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FLOW BOILING HEAT TRANSFER AND PRESSURE DROP OF PROPANE IN SMOOTH HORIZONTAL MINICHANNELS

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ABSTRACT

This study examined convective boiling heat transfer and pressure drop in horizontal minichannels using propane. The local heat transfer coefficients and pressure drop were obtained for heat fluxes ranging from 5-20 kW m^{-2}, mass fluxes ranging from 50-400 kg m^{-2} s^{-1}, saturation temperatures of 10, 5, and 0°C, and quality up to 1.0. The test section was made of stainless steel tubes with inner diameters of 1.5 mm and 3.0 mm, and lengths of 1000 mm and 2000 mm, respectively. The section was heated uniformly by applying an electric current to the tubes directly. The present study showed an effect of mass flux, heat flux, inner tube diameter, and saturation temperature on heat transfer coefficient and pressure drop. The experimental results were compared against several two-phase heat transfer coefficient and pressure drop prediction methods. A new boiling heat transfer coefficient correlation based on the superposition model for propane in minichannels was developed with a mean deviation of 8.27%.

INTRODUCTION

Several studies on natural refrigerants have been devoted as an environmental protection effort. Recent awareness of the advantages of process intensification has also led to a demand for smaller evaporators for use in the refrigeration and air conditioning and processing industries. However, heat transfer for two-phase flows in small channels cannot be properly predicted using existing procedures and correlations that are intended for large channels. There is a small quantity of published data that relates to two-phase flow heat transfer and pressure drop for propane in small channels compared with the data for conventional refrigerants in large channels.

This study was undertaken to obtain experimental data for propane and to determine their local heat transfer coefficient and pressure drop during evaporation in minichannels. The results were compared with several existing heat transfer coefficient and pressure drop correlations, and a new correlation of heat transfer coefficient for propane in minichannels is developed in this study.

EXPERIMENTAL APPARATUS AND METHOD

The experimental facility is schematically shown in Figure 1. The flow rate of the refrigerant was controlled by a variable

NOMENCLATURE

a	[-]	Accelerational contribution
Bo	[-]	Boiling number
C	[-]	Chisholm parameter
D	[m]	Diameter
F	[-]	Convective two-phase multiplier, Eq. (10)
F	[-]	Frictional contribution, Eqs. (4), (5) and (7)
f	[-]	Friction factor
fn	[-]	Function
G	[kg/m^{2}s]	Mass flux
h	[kW/m^{2}K]	Heat transfer coefficient
i	[kJ/kg]	Enthalpy
L	[m]	Length
M	[kg/kmol]	Molecular weight
P	[kPa]	Pressure
Q	[kW]	Electric power
q	[kW/m^2]	Heat flux
Re	[-]	Reynolds number
S	[-]	Nucleate boiling suppression factor
T	[K]	Temperature
υ	[m^3/kg]	Specific volume
W	[kg/s]	Mass flow
X	[-]	Martinelli parameter
x	[-]	Mass quality
z	[m]	Axial coordinate

Special characters

α	[-]	Void fraction
μ	[Ns/m^2]	Dynamic viscosity
ρ	[kg/m^3]	Density
σ	[N/m]	Surface tension
ϕ^2	[-]	Two-phase frictional multiplier
(dp/dz)	[N/m^2 m]	Pressure gradient

Subscripts

exp	Experimental value
f	Saturated liquid
fo	Liquid only
g	Saturated vapour
i	Inner tube
in	Inlet of the test section
nb	Nucleate pool boiling
nbc	Nucleate boiling contribution
o	Outlet of the test section
pred	Prediction value
r	Reduced
sat	Saturation
sc	Subcooled
t	Turbulent
tp	Two-phase
v	Laminar
w	Wall

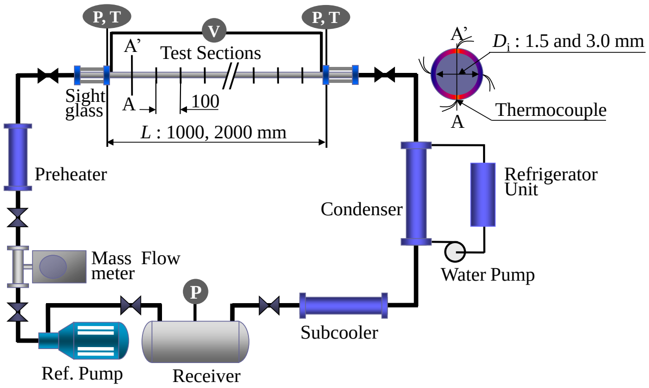

Figure 1 Experimental test facility

AC output motor controller. A Coriolis-type mass flow meter was used to measure the refrigerant flow rate. To control mass quality at the test section inlet, a preheater was installed. Then, the vapor refrigerant from the test section was condensed in the condenser and subcooler, and then it was supplied to the receiver.

The test sections were uniformly and constantly heated by applying an electric current directly to their tube walls, and were insulated well. The outside tube wall temperatures at the top, both sides and bottom were measured at 100 mm axial intervals from the start of the heated length using thermocouples at each site that was measured. The local saturation pressure was measured at the inlet and the outlet of the test section. Sight glasses with the same inner diameter as the test section were installed to visualize the flow. The experimental test setup specifications that were used in this study are listed in Table 1.

The inside tube wall temperature, T_{wi} was the average temperature of the top, both right and left sides, and bottom wall temperatures, and was determined using steady-state one-dimensional radial conduction heat transfer through the wall with internal heat generation. The quality, x, at the measurement locations, z, were determined based on the thermodynamic properties

$$x = \frac{i - i_{\mathrm{f}}}{i_{\mathrm{fg}}} \tag{1}$$

The refrigerant flow at the inlet of the test section was not completely saturated. Even though it was just short, it was necessary to determine the subcooled length for reduction data accuracy. The subcooled length was calculated using the following equation to determine the initial point of saturation.

$$z_{\mathrm{sc}} = L\frac{i_{\mathrm{f}} - i_{\mathrm{f,in}}}{\Delta i} = L\frac{i_{\mathrm{f}} - i_{\mathrm{f,in}}}{(Q/W)} \tag{2}$$

The outlet mass quality was then determined using the following equation:

$$x_{\mathrm{o}} = \frac{\Delta i + i_{\mathrm{f,in}} - i_{\mathrm{f}}}{i_{\mathrm{fg}}} \tag{3}$$

The saturation pressure at the initial point of saturation was then determined by interpolating the measured pressure and the subcooled length. The experimental two-phase frictional pressure drop can be obtained by subtracting the calculated accelerational pressure drop from the measured pressure drop.

Table 1 Experimental conditions

Working refrigerant	Propane
Test section	Horizontal smooth stainless steel minichannels
Inner tube diameter (mm)	1.5, 3.0
Quality	0.0 – 1.0
Tube length (mm)	1000, 2000
Mass flux (kg/m^{2}s)	50 – 400
Heat flux (kW/m^2)	5 – 20
Inlet T_{sat} (ºC)	10, 5, 0

$$\begin{aligned}\left(-\frac{\mathrm{d}p}{\mathrm{dz}}F\right) &= \left(-\frac{\mathrm{d}p}{\mathrm{dz}}\right) - \left(-\frac{\mathrm{d}p}{\mathrm{dz}}a\right) \\ &= \left(-\frac{\mathrm{d}p}{\mathrm{dz}}\right) - G^2\frac{\mathrm{d}}{\mathrm{dz}}\left(\frac{x^2\upsilon_{\mathrm{g}}}{\alpha} + \frac{(1-x)^2\upsilon_{\mathrm{f}}}{(1-\alpha)}\right)\end{aligned} \tag{4}$$

In the present study, the void fraction is obtained from Steiner [1]. The friction factor was determined from the measured pressure drop for a given mass flux by using the Fanning equation:

$$f_{\mathrm{tp}} = \frac{D\bar{\rho}}{2G^2}\left(-\frac{\mathrm{d}p}{\mathrm{dz}}F\right) \tag{5}$$

where the average density is calculated with the equation:

$$\frac{1}{\bar{\rho}} = \frac{x}{\rho_{\mathrm{g}}} + \frac{1-x}{\rho_{\mathrm{f}}} \tag{6}$$

In order to obtain the two-phase frictional multiplier based on pressure drop for the total flow assumed liquid ϕ_{fo}^2, the calculated two-phase frictional pressure drop is divided by the calculated frictional two-phase pressure drop assuming total flow to be liquid.

$$\phi_{\mathrm{fo}}^2 = \left(-\frac{\mathrm{d}p}{\mathrm{dz}}F\right)_{\mathrm{tp}} \Bigg/ \left(-\frac{\mathrm{d}p}{\mathrm{dz}}F\right)_{\mathrm{fo}} = \left(-\frac{\mathrm{d}p}{\mathrm{dz}}F\right)_{\mathrm{tp}} \Bigg/ \left(\frac{2f_{\mathrm{fo}}G^2}{D\rho_{\mathrm{f}}}\right) \tag{7}$$

The friction factor in equation (7) is obtained using the Blasius equation.

RESULTS AND DISCUSSION

Heat Transfer Coefficient

Figure 2 shows the effect of mass flux, heat flux, inner tube diameter, and saturation temperature on heat transfer coefficient. Mass flux has an insignificant effect on heat transfer coefficient at low quality region. The insignificant effect of mass flux on heat transfer coefficient means that nucleate boiling heat transfer is predominant. Several previous studies using small tubes that were performed by Kew and Cornwell [2], Lazarek and Black [3], Wambsganss *et al*. [4], Tran *et al*. [5] and Bao *et al*. [6] showed that, in small channels, nucleate boiling is predominant. The high nucleate boiling heat transfer occurs because of the physical properties of the

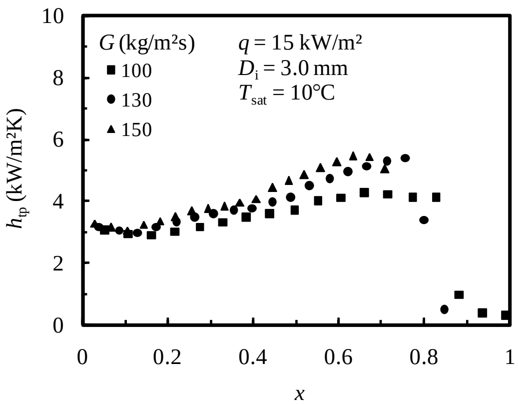

Figure 2 The effect of mass flux on heat transfer coefficient

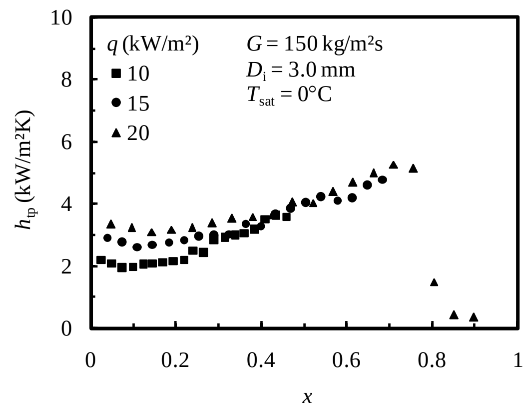

Figure 3 The effect of heat flux on heat transfer coefficient

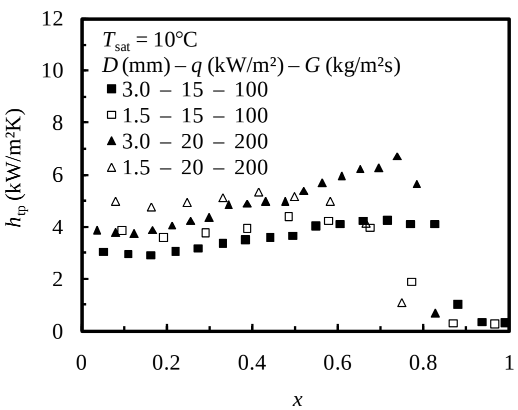

Figure 4 The effect of inner tube diameter on heat transfer coefficient

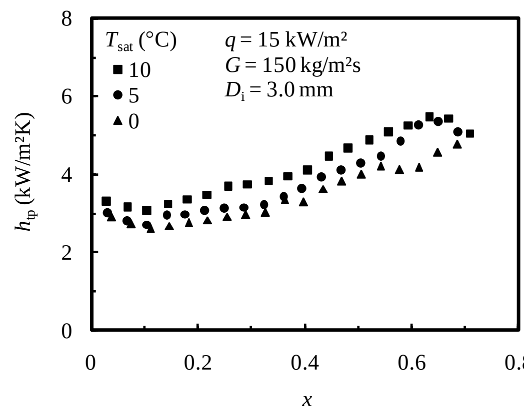

Figure 5 The effect of saturation temperature on heat transfer coefficient

Table 2 Deviation of the heat transfer coefficient comparison between the present data and the previous correlations.

Deviation (%)	Shah [7]	Tran *et al*. [5]	Jung *et al*. [8]	Gungor-Winterton [9]	Takamatsu *et al*. [10]	Kandlikar-Steinke [11]	Chen [12]
Mean Deviation	15.84	18.26	20.38	21.22	23.55	25.92	36.00
Average Deviation	-0.59	-9.82	19.70	16.79	22.52	16.70	17.73

Table 3 Deviation of the pressure drop comparison between the present data and the previous correlations.

Deviation (%)	Dukler *et al*. [14]	McAdams [15]	Beattie and Whalley [16]	Cichitti *et al*. [17]	Kawahara *et al*. [18]	Zhang *et al*. [19]	Friedel [20]
Mean Deviation	17.34	17.53	20.34	23.99	30.03	34.57	67.72
Average Deviation	-0.83	0.27	13.65	15.61	-29.66	22.41	67.32

refrigerants, namely surface tension and pressure, and the geometric effect of small channels. Higher mass flux is corresponding to the higher heat transfer coefficient at moderate-high vapor quality due to the increase of convective boiling heat transfer contribution. At the high quality region, a drop of heat transfer coefficient occurs at a lower quality for a relatively higher mass flux. The steep decreasing of heat transfer coefficient at high quality is due to the effect of small diameter on boiling flow pattern wherein dry-patch easier to occur in smaller diameter tube and higher mass flux.

Figure 3 depicts a dependence on heat transfer coefficients for heat flux appears in the low-moderate quality region. The high effect of heat flux on heat transfer coefficient shows a domination of the nucleate boiling heat transfer contribution. Nucleate boiling is suppressed at high quality. As the heat flux increases, the evaporation is more active and the dry-out quality becomes lower.

Figure 4 shows that, at low quality region, smaller inner tube diameter shows higher heat transfer coefficient. This is due to a more active nucleate boiling in a smaller diameter tube. As the tube diameter smaller, the contact surface area of heat transfer increases. The more active nucleate boiling causes dry-patches to appear earlier. The quality for rapid decrease in 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. The dry-out quality is relatively lower for the smaller tube.

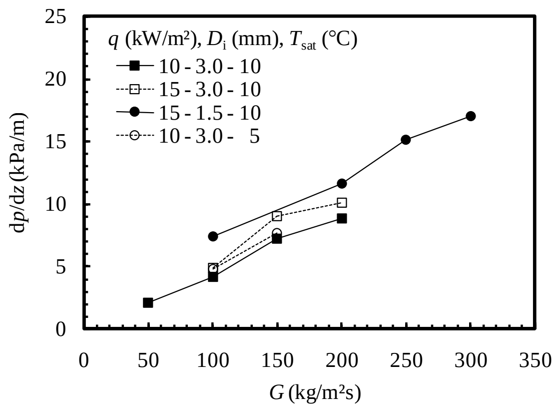

Figure 6 The effect of mass flux, heat flux, tube diameter, and saturation temperature on heat transfer coefficient

The effect of saturation temperature on heat transfer coefficient is depicted in Figure 5. The heat transfer coefficient increases with an increase in saturation temperature, which is due to a more active nucleate boiling.

The pre-dryout heat transfer coefficients of the present study were compared with seven correlations for boiling heat transfer coefficient as shown in Table 2. For the overall data, Shah's [7] correlation gave the best prediction among the others. Shah's [7] correlation was developed using conventional refrigerant in a conventional channel. The correlation of Tran *et al*. [5] was developed for flow boiling heat transfer in small channel also work well with the present experimental data.

Pressure Drop

Figure 6 shows mass flux has a strong effect on the pressure drop. Increasing mass flux results in a higher flow velocity, and then it increases the frictional and accelerational pressure drops.

Figure 6 also illustrates that pressure drop increases with heat flux increases. It is presumed that the increasing heat flux result in a higher vaporization, and then it increases the average fluid vapor quality and flow velocity; this trend is similar to that shown by Zhao *et al*. [13].

The effect of the inner tube diameter on the pressure drop is also illustrated in the Figure 6. The pressure gradient in the 1.5 mm tube is higher than that in the 3.0 mm tube. The explanation is that the smaller inner tube diameter results in a higher wall shear stress, wherein for a given temperature test condition it results in a higher friction factor and flow velocity, and then results in higher frictional and accelerational pressure drops.

Figure 6 depicts the effect of saturation temperature on pressure drop. The lower saturation temperature results in a higher pressure drop. This can be explained by the effect of the physical properties of density and viscosity on pressure drop at different temperatures. The liquid density, ρ_f, and liquid viscosity, μ_f, increase with the temperature decrease, whereas the vapor density, ρ_g, and vapor viscosity, μ_g, decrease with the temperature decrease. As the temperature decreases for a constant mass flux condition, the increasing liquid density and liquid viscosity result in a lower liquid velocity, whereas the decreasing vapor density and vapor viscosity result in a higher vapor velocity. It is clear that during evaporation the pressure drop increases, and the increasing of the pressure drop is higher for a condition of lower saturation temperature. The higher vapor velocity, due to the decreasing of saturation temperature, also causes a higher entrainment, and further it may cause dry-out to occur earlier.

The experimental two-phase pressure drop data were compared with several existing correlations, as shown in Table 3. For the overall data, the homogeneous model of Dukler *et al*. [14] gave the best prediction among the other methods. The homogeneous model assumed equal vapor and liquid velocities and that the mixture was defined as a single phase with average fluid properties.

DEVELOPMENT OF A NEW CORRELATION

Modification of Factor *F*

As well known, flow boiling heat transfer is governed mainly by two important mechanisms, namely: nucleate boiling and forced convective evaporation. The appearance of convective heat transfer for boiling in small channels is later than it is in large channels because of its high boiling nucleation. The new heat transfer coefficient correlation in this study is developed with only using the experimental data that is prior to the dry-out. Chen [12] 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. By considering the flow conditions (laminar or turbulent) in the Reynolds number factor, F, Zhang *et al*. [21] introduced a relationship between the factor F and the two-phase frictional multiplier that is based on pressure gradient for liquid alone flow, ϕ_f^2, F=fn(ϕ_f^2), where ϕ_f^2 is a general form for four conditions according to Chisholm [22]

$$\phi_f^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{8}$$

For liquid-vapor flow condition of turbulent-turbulent (tt), laminar-turbulent (vt), turbulent-laminar (tv) and laminar-laminar (vv), the values of the Chisholm parameter, C, are 20, 12, 10, and 5, respectively. The value of C in this study is found by an interpolation of the Chisholm parameter, C, with thresholds of Re=1000 and Re=2000 for the laminar and turbulent flows, respectively.

On this study, the Blasius equation of friction factor is used for the friction factors, f_f and f_g, then the Martinelli parameter can be rewritten as

$$X = \left(\frac{f_f}{f_g}\right)^{1/2} \left(\frac{1-x}{x}\right) \left(\frac{\rho_g}{\rho_f}\right)^{1/2} = \left(\frac{\mu_f}{\mu_g}\right)^{1/8} \left(\frac{1-x}{x}\right)^{7/8} \left(\frac{\rho_g}{\rho_f}\right)^{1/2} \tag{9}$$

The liquid heat transfer is defined by the Dittus Boelter correlation. A new factor F, as shown in Figure 7, is developed using a regression method that was applied to the experimental data.

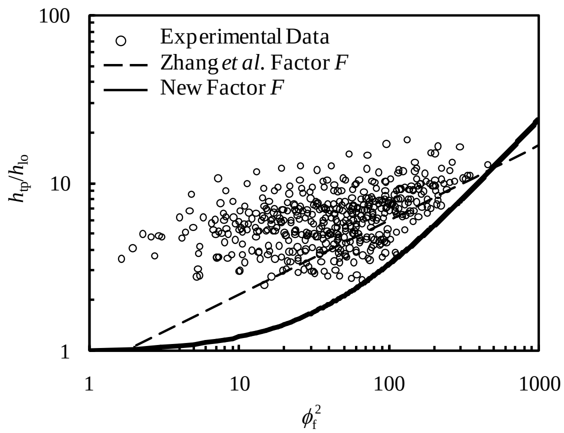

Figure 7 Two-phase heat transfer multiplier as a function of ϕ_f^2

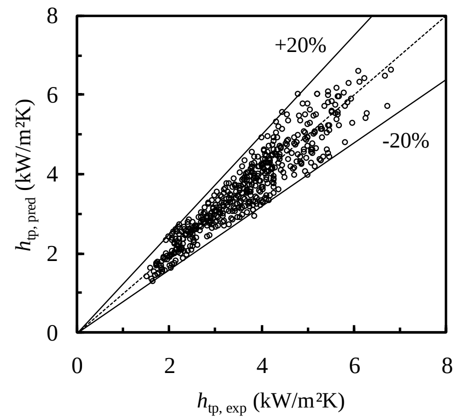

Figure 8 Comparison of the experimental and predicted heat transfer coefficients using the newly correlation

$$F = 0.023\phi_f^2 + 0.977 \tag{10}$$

Nucleate Boiling Contribution

The present study shows that surface tension, density ratio, ρ_f/ρ_g, viscosity ratio, μ_f/μ_g, mass flux and saturation temperature have a strong effect on the nucleate boiling heat transfer contribution. For evaporation in a small channel, the suppression is lower than that in a conventional channel. The prediction of the nucleate boiling heat transfer for the present experimental data used Cooper [23]. For a surface roughness that is set equal to 1.0 µm, his correlation is given as:

$$h = 55P_r^{0.12}\left(-0.4343\ln P_r\right)^{-0.55}M^{-0.5}q^{0.67} \tag{11}$$

where the heat flux, q, is in W m^{-2}.

Chen [12] defined the nucleate boiling suppression factor, S, as a ratio of the mean superheat, ΔT_e, to the wall superheat, ΔT_{sat}. Jung *et al*. [8] proposed a convective boiling heat transfer multiplier factor, N, as a function of quality, heat flux and mass flow rate (represented by employing X_{tt} and Bo) to represent the strong effect of nucleate boiling in flow boiling as it is compared with that in nucleate pool boiling, h_{nbc}/h_{nb}. To consider laminar flow in minichannels, the Martinelli parameter, X_{tt}, is replaced by a two-phase frictional multiplier, ϕ_f^2. By using the experimental data of this study, a new nucleate boiling suppression factor, as a ratio of h_{nbc}/h_{nb}, is proposed as follows

$$S = 0.6226\left(\phi_f^2\right)^{0.1068}Bo^{0.0777} \tag{12}$$

Heat transfer Coefficient Comparison

The new heat transfer coefficient correlation is developed using a regression method with 479 data points. The comparison of the experimental heat transfer coefficient, $h_{tp,\ exp}$, and the predicted heat transfer coefficient, $h_{tp,\ pred}$, is illustrated in Figure 8. The new correlation shows a good agreement on the comparison with a mean deviation of 8.27% and an average deviation of -0.01%.

CONCLUDING REMARKS

Convective boiling heat transfer and pressure drop experiments were performed in horizontal minichannels with propane. Mass flux, heat flux, inner tube diameter, and saturation temperature have an effect on heat transfer coefficient and pressure drop. The geometric effect of the small channels contributed to the higher boiling nucleation and pressure drop.

The geometric effect of small tube must be considered to develop a new heat transfer coefficient correlation. Laminar flow appears 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, are developed in this study using a laminar flow consideration. A new boiling heat transfer coefficient correlation that is based on a superposition model for refrigerants in minichannels was presented with 8.27% mean deviation and -0.01% average deviation.

REFERENCES

[1] Steiner D., 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), 1993.

[2] Kew P.A., Cornwell K., Correlations for the prediction of boiling heat transfer in small-diameter channels, *Applied Thermal Engineering*, Vol. 17(8-10), 1997, pp. 705-715.

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

[4] Wambsganss M.W., France D.M., Jendrzejczyk J.A., Tran T.N., Boiling heat transfer in a horizontal small-diameter tube, *J. of Heat Transfer*, Vol. 115, 1993, pp. 963-975.

[5] Tran T.N., Wambsganss M.W., France D.M., Small circular- and rectangular-channel boiling with two refrigerants, *Int. J. Multiphase Flow*, Vol. 22(3), 1996, pp. 485-498.

[6] Bao Z.Y., Fletcher D.F., Haynes B.S., Flow boiling heat transfer of freon R11 and HCFC123 in narrow passages, *Int. J. Heat and Mass Transfer*, 43, 2000, pp. 3347-3358.

[7] Shah M.M., Chart correlation for saturated boiling heat transfer: equations and further study, *ASHRAE Trans* 2673, 1988, pp. 185-196.

[8] Jung D.S., McLinden M., Radermacher R., Didion D., A study of flow boiling heat transfer with refrigerant mixtures, *Int. J. Heat Mass Transfer*, Vol. 32(9), 1989, pp. 1751-1764.

[9] Gungor K.E., Winterton H.S., Simplified general correlation for saturated flow boiling and comparisons of correlations with data. *Chem Eng Res*, Vol. 65, 1987, pp. 148-156.

[10] Takamatsu H., Momoki S., Fujii T., A correlation for forced convective boiling heat transfer of a nonazeotropic refrigerant mixture of HCFC22/CFC114 in a horizontal smooth tube, *Int J Heat Mass Transfer*, Vol. 36(14), 1993, 3555-3563.

[11] Kandlikar S.G., Steinke M.E., Predicting heat transfer during flow boiling in minichannels and microchannels, *ASHRAE Trans* CH-03-13-1, 2003, pp. 667-676.

[12] Chen J.C., A correlation for boiling heat transfer to saturated fluids in convective flow, *Industrial and Engineering Chemistry, Process Design and Development* 5, 1966, pp. 322-329.

[13] Zhao Y., Molki M., Ohadi M.M., Dessiatoun S.V., Flow Boiling of CO_2 in Microchannels, *ASHRAE Trans* DA-00-2-1, 2000, pp. 437-445.

[14] Dukler A.E., Wicks I.I.I.M., Cleveland R.G., Pressure rop and Hold-up in Two-phase Flow, *AIChE J*., Vol. 10(1), 1964, pp. 38-51.

[15] McAdams W.H., Heat Transmission, third ed., McGraw-Hill, New York, 1954.

[16] Beattie D.R.H., Whalley P.B., A Simple Two-phase Flow Frictional Pressure Drop Calculation Method, *Int. J. Multiphase Flow*, Vol. 8, 1982, pp. 83-87.

[17] Cicchitti A., Lombardi C., Silvestri M., Solddaini G., Zavalluilli R., Two-phase Cooling Experiments— Pressure Drop, *Heat Transfer and Burnout Measurement, Energia Nucl.*, Vol. 7(6), 1960, pp. 407-425.

[18] Kawahara A., Chung P.M.Y., Kawaji M., Investigation of two-phase flow pattern, void fraction and pressure drop in a minichannel, *Int. J. Multiphase Flow*, Vol. 28, 2002, pp. 1411-1435.

[19] Zhang M. and Webb R.L., Correlation of Two-phase Friction for Refrigerants in Small-diameter Tubes, *Exp. Thermal and Fluid Science*, Vol. 25, 2001, pp. 131-139.

[20] Friedel L., Improved Friction Pressure Drop Correlations for Horizontal and Vertical Two-phase Pipe Flow, *In the European Two-phase Flow Group Meeting*, Paper E2, Ispra, Italy, 1979.

[21] Zhang W., Hibiki T., Mishima K., Correlation for flow boiling heat transfer in mini-channels, *Int. J. Heat and Mass Transfer*, Vol. 47, 2004, pp. 5749-5763.

[22] Chisholm D., A theoretical basis for the Lockhart-Martinelli correlation for two-phase flow, *Int. J. Heat Mass Transfer*, Vol. 10, 1967, pp. 1767-1778.

[23] Cooper M.G., Heat flow rates in saturated nucleate pool boiling– a wide-ranging examination using reduced properties, *In: Advances in Heat Transfer. Academic Press*, Vol. 16, 1984, pp. 157-239.