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Two-phase flow heat transfer of propane vaporization in horizontal minichannels †

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Abstract

Experiments were performed on the convective boiling heat transfer in horizontal minichannels using propane. 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, and it was uniformly heated by applying an electric current directly to the tubes. Local heat transfer coefficients were obtained for a heat flux range of 5-20 kW m^{-2}, a mass flux range of 50-400 kg $m^{-2} s^{-1}$, saturation temperatures of 10, 5, and 0°C and quality ranges of up to 1.0. The nucleate boiling heat transfer contribution was predominant, particularly at the low quality region. Decreases in the heat transfer coefficient occurred at a lower vapor quality with a rise of heat flux and mass flux, and with a lower saturation temperature and inner tube diameter. Laminar flow appeared in the minichannel flows. A new boiling heat transfer coefficient correlation that is based on the superposition model for propane was developed with 8.27% mean deviation.

Keywords: Boiling; Heat transfer; Horizontal tube; Microchannel; Propane

1. Introduction

Several studies have been devoted to natural refrigerants as an environmental protection effort. Most HCFCs' chemicals break down before reaching the ozone layer, with the chlorine produced reaching the stratosphere to deplete the ozone layer. In finding a replacement for HCFCs, there have been some studies on natural refrigerants, such as propane, as an environmental conservation effort because they do not contain chlorine. 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. Smaller channels of a heat exchanger give a higher heat transfer coefficient because of its larger heat transfer contact surface. However, heat transfer for two-phase flows in small channels cannot be properly predicted by using existing procedures and correlations intended for large channels. There is a small quantity of published data that relates to two-phase flow and heat transfer for propane in small channels compared with that in large channels. Evaporation using a smaller channel may provide a higher heat transfer coefficient due to its higher contact area per unit volume of fluid. In small channels, as shown in Bao et al. [1], Zhang et al. [2], Tran et al. [3], Kandlikar-Steinke [4] and Pamitran et al. [5] the contribution of nucleate boiling is predominant and laminar flow appears.

This study was undertaken to obtain experimental data for propane and to determine the local heat transfer coefficient during evaporation by using some test conditions in minichannels. The experimental results

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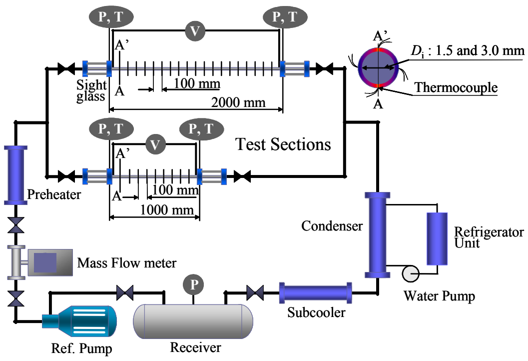

Fig. 1. The experimental test facility and test section.

were compared with several existing heat transfer correlations. A new correlation for propane in minichannels that is based on the superposition model is developed in this study because the correlation for forced convective boiling in small channels is limited.

2. Experimental aspects

2.1 Experimental apparatus and method

The experimental facility consisted of a condenser, a subcooler, a receiver, a pump, a mass flow meter, a preheater, and test sections (Fig. 1). A variable AC output motor controller was used to control the flow rate of the refrigerant. A coriolis-type mass flow meter 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 AC voltage controller. The vapor refrigerant from the test section was condensed in the condenser and subcooler, and then supplied to the receiver.

The test sections were comprised of stainless steel smooth tubes with inner diameters of 1.5 and 3.0 mm and heated lengths of 1000 mm and 2000 mm, respectively. The tubes were well insulated with rubber and foam. 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 with thermocouples at each measured site. The test sections were heated uniformly and constantly by applying an electric current directly to their tube walls. The input electric voltage and current were adjusted in order to control the input power. The local saturation pressure, which was used to determine the saturation temperature, was measured with Bourdon tube type pressure gauges at the inlet and at the outlet of the test section, as shown in Fig. 1. Sight glasses with the same inner diameter as the test section were installed in order to visualize the flow.

Table 1. Experimental conditions.

Working refrigerant	Propane
Test section	Horizontal smooth minichannels
Inner diameter (mm)	1.5, 3.0
Tube length (mm)	1000, 2000
Mass flux (kg m^{-2} s^{-1})	50 – 400
Heat flux (kW m^{-2})	5 – 20
Quality	0.0 – 1.0
Inlet T_{sat} (°C)	10, 5, 0

Table 2. Summary of estimated uncertainty.

Parameters	Uncertainty
Inner wall temperature	±1.45%
Absolute pressure	±1.25%
Mass flux	±5.85%
Heat flux	±2.89%
Mass quality	±5.99%
Heat transfer coefficient	±6.75%

Table 1 lists the experimental test setup specifications in this study. The physical properties of the refrigerant were obtained by referencing the REFPROP 6. The temperature and flow rate data were recorded by using the Darwin DAQ32 Plus logger R9.01 software program and version 2.41 of the Micro Motion ProLink Software package, respectively. Table 2 gives a summary of the estimated uncertainty associated with the test parameters. The uncertainties were obtained from both random and systematic errors.

2.2 Data reduction

The local heat transfer coefficients at position z along the length of the test section were defined as follows:

$$h = \frac{q}{T_{wi} - T_{sat}} \tag{1}$$

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 by using steady-state one-dimensional radial conduction heat transfer through the wall with internal heat generation. The quality, x, at the measurement locations, z, was determined based on the thermodynamic properties:

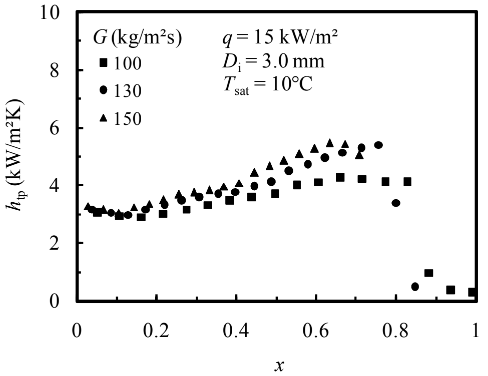

Fig. 2. The effect of mass flux on heat transfer coefficient.

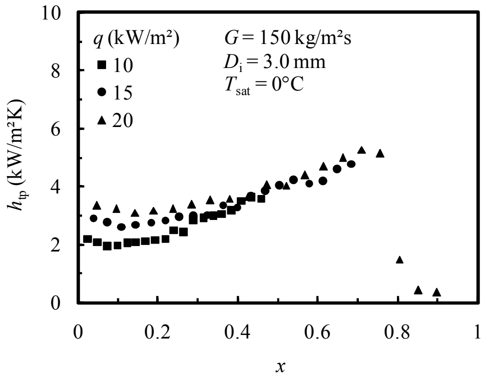

Fig. 3. The effect of heat flux on heat transfer coefficient.

$$x = \frac{i - i_{\mathrm{f}}}{i_{\mathrm{fg}}} \tag{2}$$

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 with the following equation to determine the initial point of saturation.

$$z_{\mathrm{sc}} = L\frac{i_{\mathrm{f}} - i_{\mathrm{f,in}}}{\Delta i} = L\frac{i_{\mathrm{f}} - i_{\mathrm{f,in}}}{\left(Q/W\right)} \tag{3}$$

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

$$x_{\mathrm{o}} = \frac{\Delta i + i_{\mathrm{f,in}} - i_{\mathrm{f}}}{i_{\mathrm{fg}}} \tag{4}$$

The saturation pressure at the initial point of saturation was then determined by interpolating the measured pressure and the subcooled length.

3. Results and discussion

Fig. 2 shows the effect of mass flux on the heat transfer coefficient. An insignificant effect of mass flux on the heat transfer coefficient at the low quality region of around up to 0.2 in Fig. 2 indicates that nucleate boiling heat transfer is predominant. The high nucleate boiling heat transfer is supposed because of the physical properties of the refrigerants and the geometric effect of the small channels. Several studies with small tubes, including Kew and Cornwell [6], Lazarek and Black [7], Wambsganss et al. [8], Bao et al. [1], Zhang et al. [2], Tran et al. [3] and Pamitran et al. [5], reported that nucleate boiling is predominant in small channels, which is opposite that of the predominantly convective-dominated heat transfer in conventional channels. Using a 1.95 mm tube, Bao et al. [1] showed that the heat transfer coefficients were independent of mass flux. At the moderate quality region, heat transfer coefficients increase with increasing mass flux and vapor quality. This is similar to the results reported by Kuo-Wang [9], who used R-22 in a 9.52 mm tube. The effect of mass flux on the heat transfer coefficient appears at moderate-high vapor quality of around 0.4–0.6. A higher mass flux results in a greater heat transfer coefficient at moderately high vapor quality due to the increasing convective boiling heat transfer contribution. At the high quality region of around 0.7–1.0, a decrease in the heat transfer coefficient occurs at the lower quality under the higher mass flux condition, as is shown in Fig. 2. It is supposed that the higher mass flux results in a lower dry-out quality. This trend agrees with Pettersen [10] and Yun et al. [11]. For the higher mass flux condition in the convective evaporation region, an increase in the heat transfer coefficient appears at a lower quality, which can be explained by the annular flow becoming dominant with increasing quality. Nucleate boiling suppression appears earlier for the higher mass flux, which means that convective heat transfer appears earlier under the higher mass flux condition. The lower mass flux condition results show smaller increases in the heat transfer coefficient in the convective region.

Fig. 3 shows that a strong dependence of the heat transfer coefficients on the heat flux appears at the low quality region of around up to 0.3. At the low quality region, the heat transfer coefficients increased with increasing heat flux. Nucleate boiling is known

Table 3. Physical properties of propane.

T (°C)	ρ_f/ρ_g	μ_f/μ_g	σ (10^{-3} N/m)	P (kPa)
10	37.32	13.96	8.85	636
5	43.58	15.03	9.48	551
0	51.06	16.20	10.13	474

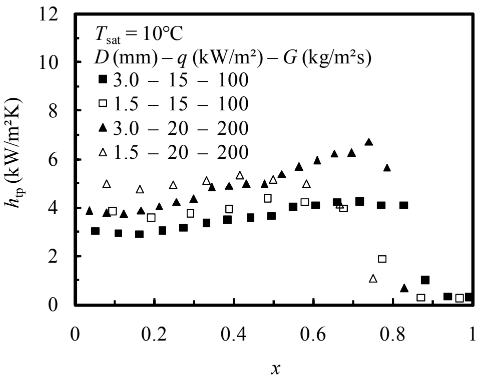

Fig. 4. The effect of inner tube diameter on heat transfer coefficient.

to be dominant in the initial stage of evaporation, particularly under high heat flux conditions. The effect of heat flux on the heat transfer coefficient shows the dominance of nucleate boiling heat transfer. Nucleate boiling is suppressed at high quality where the effect of heat flux on the heat transfer coefficient becomes lower. As the heat flux increases at high qualities, the evaporation is more active and the dry-out quality becomes lower. The trend illustrated in Fig. 3 agrees with previous studies, e.g., Kew and Cornwell [6], Yan *et al.* [12], and Kuo-Wang [9].

Fig. 4 illustrates the effect of inner tube diameter on the heat transfer coefficient. At the low quality region of around up to 0.4, smaller inner tube diameter shows higher heat transfer coefficient. This is due to more active nucleate boiling in the smaller diameter tube. As the tube diameter becomes smaller, the contact surface area of heat transfer increases. The more active nucleate boiling causes dry-patches to appear earlier. Therefore, the dry-out quality is relatively lower for the smaller tube. The quality for rapid increase 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 effect of saturation temperature is depicted in Fig. 5. The heat transfer coefficient increases with an increase in saturation temperature, which is due to the more active nucleate boiling. Table 3 shows that the higher temperature has a lower surface tension and higher pressure. The lower surface tension and higher pressure result in higher boiling nucleation. At the high quality region, dry-out appeared earlier for the lower saturation temperature. This result can be explained with the propane's density ratio, ρ_f/ρ_g, and viscosity ratio, μ_f/μ_g. As shown in Table 3, the lower temperature has higher density ratio, ρ_f/ρ_g, and viscosity ratio, μ_f/μ_g. A higher density ratio, ρ_f/ρ_g, and/or a lower viscosity ratio, μ_f/μ_g, cause higher entrainment, which is the reason that the dry-out comes easier. The experimental results show that the effect of density ratio, ρ_f/ρ_g, on the dry-out is higher than that of viscosity ratio, μ_f/μ_g. Therefore, the lower saturation temperature results in a lower dry-out quality, as shown in Fig. 5. It is concluded that the effect of density ratio, ρ_f/ρ_g, and viscosity ratio, μ_f/μ_g, on the heat transfer coefficient is necessary to be considered in the prediction of heat transfer coefficient.

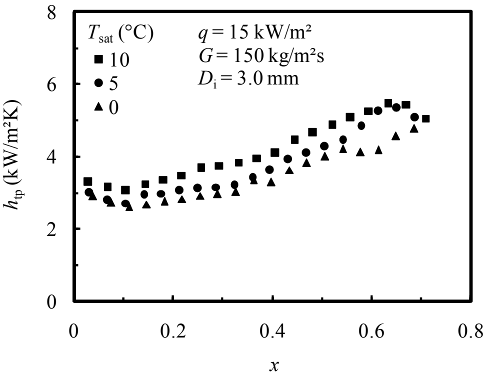

Fig. 5. The effect of saturation temperature on heat transfer coefficient.

The heat transfer coefficients in the present study are analyzed and compared by using seven previous heat transfer coefficient correlations. Table 4 shows the deviation percentage of the comparison and Fig. 6 shows the selected comparisons of the experimental heat transfer coefficient with the existing correlations. Overall, Shah's [13] correlation gave the best prediction of all seven. Shah's [13] correlation was developed by using conventional refrigerants in a conventional channel. The correlation of Tran et al. [3], which was developed for flow boiling heat transfer in small channels, also works well with the present experimental data. Jung et al.'s [14] correlation is a superposition model that was developed by using conventional channels. The prediction using Jung et al.'s [14] correlation shows a higher deviation for the smaller diameter data. Jung et al.'s [14] correlation

Table 4. Deviatnfhe heat transfer coefficient comparison between the present data and the previous correlations.

Deviation (%)	Shah [13]	Tran et al. [3]	Jung et al. [14]	Gungor-Winterton [15]	Takamatsu et al. [17]	Kandlikar-Steinke [4]	Chen [16]
MD	15.84	18.26	20.38	21.22	23.55	25.92	36.00
AD	-0.59	-9.82	19.70	16.79	22.52	16.70	17.73

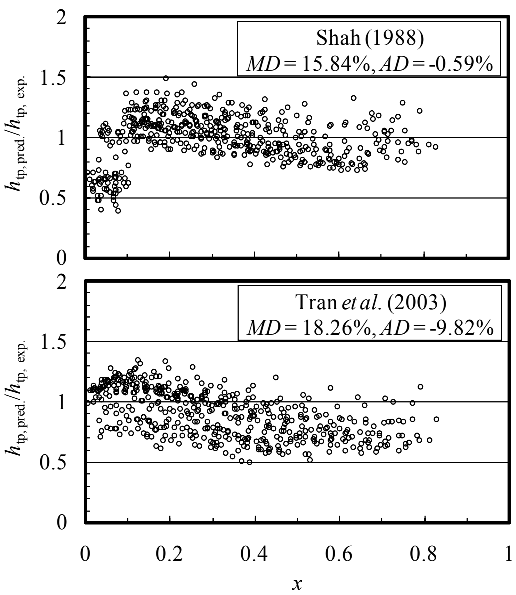

Fig. 6. The comparison of the experimental heat transfer coefficient with existing correlations.

does not consider the laminar flow which appears in flow with small channels. The high deviation, relatively, in the prediction using the Gungor-Winterton [15], Takamatsu et al. [16], and Chen [17] correlations was supposed because those correlations were developed by using conventional channels. The correlation reported by Kandlikar-Steinke [4] was developed for flow boiling heat transfer in small channels but it did not work quite well with propane.

The pool boiling correlations of Cooper [18] and Stephan-Abdelsalam [19] have been used to calculate the nucleate boiling heat transfer contribution in several previous studies such as Jung et al. [14] and Gungor-Winterton [15]. The Cooper [18] correlation is a function of reduced pressure, molecular weight and heat flux, whereas the Stephan-Abdelsalam [19] correlation is a function of the thermal conductivity, surface tension, density, heat flux, saturation temperature and the Prandtl number. Therefore, the present study used the both pool boiling heat transfer coefficients as a comparison in order to obtain the best result.

4. Development of a new correlation

4.1 Modification of factor F

Two important mechanisms, nucleate boiling and forced convective evaporation, mainly govern flow boiling heat transfer. Because of its high boiling nucleation, the appearance of convective heat transfer for evaporative refrigerants in small channels is delayed compared with that in conventional channels. Therefore, the prediction of convective heat transfer contribution for refrigerants in small channels will be different from that in conventional channels. The new heat transfer coefficient correlation in this study was developed only with the experimental data obtained prior to dry-out.

Chen [17] introduced a multiplier factor F=fn(X_{tt}) to account for the increase in convective turbulence due to the presence of a vapor phase. Chen [17] reported the factor, F, to be a function of X_{tt}, which needs to be evaluated again physically for flow boiling heat transfer in minichannels which has laminar flow condition due to the small diameter effect. The liquid-vapor flow condition of this experimental result shows 59% turbulent-turbulent, 37% laminar-turbulent and 4% turbulent-laminar. It was obtained by calculating the Reynolds number for each liquid and vapor phase. By considering the flow conditions (laminar or turbulent) in the Reynolds number factor F, Zhang *et al.* [2] introduced a relationship between the Reynolds number factor F and the two-phase frictional multiplier 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 [20],

$$\phi_f^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \quad (5)$$

For liquid-vapor flow conditions 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 Martinelli parameter is defined as follows:

$$X = \sqrt{\frac{(\mathrm{d}p/\mathrm{d}z)_\mathrm{f}}{(\mathrm{d}p/\mathrm{d}z)_\mathrm{g}}} = \left(\frac{f_\mathrm{f}}{f_\mathrm{g}}\right)^{1/2}\left(\frac{1-x}{x}\right)\left(\frac{\rho_\mathrm{g}}{\rho_\mathrm{f}}\right)^{1/2} \quad (6)$$

In this study, the Blasius equation of friction factor was used for friction factors f_f and f_g. Hence, the Martinelli parameter can be rewritten as follows:

$$X = \left(\frac{\mu_\mathrm{f}}{\mu_\mathrm{g}}\right)^{1/8}\left(\frac{1-x}{x}\right)^{7/8}\left(\frac{\rho_\mathrm{g}}{\rho_\mathrm{f}}\right)^{1/2} \quad (7)$$

The important effects of quality, density ratio $\rho_\mathrm{f}/\rho_\mathrm{g}$, and viscosity ratio $\mu_\mathrm{f}/\mu_\mathrm{g}$ on the heat transfer coefficient for boiling refrigerants are represented in Eq. (7). The liquid heat transfer is defined by the Dittus-Boelter correlation,

$$h_\mathrm{lo} = 0.023\frac{k_\mathrm{f}}{D}\left[\frac{G(1-x)D}{\mu_\mathrm{f}}\right]^{0.8}\left(\frac{C_\mathrm{pf}\mu_\mathrm{f}}{k_\mathrm{f}}\right)^{0.4} \quad (8)$$

The factor F is a convective two-phase multiplier that accounts for enhanced convective due to the co-current flow of liquid and vapor. A new factor F as shown in Fig. 7 was developed with a regression method from the experimental data. The new factor F can be expressed as follows:

$$F = 0.023\phi_\mathrm{f}^2 + 0.977 \quad (9)$$

where ϕ_f^2 is in Eq. (5).

Table 5. Summary of the new heat transfer coefficient.

$$h_\mathrm{tp} = Sh_\mathrm{nbc} + Fh_\mathrm{lo}$$
$$S = 0.6226\left(\phi_\mathrm{f}^2\right)^{0.1068} Bo^{0.0777}$$
$$h_\mathrm{nbc} = 55P_r^{0.12}\left(-0.4343\ln P_r\right)^{-0.55} M^{-0.5} q^{0.67}, \text{ where } q \text{ is in Wm}^{-2}$$
$$F = 0.023\phi_\mathrm{f}^2 + 0.977$$
$$h_\mathrm{lo} = 0.023\frac{k_\mathrm{f}}{D}\left[\frac{G(1-x)D}{\mu_\mathrm{f}}\right]^{0.8}\left(\frac{C_\mathrm{pf}\mu_\mathrm{f}}{k_\mathrm{f}}\right)^{0.4}$$
$$\phi_\mathrm{f}^2 = 1 + \frac{C}{X} + \frac{1}{X^2}, \quad X = \left(\frac{\mu_\mathrm{f}}{\mu_\mathrm{g}}\right)^{1/8}\left(\frac{1-x}{x}\right)^{7/8}\left(\frac{\rho_\mathrm{g}}{\rho_\mathrm{f}}\right)^{1/2}$$
$$C(\mathrm{tt}) = 20,\ C(\mathrm{vt}) = 12,\ C(\mathrm{tv}) = 10,\ C(\mathrm{vv}) = 5$$

4.2 Nucleate boiling contribution

Mass flux is believed to have a significant effect on the suppression of nucleate boiling. A higher mass flux is corresponding to a higher suppression of nucleate boiling. For evaporation in a minichannel, the suppression is lower than that in a conventional channel.

The nucleate boiling heat transfer for the experimental data was predicted by using the Cooper [18] correlation, which is a pool boiling correlation developed based on an extensive study. For a surface roughness of 1.0 μm, the correlation is given as follows:

$$h = 55P_r^{0.12}\left(-0.4343\ln P_r\right)^{-0.55} M^{-0.5} q^{0.67} \quad (10)$$

where the heat flux, q, is in W m^{-2} and P_r is the reduced pressure ($P_\mathrm{r}=P_\mathrm{sat}/P_\mathrm{crit}$). The correlation covers reduced pressure from 0.001 to 0.9 and molecular weights from 2 to 200. Kew and Cornwell [6], using R-141b in a tube with a length of 500 mm and inner diameters of 1.39 to 3.69 mm, showed that the Cooper [18] pool boiling correlation best predicted their experimental data. Better prediction using the Cooper [18] correlation is also shown in the Jung *et al*. [21] study.

Chen [17] defined the nucleate boiling suppression

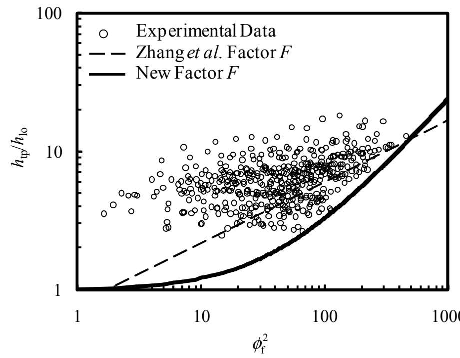

Fig. 7. Two-phase heat transfer multiplier as a function of ϕ_f^2.

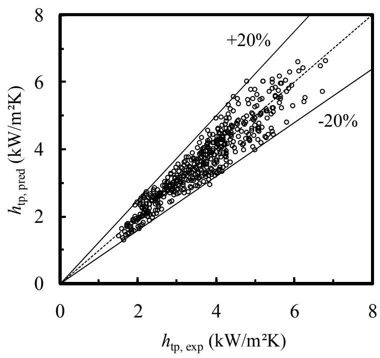

Fig. 8. Diagram of the experimental heat transfer coefficient, $h_{\mathrm{tp,\ exp.}}$, vs prediction heat transfer coefficient, $h_{\mathrm{tp,\ pred}}$.

factor, S, as a ratio of the mean superheat ΔT_e to the wall superheat ΔT_{sat}. Chen's factor S was developed with a conventional channel, hence further evaluation is needed to apply it for refrigerants in a minichannel. Jung et al. [14] proposed a convective boiling heat transfer multiplier factor N as a function of the quality, heat flux, and mass flow rate (represented by using X_{tt} and Bo) to represent the strong effect of nucleate boiling in flow boiling by comparing it with that in nucleate pool boiling, h_{nbc}/h_{pb}. To consider laminar flow in minichannels, the Martinelli parameter, $X_{tt,}$ is replaced by the two-phase frictional multiplier, ϕ_f^2 . Using the experimental data from this study, a new nucleate boiling suppression factor, a ratio of h_{nbc}/h_{pb}, is proposed as follows:

$$S = 0.6226\left(\phi_f^2\right)^{0.1068} Bo^{0.0777} \quad (11)$$

4.3 Heat transfer coefficient comparison

The new heat transfer coefficient correlation was developed by using a regression method with 479 experimental data points. The comparison of the experimental heat transfer coefficient, h_{exp},, and the prediction heat transfer coefficient, h_{pred}, is shown in Fig. 8. The new correlation showed good agreement with a mean deviation of 8.27% and an average deviation of -0.01%. Table 5 gives a summary of the new correlation. The use of Stephan-Abdelsalam's [19] correlation to predict the nucleate boiling heat transfer contribution in developing a new correlation showed a slightly higher deviation (mean deviation and average deviation of 9.05% and 0.06%, respectively).

5. Concluding remarks

Convective boiling heat transfer experiments were performed in horizontal minichannels with propane. Mass flux has an insignificant effect on the heat transfer coefficient at the low quality region. At the moderate quality region, the heat transfer coefficients increase with mass flux and vapor quality. At the high quality region, a decrease in the heat transfer coefficient occurs at a lower quality for a higher mass flux condition. A strong dependence of the heat transfer coefficients on the heat flux appears at the low quality region. Jung *et al.*'s [14] correlation gave the best prediction among the six reported correlations.

The physical properties of the refrigerant and geometric effect of the small tube must be considered when developing a new heat transfer coefficient correlation. Laminar flow appears during flow boiling in small channels. Therefore, in this study, a modified correlation of the multiplier factor on the convective boiling contribution, *F,* and the nucleate boiling suppression factor, *S,* was developed by using a laminar flow consideration. The new boiling heat transfer coefficient correlation based on the superposition model for propane in minichannels showed a mean deviation and an average deviation of 8.27% and -0.01%, respectively, which highlights the good agreement between the measured data and the calculated heat transfer coefficients.

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