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TWO-PHASE FLOW BOILING HEAT TRANSFER OF R-410A AND R-134A IN HORIZONTAL SMALL TUBES

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ABSTRACT

Experimental investigation on two-phase flow boiling heat transfer of R-410A and R-134A in horizontal small tubes is reported. The pressure drop and local heat transfer coefficients were obtained over heat flux range of 5 to 40 kW/m^2, mass flux range of 70 to 600kg/ m^{2}s, saturation temperature range of 2 to 12°C, and quality up to 1.0 in test section with inner tube diameters of 3.0 and 0.5mm, and lengths of 2000 and 330mm, respectively. The section was heated uniformly by applying a direct electric current to the tubes. The effects of mass flux, heat flux, and inner tube diameter on pressure drop and heat transfer coefficients are presented. The experimental results are compared against several existing correlations. A new boiling heat transfer coefficient correlation based on the superposition model for refrigerants in small tubes is also presented.

Keywords: Heat transfer coefficient; Horizontal minichannel; Pressure drop; R-134A; R-410A; Two-phase vaporization

1. INTRODUCTION

The demand for refrigeration systems with smaller evaporators is increasing because of greater awareness of the advantages of process intensification. However, flow pattern, pressure drop and heat transfer for two-phase flow in small tubes cannot be properly predicted using the existing correlations that are intended to be applied for large tubes. Several studies dealing with two-phase flow heat transfer in small tubes, as reported in Zhang et al. (2004), Tran et al. (1996), and Mishima & Hibiki (1996), have been published in past years. However, the published studies did not present flow pattern, pressure drop and heat transfer coefficients all at once.

The flow pattern of the experimental data of this investigation was evaluated with published flow pattern maps in this study. The pressure drop and heat transfer coefficients of flow boiling of R-410A and R-134A in horizontal small tubes were measured. The effects of mass flux, heat flux, and tube diameter on pressure drop and heat transfer coefficients were presented.

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2. EXPERIMENTAL ASPECTS

The experimental facility, as shown in Figures 1(a) and (b), consisted of a condenser, a subcooler, a receiver, a pump, a mass flow meter, a preheater, and test sections. For the test with 3.0 mm tube, the flow rate was controlled with a variable A.C output motor controller, and a Coriolis-type mass flow meter was used to measure the refrigerant flow rate. For the test with 0.5 mm tube, a needle valve was used to control the flow rate of refrigerant, and a weighing balance was used to measure the refrigerant flow rate. The mass quality at the test section inlet was controlled by installing a preheater. For evaporation at the test section, a certain heat flux was conducted from a variable A.C voltage controller. The vapor refrigerant from the test section was condensed in the condenser, and then supplied to the receiver.

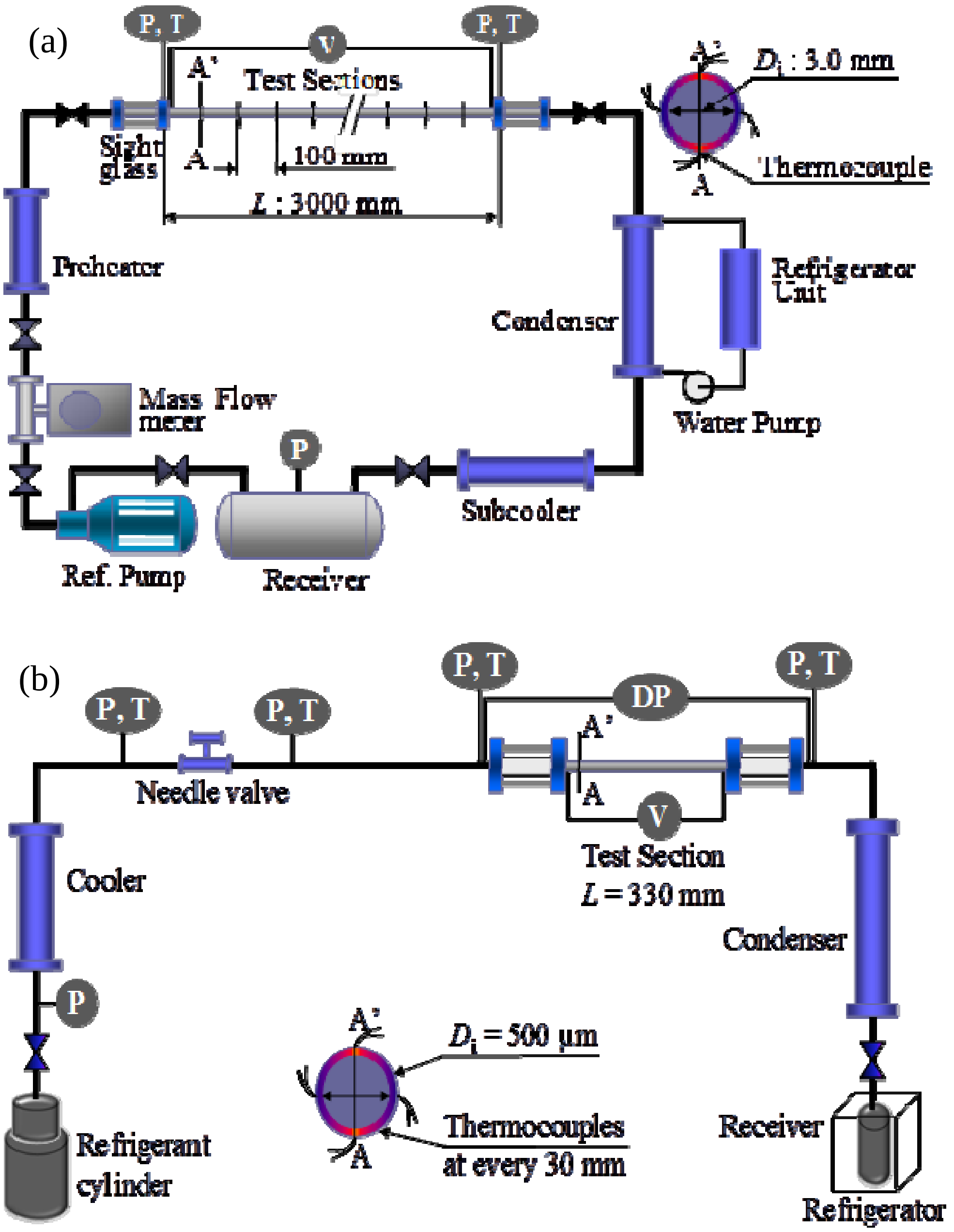

Figure 1 The experimental facility

The experimental test setup specifications were tabulated in Table 1. The local saturation pressure, which was used to determine the saturation temperature, was measured using bourdon tube type pressure gauges at the inlet and the outlet of the test section. The saturation pressure at the initial point of saturation was determined by interpolating the measured pressure and the calculated subcooled length. The subcooled length was calculated using Equation (1).

$$z_{sc} = L\frac{i_f - i_{fi}}{\Delta i} = L\frac{i_f - i_{fi}}{(Q/W)} \quad (1)$$

Where the z_{sc} is subcooled length, L is length of test section, i_f is saturated liquid enthalpy, i_{fi} is liquid enthalpy at the inlet of the test section, Q is electric power and W is mass flow rate.

Table 1 Experimental conditions

Working fluid	R-410A	R-134A
Mass flux (kg/㎡s)	70–600	100–600
Inlet T_{sat}(°C)	2 –12	5–10
Heat flux (kW/㎡)	5–40	
Tube length (mm)	0.5, 3.0	
Inner tube diameter (mm)	330, 2000	
Test section	Horizontal smooth minichannels	
Quality	0.0–1.0	

The outside tube wall temperatures at the top, both sides, and bottom of the test section were measured at certain axial intervals from the start of the heated length with thermocouples at each measured site. The tubes were well insulated with rubber and foam. The experimental two-phase frictional pressure drop can be obtained by subtracting the calculated accelerational pressure drop from the measured pressure drop.

The void fraction is predicted using the Steiner (1993) void fraction. In order to obtain the two-phase frictional multiplier based on the pressure drop for the total flow assumed for the liquid, the calculated two-phase frictional pressure drop is divided by the calculated frictional two-phase pressure drop assuming the total flow to be liquid.

$$\phi_f^2 = \left(-\frac{dp}{dz}F\right)_{tp} \Big/ \left(-\frac{dp}{dz}F\right)_f = \left(-\frac{dp}{dz}F\right)_{tp} \Big/ \left(\frac{2 f_f G^2}{D\rho_f}\right) \quad (2)$$

Where the $\emptyset_f{}^2$ is two-phase frictional multiplier, f_f is liquid friction factor, G is mass flux, D is diameter of test section and ρ_f is liquid density.

The inside tube wall temperature was determined by steady-state one-dimensional radial conduction heat transfer through the wall with internal heat generation. The vapor quality was determined based on the thermodynamic properties. The outlet mass quality was determined using Equation (3).

$$x_o = \frac{\Delta i + i_{fi} - i_f}{i_{fg}} \tag{3}$$

Where the x_0 is outlet mass quality, i_{fi} is liquid enthalpy at the inlet of the test section, i_f is saturated liquid enthalpy and i_{fg} is saturated latent heat enthalpy of vaporization.

3. RESULTS AND DISCUSSION

3.1. Pressure drop

Figure 2 shows that mass flux has a strong effect on the pressure drop. An increase in the mass flux results in a higher flow velocity, which increases the pressure drops. The figure also illustrates that the pressure drop increases as the heat flux increases. It is presumed that increasing heat flux results in a higher vaporization, which increases the average fluid vapor quality and flow velocity.

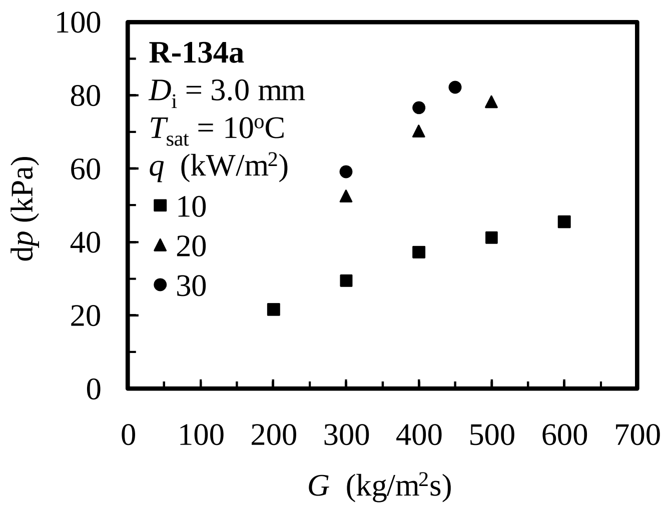

Figure 2 The effect of mass flux and heat flux on pressure drop

As shown in Figure 3, the pressure drop in the 0.5 mm tube is higher than that in the 3.0 mm tube. Therefore, it is assumed that a smaller tube diameter results in a higher wall shear stress, wherein for a given temperature condition it results in a higher friction factor and flow velocity, and then provides a greater pressure drop.

3.2. Heat transfer coefficient

Figure 4 shows that mass flux has an insignificant effect on the heat transfer coefficient in the low quality region. It indicates that nucleate boiling heat transfer is predominant. A higher mass flux corresponds to a higher heat transfer coefficient at intermediate/high vapor quality, due to an increase of the convective boiling heat transfer contribution. The steep decrease of the heat transfer coefficient at high qualities can be attributed to the effect of a small diameter on the boiling flow pattern because dry patch occurs more easily at a higher mass flux.

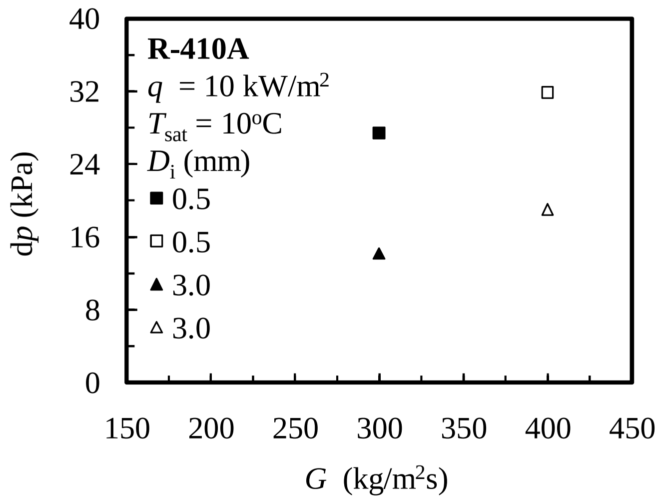

Figure 3 The effect of mass flux and heat flux on pressure drop

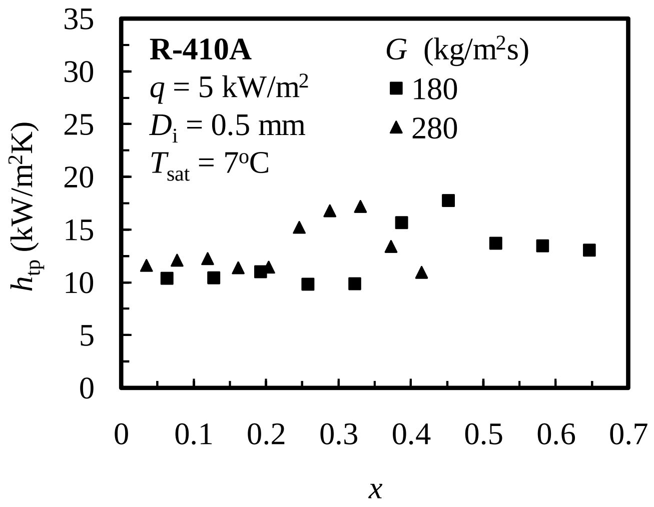

Figure 4 The effect of mass flux on heat transfer coefficient

Figure 5 depicts the dependence of heat flux on heat transfer coefficients in the low/intermediate quality region. At the quality of around 0.2, nucleate boiling is suppressed and convective heat transfer contribution is predominant, it is indicated by a low effect of heat flux on the heat transfer coefficient.

Figure 6 shows that a smaller inner tube has a higher heat transfer coefficient at low quality regions. As the tube diameter becomes smaller, the contact surface area for heat transfer increases, hence the nucleate boiling is more active. It then causes dry patches to appear earlier. The quality for a rapid decrease in the heat transfer coefficient is lower for the smaller tube. It is supposed that the annular flow appears at a lower quality in the smaller tube and therefore, the dry-out quality is relatively lower for the smaller tube.

The present heat transfer coefficients were compared with six existing correlations, as shown in Table 2. The Gungor-Winterton correlation (1987) provided the best prediction. This correlation was developed using some fluids in several small and conventional channels under various testing conditions. There was a large deviation in the prediction because the previous correlations failed to predict a higher nucleate boiling heat transfer contribution for evaporative refrigerants in small channels, as well as the appearance of laminar flow.

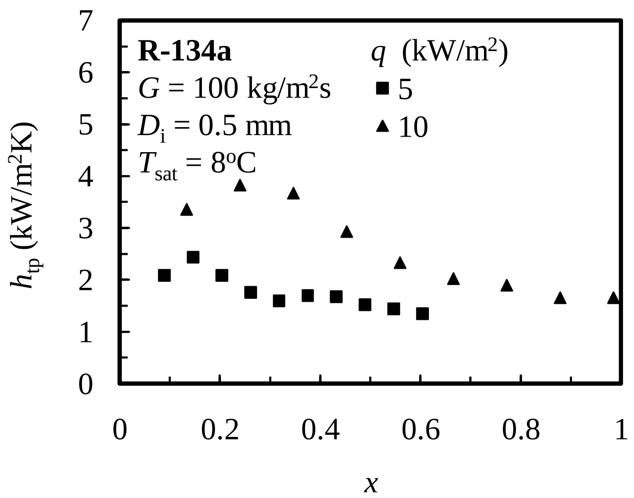

Figure 5 The effect of mass flux on heat transfer coefficient

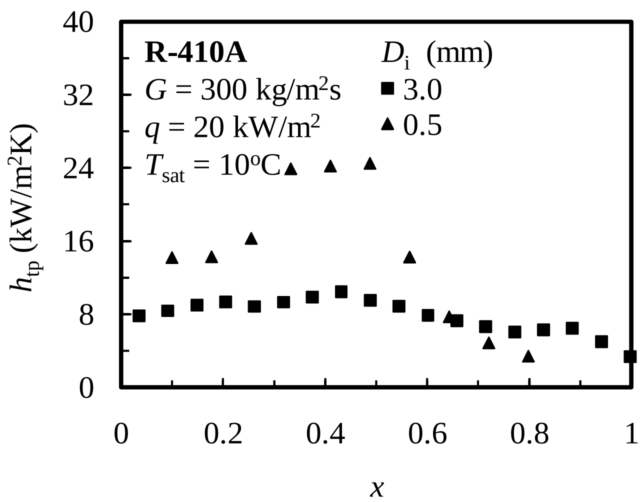

Figure 6 The effect of inner tube diameter on heat transfer coefficient

Table 2 Deviation of the pressure drop comparison between the experimental test data and previous correlations

Deviation(%)	Tran *et al.*	Shah	Chen	Gungor-Winterton	Wattelet	Jung *et al.*
Mean	30.79	32.80	35.39	38.92	39.87	44.94
Average	-0.26	8.26	-13.49	9.84	-26.37	-0.14

4. NEW CORRELATION DEVELOPMENT

A modification of pressure drop correlation was proposed on the basis of the Lockhart-Martinelli method (1967). The two-phase pressure drop of Lockhart-Martinelli consisted of the following three terms: the liquid phase pressure drop, the interaction between the liquid phase and the vapor phase, and the vapor phase pressure drop. The two-phase frictional multiplier based on the pressure gradient for liquid alone flow is calculated by Equation (4).

$$\phi_f^2 = \frac{\left(-\frac{dp}{dz}F\right)_{tp}}{\left(-\frac{dp}{dz}F\right)_f} = 1 + C\left[\frac{\left(-\frac{dp}{dz}F\right)_g}{\left(-\frac{dp}{dz}F\right)_f}\right]^{1/2} + \frac{\left(-\frac{dp}{dz}F\right)_g}{\left(-\frac{dp}{dz}F\right)_f} = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{4}$$

Where the $\emptyset_f{}^2$ is two-phase frictional multiplier, C is Chisholm parameter and X is Martinelli parameter. The value of C is found by an interpolation of the Chisholm parameter (1967). For the liquid-vapor flow condition of turbulent-turbulent(tt), laminar-turbulent(vt), turbulent-laminar(tv) and laminar-laminar(vv), the values of the C parameters are 20, 12, 10, and 5, respectively. The Martinelli parameter (1967), X, is defined by the Equation (5).

$$X = \left[\frac{\left(-\frac{dp}{dz}F\right)_f}{\left(-\frac{dp}{dz}F\right)_g}\right]^{1/2} = \left[\frac{2f_f G^2(1-x)^2 \rho_g / D}{2f_g G^2 x^2 \rho_f / D}\right]^{1/2} = \left(\frac{f_f}{f_g}\right)^{1/2}\left(\frac{1-x}{x}\right)\left(\frac{\rho_g}{\rho_f}\right)^{1/2} \tag{5}$$

Where the X is Martinelli parameter, f_f is liquid friction factor, G is mass flux, ρ_g is vapor density, D is diameter of test section, f_g is vapor friction factor, ρ_f is liquid density and x is mass quality.

The friction factor was obtained by considering the flow conditions of laminar (for Re < 2300, f=16Re-1) and turbulent (for Re>3000, f=0.079Re-0.25). It is well known that the flow boiling heat transfer is mainly governed by the following two important mechanisms: nucleate boiling and forced convective evaporation. Chen (1966) introduced a multiplier factor, F=fn(X_{tt}), to account for the increase in the convective turbulence that is due to the presence of the vapor phase. The function should be physically evaluated again for flow boiling heat transfer in a minichannel that has a laminar flow condition, which is due to the small diameter effect. The F factor in this study is developed as a function of $\emptyset_f{}^2$, F=fn($\emptyset_f{}^2$), where ϕ_f^2 is obtained from Equations (4) and (5). Liquid heat transfer is defined by the Dittus Boelter correlation(1930), and a new factor F (shown in Figure 7), is developed using a regression method. The prediction of the nucleate boiling heat transfer used Cooper (1984). A new nucleate boiling suppression factor, as a ratio of h_{nbc}/h_{nb}, is proposed as follows:

$$S = 2.3\left(\phi_f^2\right)^{-0.127} Bo^{0.066} \tag{6}$$

Where the S is suppression factor of nucleate boiling, $\emptyset_f{}^2$ is two-phase frictional multiplier and B_o is boiling number.

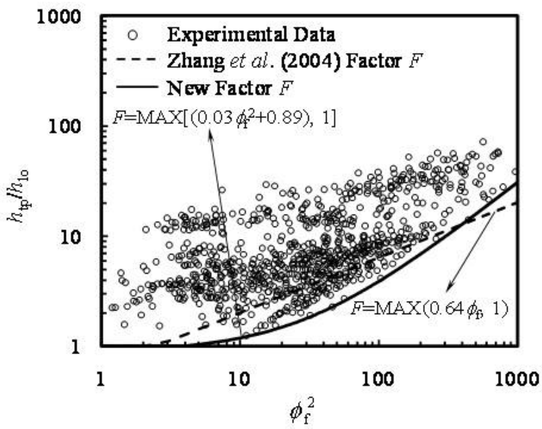

Figure 7 Two-phase heat transfer multiplier as a function of $\emptyset_f{}^2$

5. CONCLUSION

Convective boiling pressure drop and heat transfer experiments were performed in horizontal minichannels with R-410A and R-134A. The pressure drop was higher for the conditions of higher mass and heat fluxes, and for the conditions of smaller inner tube diameter. Mass flux, heat flux, inner tube diameter, and saturation temperature clearly affected the heat transfer coefficient. In our experiments, the heat transfer coefficient increased with a decreased inner tube diameter and with an increased saturation temperature. The geometric effect of the small tube must be considered to develop a new heat transfer coefficient correlation. Laminar flow appeared for flow boiling in small channels, so the modified correlation of the multiplier factor for the convective boiling contribution, F, and the nucleate boiling suppression factor, S, was developed using laminar and turbulent flow considerations.

6. ACKNOWLEDGEMENT

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7. REFERENCES

Chen, J.C., 1966. A Correlation for Boiling Heat Transfer to Saturated Fluids in Convective Flow. *Industrial and Engineering Chemistry*, Process Des. Dev. 5, pp. 322-329.

Chisholm, D., 1967. A Theoretical Basis for the Lockhart-Martinelli Correlation for Two-Phase Flow. *International Journal Heat Mass Transfer*, Vol. 10, pp. 1767-1778.

Cooper, M.G., 1984. Heat Flow Rates in Saturated Nucleate Pool Boiling a Wide-Ranging Examination using Reduced Properties. *In: Advances in Heat Transfer*, Vol. 16, Academic Press, pp.157-239.

Dittus, F. W., Boelter, L. M. K., 1930. *Heat Transfer in Automobile Radiators of the Tubular Type*. University of California Publication in Engineering 2, pp. 443-461.

Gungor, K. E., Winterton, H. S., 1987. Simplified General Correlation for Saturated Flow Boiling and Comparisons of Correlations with Data. *Chemical Engineering Res*., Vol. 65, pp.148-156, Elsevier.

Mishima, K., Hibiki, T., 1996. Some Characteristics of Air-Water Two-Phase Flow in Small Diameter Vertical Tubes. *International Journal Multiphase Flow*, Vol. 22, pp.703-712, Elsevier.

Steiner, D., 1993. Heat Transfer to Boiling Saturated Liquids. VDI-Wärmeatlas (VDI Heat Atlas), Verein Deutcher Ingenieure, ed., VDI-Gessellschaft Verfahrenstechnik und Chemieinge- nieurwesen (GCV), *Düsseldorf*, Germany, (J.W. Fullarton, translator).

Tran, T. N., Wambsganss, M. W., France, D.M., 1996. Small Circular- and Rectangular-channel Boiling with Two Refrigerants. *International Journal Multiphase Flow*, Vol. 22, Number 3, pp. 485-498, Elsevier.

Zhang, W., Hibiki, T., Mishima, K., 2004. Correlation for Flow Boiling Heat Transfer in Mini-channels, *International Journal Heat and Mass Transfer*, Vol. 47, pp. 5749-5763, Elsevier.