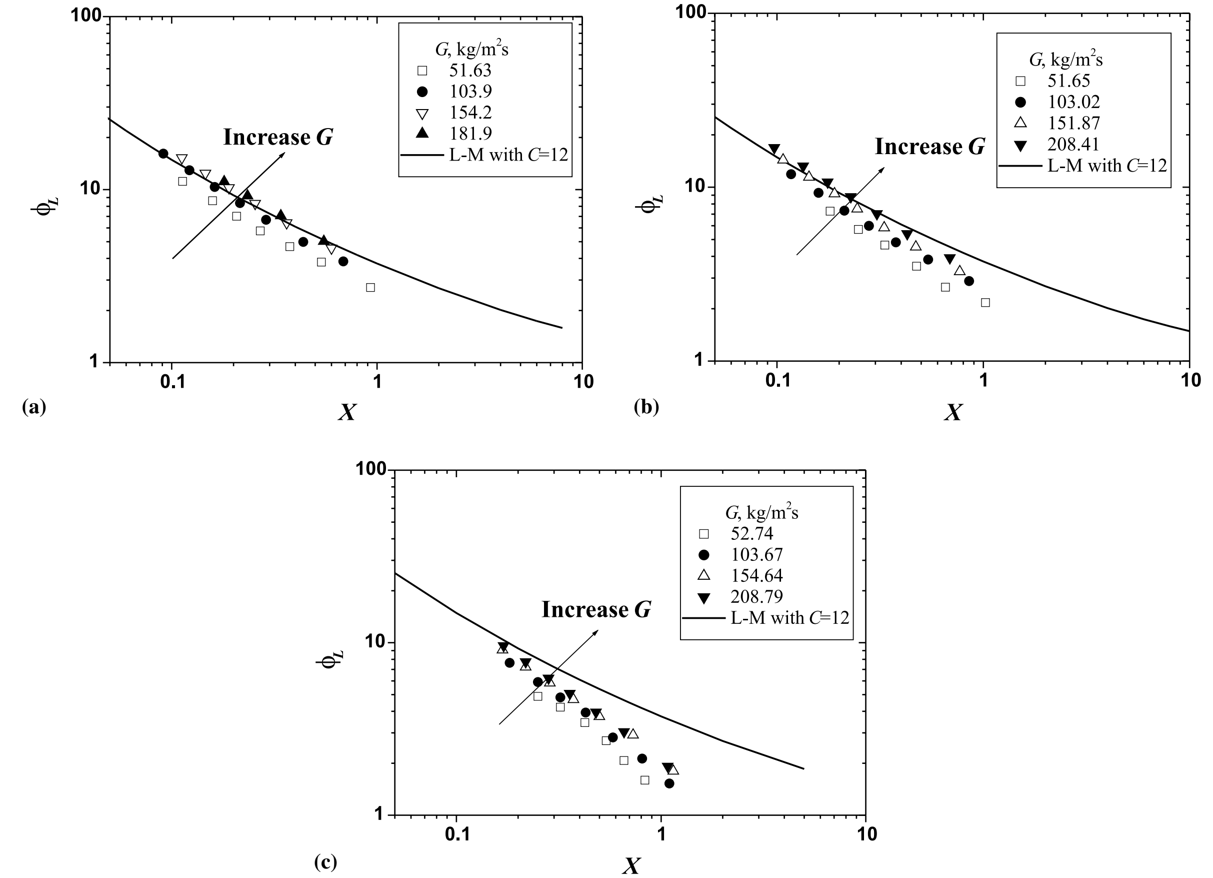

Fig. 5. Comparison between the present measurements and Lockhart–Martinelli correlation: (a) gap 2 mm; (b) gap 1 mm; (c) gap 0.4 mm.

Fig. 6 shows that, though there is a tendency of underprediction, the values of ϕ_{L} obtained by Eqs. (23) and (24) (for the air–water flow) agree with the present measurements (for R-113) within $\pm 15\%$ without adjusting the value of C.

To predict the frictional pressure drop for a boiling flow, the quality variation along the flow direction must be taken into account as

$$\Delta p_{\mathrm{F}} = \frac{L}{(x_{\mathrm{out}} - x_{\mathrm{in}})} \int_{x_{\mathrm{in}}}^{x_{\mathrm{out}}} \phi_L^2 \left(\frac{\mathrm{d}p}{\mathrm{d}z} F \right)_{\mathrm{L}} \mathrm{d}x. \tag{25}$$

Here, L denotes the distance between the pressure taps. Fig. 7 compares the measured pressure drops with the predicted ones obtained from Eq. (25) using the two-phase frictional multipliers by Lockhart–Martinelli (1949), Mishima and Hibiki (1996), Friedel (1979), Tran et al. (2000) and Lee (2001). The results of Lockhart–Martinelli (1949) and Mishima and Hibiki (1996) show somewhat larger deviations than the results of Lee and Lee (2001). This is because Lockhart–Martinelli correlation on ϕ_{L} does not count the gap size and the mass flux effects at all, and also Mishima

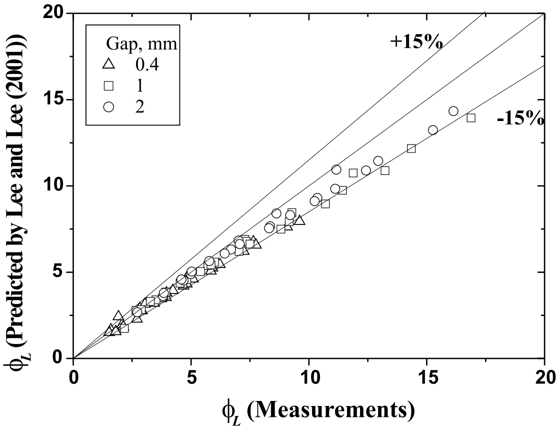

Fig. 6. Comparison between the present measurements and the correlation by Lee and Lee (2001).

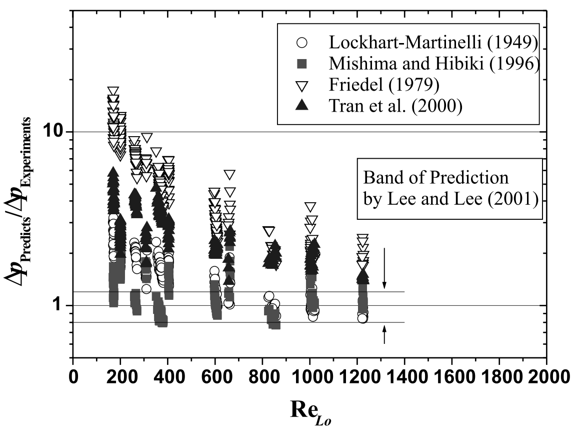

Fig. 7. Comparison between the present measurements and the previous correlations.

and Hibiki (1996) took account of the size effect only. The Friedel correlation has been recommended to predict frictional pressure drop for the condition of $\mu_L/\mu_G < 1000$ (Collier and Thome, 1994). However, the deviations from the measured results are large for the present case as exhibited in Fig. 7. Especially, the errors are large at low Reynolds numbers. This is because the Friedel correlation was basically developed for turbulent flows. Triplett et al. (1999) has reported that the Friedel correlation over-predicts the pressure drop more than 10 times that in the laminar flow region. Tran et al. (2000) modified B-coefficient method proposed by Chisolm (1983) using the Confinement number. The present experimental conditions belong to the two-phase forced convection region where the flow pattern is most likely to be annular as discussed by Lee (2001). Thus the Confinement number which represents the restriction on the bubble motion by the channel gap size is not suitable for the present experimental conditions. Therefore the deviations of the measured data from the correlation of Tran et al. (2000) are large as shown in Fig. 7. On the other hand, Eq. (25) with the modified Lockhart–Martinelli correlation using the parameter C

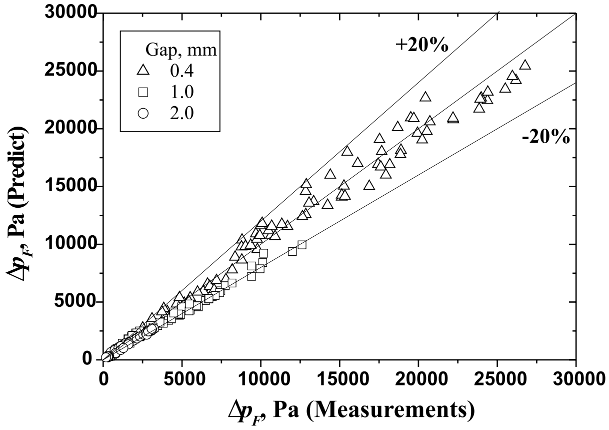

Fig. 8. Comparison between the present measurements and the correlation by Lee and Lee (2001).

(Eq. (24)) proposed by Lee and Lee (2001) represents the experimental results well within ±20% as shown in Figs. 7 and 8.

5. Boiling heat transfer

Fig. 9 shows the variations of boiling heat transfer coefficients with quality, mass flux, heat flux and channel gap size. Except for the lowest mass flux conditions ($G = 51.5$–51.7 kg/m^{2}s) in Figs. 9(a) and (b), the heat flux has minor effect on the boiling heat transfer coefficient in all gap sizes. On the other hand, the boiling heat transfer coefficient obviously increases with the increases of the quality and the mass flux. From these qualitative observations, the two-phase forced convection is considered to be the predominant mechanism in heat transfer. When Fig. 9 is carefully examined, the effect of the mass flux on the heat transfer coefficient becomes smaller as the gap size gets smaller. In other words, the effect of the quality change (i.e., the variation of the film thickness) on the heat transfer coefficient becomes greater than the mass flux effect with the smaller gap size.

In the second column of Table 1, all the present experimental results were compared with the flow boiling correlations by Kandlikar (1990), Shah (1982), Jung et al. (1989) and Wattelet (1995). Less than 70% of the data points stay within ±20% range of their correlations. This attributes to the fact that the above general correlations are based on the turbulent flow results. Robertson (1982) stated that the film-flow model of Hewitt and Hall-Taylor (1970) is applicable to the low flow rate conditions inside the small passages when the local two-phase pressure gradient is known. This model was developed for the vertical annular flow with the assumption of the universal velocity profile within the liquid film. In small channels, the gravitational effect is negligibly small and the model is also applicable to the horizontal flow case. Thus the force balance for the control volume including the liquid and the gas phases can be written simply as

$$\left(-\frac{\mathrm{d}p}{\mathrm{d}z}\right) = \frac{4\tau_\mathrm{w}}{D_\mathrm{h}}. \tag{26}$$

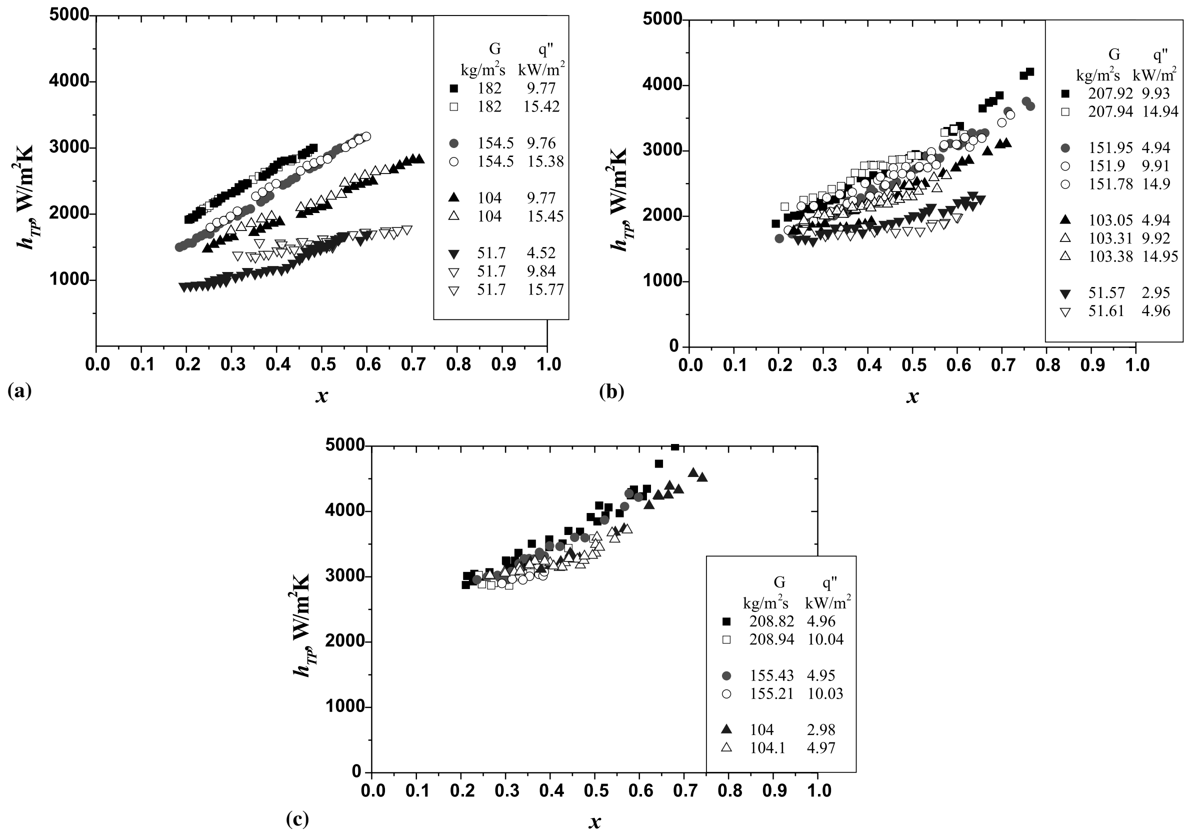

Fig. 9. Boiling heat transfer coefficients in rectangular channels with qualities, mass fluxes, and heat fluxes: (a) gap 2 mm; (b) gap 1 mm; (c) gap 0.4 mm.

Table 1
Statistical comparison of correlations

	All data		Data within $Re_{LF} > 200$	
	MD[a] (%)	Data within ±20% range (%)	MD[a] (%)	Data within ±20% range (%)
Kandlikar (1990)	16	69.8	10.7	86.7
Shah (1982)	17	68.6	11.1	88.1
Wattelet (1995)	22.5	54.3	15.8	71.1
Jung et al. (1989)	14.6	68.9	11.1	81.5

[a] Mean deviation $= (1/N)\sum \frac{|h_{\text{meas.}} - h_{\text{pred.}}|}{h_{\text{meas.}}}$.

Here, τ_w denotes the wall shear stress. The relationship between the dimensionless film thickness and the film flow rate has already been proposed by Hewitt and Hall-Taylor (1970) as

$$\begin{aligned} W^+ &= 0.5m^{+2}, \quad 0 < m^+ < 5, \\ W^+ &= -0.805m^{+2} + 5m^+ \ln m^+ + 12.45, \quad 5 \leqslant m^+ < 30, \\ W^+ &= 8.0m^{+2} + 2.5m^+ \ln m^+ - 214, \quad m^+ \geqslant 30, \end{aligned} \tag{27}$$

$$W^+ = \frac{W_{\mathrm{LF}}}{P\mu_{\mathrm{L}}} = \frac{Re_{\mathrm{L}}}{4}, \tag{28}$$

$$m^+ = \frac{\delta\sqrt{\tau_{\mathrm{w}}\rho_{\mathrm{L}}}}{\mu_{\mathrm{L}}}. \tag{29}$$

Here, W_{LF}, P and δ denote the mass flow rate of the liquid film, the wall perimeter and the liquid film thickness, respectively. Hewitt and Hall-Taylor (1970) provided the graphical relationships between the film Nusselt number and the film Reynolds number defined as

$$Nu_{\mathrm{LF}} = \frac{h_{\mathrm{TP}}\delta}{k_{\mathrm{L}}}, \tag{30}$$

$$Re_{\mathrm{LF}} = \frac{4\delta u_{\mathrm{LF}}\rho_{\mathrm{L}}}{\mu_{\mathrm{L}}} = \frac{\rho_{\mathrm{L}} j_{\mathrm{L}} D_{\mathrm{h}}}{\mu_{\mathrm{L}}} \tag{31}$$

for different values of the Prandtl number. Here, h_{TP}, k_{L}, u_{LF} and j_{L} denote the two-phase heat transfer coefficient, liquid thermal conductivity, mean velocity of the liquid film and the liquid superficial velocity, respectively. The liquid film thickness, δ can be calculated from Eqs. (26)–(29). The same relationships were tabulated in the book of Collier and Thome (1994).

In Fig. 10, the calculated results were plotted using the film-flow model for the mass flux and the gap size ranges of 50–200 $\mathrm{kg/m^2\,s}$ and 0.4–4 mm, respectively. With the low mass flux (region A), the heat transfer coefficients are the function of the liquid film thickness only and the mass flux has almost no effect. That is, the heat transfer coefficients increase only with the decrease of the film thickness. If the liquid flow rate becomes extremely small, heat is transferred only by the conduction mode through the liquid film as

$$h_{\mathrm{TP}} = \frac{k_{\mathrm{L}}}{\delta}. \tag{32}$$

Region A corresponds to the range of the low film Reynolds number; and the film Reynolds number decreases with the decrease of the gap size for the same mass flux conditions. In region B,

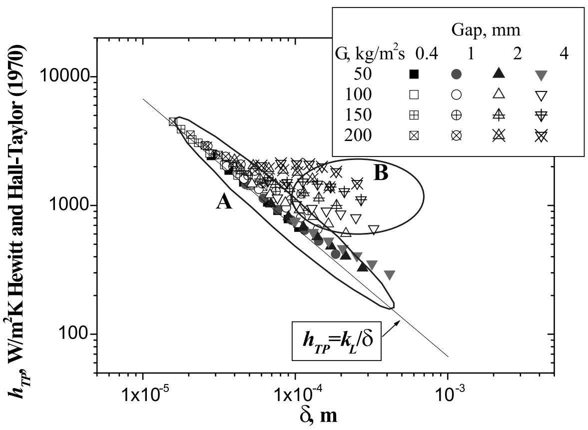

Fig. 10. Boiling heat transfer coefficients obtained from the film-flow model by Hewitt and Hall-Taylor (1970) (Region A is for the lower Re_{LF}; Region B is for the higher Re_{LF}).

the film Reynolds numbers are higher than those in region A. In other words, the film Reynolds number increases with increases in the mass flux and/or the channel gap size. Higher heat transfer coefficient with the larger mass flux (in Region B) attributes to the convection effect in the liquid film. Furthermore, the liquid film tends to be turbulent with the increase of the liquid mass flow rate. Collier and Thome (1994) reported that the laminar liquid film with the smooth interface could be sustained below the film Reynolds number (Re_{LF}) of 200. In this region, the mixing effect within the liquid film is negligible regardless of the core-gas velocity. The measured boiling heat transfer coefficients in Fig. 9 were replotted in Fig. 11 for the high and low liquid film Reynolds numbers with the criterion at $Re_{LF} = 200$. Fig. 11(a) again exhibits the heat transfer coefficient is only a function of the film thickness. In Fig. 11(b), however, the heat transfer coefficient depends on the factors other than the film thickness and the mixing effect (i.e., the film turbulence or, at least, the surface wavyness) is considered to be the most probable factor. As already stated, the general correlations reported by the researchers in Table 1 are based on the experimental results on the turbulent flow. That is why mean deviations shown for the case with $Re_{LF} > 200$ in the third column of Table 1 are much smaller than that with all data points (in the first column of

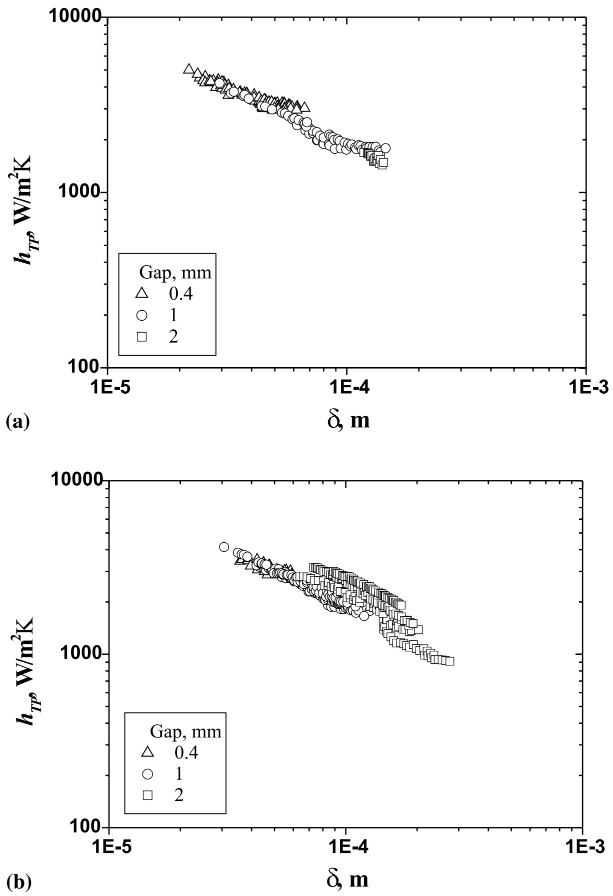

Fig. 11. Measured boiling heat transfer coefficients with the liquid film thickness: (a) $Re_{LF} \leqslant 200$; (b) $Re_{LF} > 200$.

Table 1). The Kandlikar (1990) correlation best predicts the boiling heat transfer coefficients for the ranges of $Re_{LF} > 200$. The Kandlikar correlation, developed for the saturated flow boiling inside vertical and horizontal tubes based on a total of 5246 data points for water, R-11, R13-B1, R-22, R-113, R-114, R-152a, nitrogen, and neon, is given by

$$h_{TP} = \lfloor D_1 C_0^{D_2} (25Fr_L)^{D_5} + D_3 Bo^{D_4} F_{fl} \rfloor h_L, \tag{33}$$

where the symbols and the values of the constants are given in Table 2.

In the low flow rate condition ($Re_{LF} \leqslant 200$), the heat transfer coefficients solely depends on the liquid film thickness as in Fig. 11(a). Chen (1963) reported that the enhancement factor, F (the ratio of the boiling heat transfer coefficient to the single-phase heat transfer coefficient), could be represented by the Martinelli parameter basically in the turbulent film-flow region. However, the same concept may be extended to the laminar film flows as far as the flow pattern remains to be annular. Therefore the Chen (1963)'s process was re-examined to predict the heat transfer coefficient for the laminar film flows within rectangular channels. In this case, the information on the film thickness is essential in predicting the heat transfer coefficient. For a rectangular channel with the aspect ratio, α, parameter F can be written as

$$F = \frac{h_{TP}}{h_L}. \tag{34}$$

Here, h_{TP} and h_L denote the two-phase heat transfer coefficient and the single-phase liquid heat transfer coefficient based on the liquid component flow, respectively, and the heat transfer coefficients for the both cases can be written down as

$$h_L = f(\alpha) \frac{k_L}{D_h}, \tag{35}$$

$$h_{TP} = g(\alpha) \frac{k_L}{4\delta}. \tag{36}$$

Table 2
Constants and symbols for Eq. (33) (Kandlikar, 1990)

Constant	For $C_0 < 0.65$	For $C_0 > 0.65$
D_1	1.136	0.6683
D_2	−0.9	−0.2
D_3	667.2	1058
D_4	0.7	0.7
D_5*	0.3	0.3

$$C_0 = \left(\frac{1-x}{x}\right)^{0.8} \left(\frac{\rho_G}{\rho_L}\right)^{0.5}$$

$$h_L = 0.023 Re_L^{0.8} Pr_L^{0.4} \frac{k_L}{D}$$

$F_{fl} = 1.3$ for R-113

$$Fr_L = \frac{G^2}{\rho_L^2 g D}$$

* $D_5 = 0$ for $Fr_L > 0.04$.

Here, $f(\alpha)$ and $g(\alpha)$ stand for the effects of the aspect ratios. As already noted, $f(\alpha)$ is defined in Eq. (16), and $g(\alpha) \neq f(\alpha)$. The experimental conditions for the boiling flow with $Re_{\mathrm{LF}} \leqslant 200$ are indicated in Fig. 3 as the hatched regions. The single-phase heat transfer results for $Re_{\mathrm{LF}} \leqslant 200$ correspond to the thermally fully developed condition expressed by Eq. (15) regardless of the gap size. Thus F can be rewritten as

$$F = \frac{g(\alpha)}{f(\alpha)} \frac{D_{\mathrm{h}}}{4\delta} = fn\left(\alpha, \frac{D_{\mathrm{h}}}{4\delta}\right). \tag{37}$$

On the other hand, from Eqs. (21), (26) and (29), film thickness, δ may be expressed as

$$\delta \sim \tau_{\mathrm{w}}^{-0.5} \sim \left(-\frac{\mathrm{d}p}{\mathrm{d}z}F\right)_{\mathrm{TP}}^{-0.5} = \phi_{\mathrm{L}}^{-1}\left(-\frac{\mathrm{d}p}{\mathrm{d}z}F\right)_{\mathrm{L}}^{-0.5}. \tag{38}$$

Here,

$$\left(-\frac{\mathrm{d}p}{\mathrm{d}z}F\right)_{\mathrm{L}} = \frac{2\mu_{\mathrm{L}} j_{\mathrm{L}} h(\alpha)}{D_{\mathrm{h}}^2}$$

and $h(\alpha)$ is defined in Eq. (13). Since the two-phase frictional multiplier is a function of the mass flow rate, quality and the fluid properties, Eq. (38) should be simplified as

$$\delta \sim \phi_{\mathrm{L}}^{-1} h(\alpha)^{-0.5}. \tag{39}$$

Hence parameter F in Eq. (37) becomes a function only of the aspect ratio and the two-phase frictional multiplier as

$$F = fn(\alpha, \phi_{\mathrm{L}}). \tag{40}$$

This implies that the two-phase effect in heat transfer (F) can be directly correlated to the same effect in pressure drop (ϕ_{L}) for the laminar(liquid)–turbulent(gas) flow regime. From the measured data, F parameter for the ranges of $Re_{\mathrm{LF}} \leqslant 200$ can be expressed as

$$F = 10.3\alpha^{0.398}\phi_{\mathrm{L}}^{0.598}. \tag{41}$$

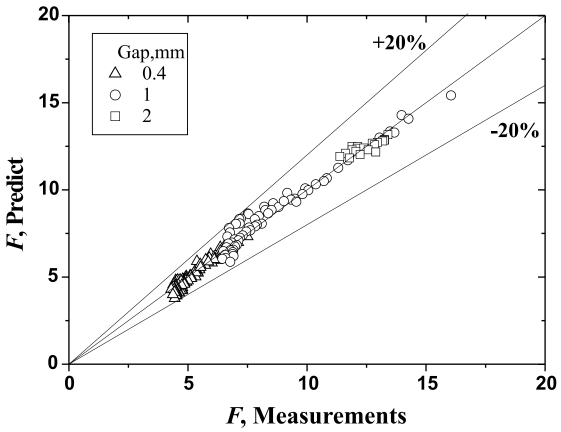

Fig. 12. Comparison between the measured F parameter and the predicted value.

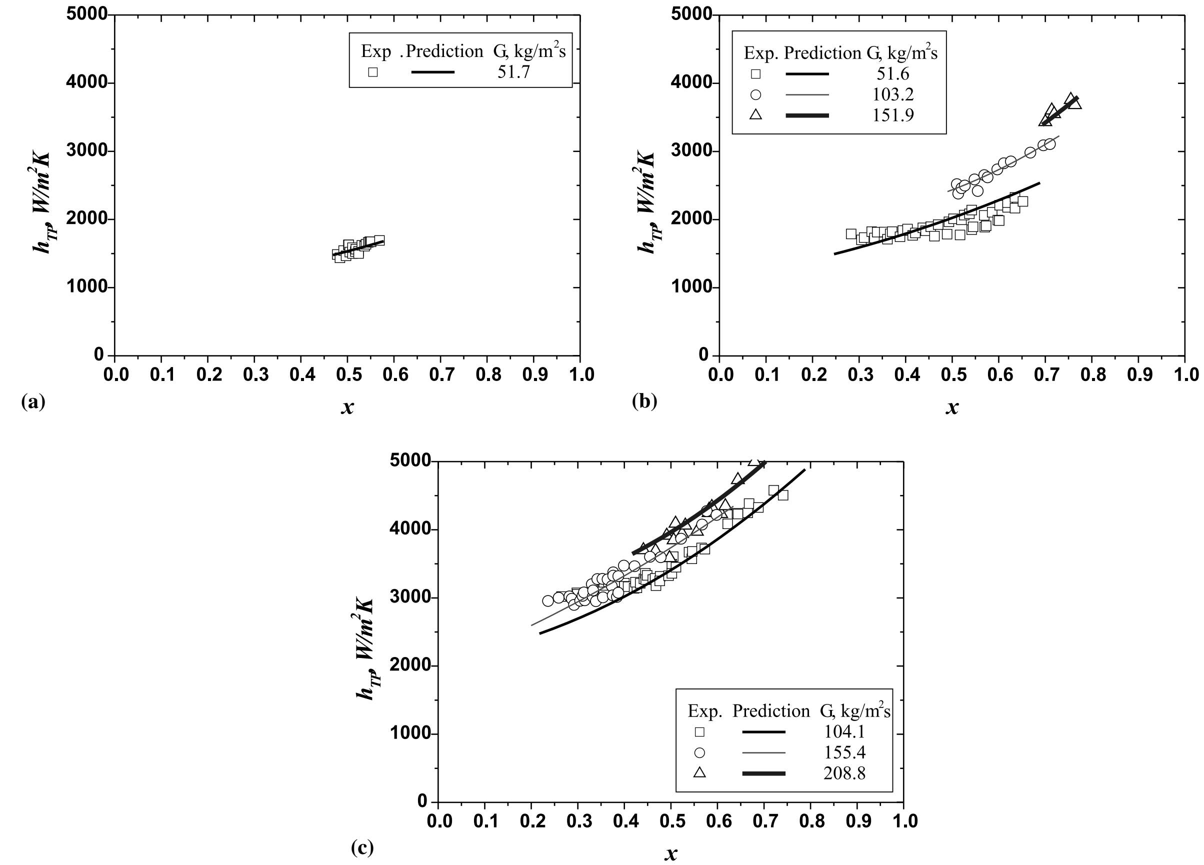

Fig. 13. Comparison between the measured boiling heat transfer coefficients and the predictions: (a) gap 2 mm; (b) gap 1 mm; (c) gap 0.4 mm.

The above equation represents the present experimental results within ±20% as shown in Fig. 12. Also, the measurements and the predictions of the boiling heat transfer coefficients for the ranges of $Re_{\mathrm{LF}} < 200$ are compared in Fig. 13. The newly developed correlation represent well the measured heat transfer coefficient with variations of the quality and the mass flux as indicated in Fig. 13.

6. Conclusions

In the present experimental study, a correlation is proposed to represent the heat transfer coefficients of the boiling flows through horizontal rectangular channels with low aspect ratios. The frictional pressure drops for the boiling flow predicted by using the modified Lockhart–Martinelli correlation (Lee and Lee (2001)) agree well with the measurements within the deviation of ±20%. The heat transfer coefficient increases with the mass flux and the local quality; however the effect of heat flux appeared to be minor. For the smaller gap size and the lower flow rate conditions, the heat transfer is primarily controlled by the film thickness. Parameter F for the heat transfer coefficient correlates the experimental data in the ranges of $Re_{LF} \leqslant 200$ within the deviation of ±20% when the two-phase frictional multiplier correlation proposed by Lee and Lee (2001) is adopted. For the higher flow rate ($Re_{LF} > 200$), the Kandlikar's flow boiling correlation represents the heat transfer data with 10.7% mean deviation.

Acknowledgements

This work was supported by a grant from the Critical Technology Project of the Ministry of Science and Technology, Korea and in part by the Brain Korea 21 Project in 2001.

References

Chen, J.C., 1963. A correlation for boiling heat transfer to saturated fluids in convective flow. ASME paper No. 63-HT-34.

Chisolm, D., 1967. A theoretical basis for the Lockhart–Martinelli correlation for two-phase flow. Int. J. Heat Mass Transfer 10, 1767–1778.

Chisolm, D., 1983. Two-Phase Flow in Pipelines and Heat Exchangers. Longman, New York.

Collier, J.G., Thome, J.R., 1994. Convective Boiling and Condensation. Clarendon, Oxford.

Cornwell, K., Kew, P.A., 1993. Boiling in small parallel channels. In: Pilavachi, P.A. (Ed.), Energy Efficiency in Process Technology. Elsevier, New York, pp. 624–638.

Hartnett, J.P., Kostic, M., 1989. Heat transfer to Newtonian and non-Newtonian fluids in rectangular ducts. Adv. Heat Transfer 19, 247–356.

Hewitt, G.F., Hall-Taylor, N.S., 1970. Annular Two-Phase Flow. Pergamon, Oxford.

Jung, D.S., Mclinden, M., Radermacher, R., Diddon, D., 1989. A study of flow boiling heat transfer with refrigerant mixtures. Int. J. Heat Mass Transfer 32, 1751–1764.

Kandlikar, S.G., 1990. A general correlation for saturated two-phase flow boiling heat transfer inside horizontal and vertical Tubes. Trans. ASME, J. Heat Transfer 112, 219–228.

Kays, W.M., Perkins, H.C., 1985. Forced convection, internal flow in ducts. In: Rohsenow, W.M., Hartnett, J.P., Ganic, E.N. (Eds.), Handbook of Heat Transfer (Chapter 7).

Kew, P.A., Cornwell, K., 1995. Confined bubble flow and boiling in narrow spaces. 10th International Heat Transfer Conference, 473–478.

Kline, S.J., 1985. The purposes of uncertainty analysis. Trans. ASME, J. Fluids Eng. 107, 153–160.

Kuznetsov, V.V., Shamirzaev, A.S., 1999. Two-phase flow pattern and flow boiling heat transfer in non-circular channel with a small gap. In: Proceedings of the Two-Phase Flow Modelling and Experimentation, pp. 249–253.

Lazarek, G.M., Black, S.H., 1982. Evaporative heat transfer, pressure drop and critical heat flux in a small vertical tube with R-113. Int. J. Heat Mass Transfer 25, 945–960.

Lee, H.J., 2001. An experimental study on two-phase flow and boiling heat transfer in horizontal rectangular channels with small heights. Ph.D. dissertation, KAIST.

Lee, H.J., Lee, S.Y., 2001. Pressure drop correlations for two-phase flow within horizontal rectangular channels with small heights. Int. J. Multiphase Flow 27, 783–796.

Lockhart, R.W., Martinelli, R.C., 1949. Proposed correlation of data for isothermal two-phase two-component flow in pipes. Chem. Eng. Prog. 45, 39–48.

Lowry, B., Kawaji, M., 1988. Adiabatic vertical two-phase flow in narrow flow channels. AIChE Symp. Ser. 84 (263), 133–139.

Mandrusiak, G.D., Carey, V.P., 1989. Convective boiling in vertical channels with different offset strip fin geometries. Trans. ASME, J. Heat Transfer 111, 156–165.

Mishima, K., Hibiki, T., 1996. Some characteristics of air–water two-phase flow in small diameter vertical tubes. Int. J. Multiphase Flow 22, 703–712.

Mishima, K., Hibiki, T., Nishihara, H., 1993. Some characteristics of gas–liquid flow in narrow rectangular ducts. Int. J. Multiphase Flow 19, 115–124.

Oh, H.K., Katsuta, M., Shibata, K., 1998. Heat transfer characteristics of R-134a in a capillary tube heat exchanger. 11th International Heat Transfer Conference 6, 473–478.

Robertson, J.M., 1982. The correlation of boiling coefficients in plate-fin heat exchanger passages with a film-flow model. 7th International Heat Transfer Conference, 341–345.

Robertson, J.M., 1983. The boiling characteristics of perforated plate-fin channels with liquid Nitrogen in upflow. Trans. ASME, HTD 27, 35–40.

Robertson, J.M., Lovegrove, P.C., 1983. Boiling heat transfer with Freon 11(R11) in brazed aluminum, plate-fin heat exchangers. Trans. ASME, J. Heat Transfer 105, 605–610.

Shah, M.M., 1982. Chart correlation for saturated boiling heat transfer: equations and further study. ASHRAE Trans. 88, 66–86.

Tran, T.N., 1998. Pressure drop and heat transfer study of two-phase flow in small channels. Ph.D. dissertation, Texas, Tech University.

Tran, T.N., Chyu, M.-C., Wambsganss, M.W., France, D.M., 2000. Two-phase pressure drop of refrigerants during flow boiling in small channels: an experimental investigation and correlation development. Int. J. Multiphase Flow 26, 1739–1754.

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.

Tran, T.N., Wambsganss, M.W., France, D.M., Jendrzejczyk, F.A., 1993. Boiling heat transfer in a small, horizontal, rectangular channel. AIChE Symp. Ser. 89 (295), 253–261.

Triplett, K.A., Ghiaasiaan, S.M., Abdel-Khalik, S.I., LeMouel, A., 1999. Gas–liquid two-phase flow in microchannels Part II: Void fraction and pressure drop. Int. J. Multiphase Flow 25, 395–410.

Wambsganss, M.W., France, D.M., Jendrzejczyk, F.A., Tran, T.N., 1993. Boiling heat transfer in a horizontal small-diameter tube. Trans. ASME, J. Heat Transfer 115, 963–972.

Wambsganss, M.W., Jendrzejczyk, J.A., France, D.M., Obot, N.T., 1992. Frictional pressure gradients in two-phase flow in a small horizontal rectangular channel. Exp. Thermal Fluid Sci. 5, 40–56.

Wattelet, J.P., 1995. Predicting boiling heat transfer in a small-diameter round tube using an asymptotic method. Proceedings of the Convective Flow Boiling, 377–382.

Zivi, S.M., 1964. Estimation of steady-state steam void-fraction by means of the principle of minimum entropy production. Trans. ASME, J. Heat Transfer 86, 247–252.