HEFAT2010
7th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
19-21 July 2010
Antalya, Turkey

CHARACTERISTICS OF TWO-PHASE FLOW BOILING HEAT TRANSFER OF NH_3, C_3H_8 AND CO_2 IN HOZONTAL CIRCULAR SMALL TUBES

Choi K.I.[a], Pamitran A.S.[b], Oh J.T.[c]*
[a]Graduate School, Chonnam National University, Yeosu, Chonnam 550-749, South Korea
[b]Department of Mechanical Engineering, University of Indonesia, Kampus Baru UI, Depok 16424, Indonesia
[c]*Department of Refrigeration and Air Conditioning Engineering, Chonnam National University, San 96-1, Dunduk-Dong, Yeosu, Chonnam 550-749, South Korea, E-mail: ohjt@chonnam.ac.kr

ABSTRACT

An experimental investigation on the characteristics of two-phase boiling heat transfer of NH_3, C_3H_8 and CO_2 in horizontal small stainless steel tubes of 1.5 and 3.0 mm inner diameters are presented in this paper. Experimental data were obtained over a heat flux range of 5 to 70 kW/m^2, mass flux range of 50 to 600 kg/m^{2}s, saturation temperature range of 0 to 12°C, and quality up to 1.0. The test section was heated uniformly by applying an electric current to the tubes directly. Nucleate boiling heat transfer was the main contribution, particularly at the low quality region. The heat transfer coefficient of the present working refrigerants was compared. Laminar flow was observed in the small tubes. A new boiling heat transfer coefficient correlation based on the superposition model for refrigerants in small tubes was developed.

INTRODUCTION

Micro- and minichannels are increasingly being used to achieve high heat transfer rates with compact heat exchangers. Evaporation inside small diameter channels finds applications in heat pipes and compact heat exchangers for electronic equipment or spacecraft thermal control, in residential air conditioning and in refrigeration applications. For the design of these compact evaporators, generally accepted correlations are required to calculate the heat transfer coefficients and the pressure drop. There have been little data published relating to two-phase flow and heat transfer in small channels with natural refrigerants compared with the data for large channels.

This study was undertaken to obtain experimental data for NH_3, C_3H_8, CO_2 and to determine their local heat transfer coefficient during evaporation in minichannels. The results were compared with several existing heat transfer coefficient correlations, and a new correlation of heat transfer coefficient for NH_3, C_3H_8, CO_2 in minichannels is developed in this study.

NOMENCLATURE

a	[-]	Accelerational contribution
Bo	[-]	Boiling number
C	[-]	Chisholm parameter
D	[m]	Diameter
F	[-]	Convective two-phase multiplier, Eq. (8)
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
W	[kg/s]	Mass flow
X	[-]	Martinelli parameter
x	[-]	Mass quality
z	[m]	Axial coordinate

Special characters

μ	[Ns/m^2]	Dynamic viscosity
ρ	[kg/m^3]	Density
σ	[N/m]	Surface tension
ϕ^2	[-]	Two-phase frictional multiplier

Subscripts

exp	Experimental value
f	Saturated liquid
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
tp	Two-phase
w	Wall

EXPERIMENTAL APPARATUS AND METHOD

The experimental facility is schematically shown in Figure 1. The flow rate of the refrigerant was controlled by a variable 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. The vapor refrigerant from the test section was condensed in the condenser and subcooler, and then it was supplied to the receiver.

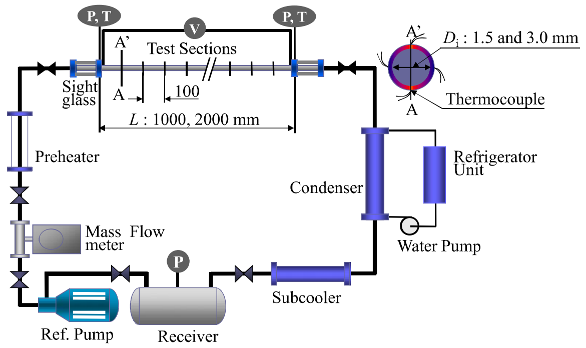

Figure 1 Experimental test facility

The test sections composed of stainless steel smooth tubes with inner diameters of 1.5 mm and 3.0 mm and a heated length of 2000 mm. The tubes were well insulated with rubber and foam. The test sections were uniformly and constantly heated by applying an electric current directly to their tube walls. 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 T-type cooper-constantan thermocouples at each site. 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 physical properties of the refrigerant were obtained by referencing the REFPROP 8.1.

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

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

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 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

$$T_{\mathrm{wi}} = T_{\mathrm{wo}} + \frac{Q}{16k}\left(D_o^2 - D_i^2\right) - \frac{q}{8k} D_o^2 \ln\frac{D_o}{D_i} \tag{2}$$

$$x = \frac{i - i_{\mathrm{f}}}{i_{\mathrm{fg}}} \tag{3}$$

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.

$$z_{\mathrm{sc}} = L\frac{i_{\mathrm{f}} - i_{\mathrm{f,in}}}{\Delta i} = L\frac{i_{\mathrm{f}} - i_{\mathrm{f,in}}}{\left(Q/W\right)} \tag{4}$$

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

$$x_{\mathrm{o}} = \frac{\Delta i + i_{\mathrm{f,in}} - i_{\mathrm{f}}}{i_{\mathrm{fg}}} \tag{5}$$

RESULTS AND DISCUSSION

Figure 2(a) to (c) shows the effect of mass flux on heat transfer coefficient for NH_3, C_3H_8, and CO_2, respectively. Figure 2 shows an insignificant effect of mass flux and vapour quality 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 [1], Lazarek and Black [2], Wambsganss *et al.* [3], Tran *et al.* [4] , Bao *et al.* [5] and Thome and Collier [6] showed that, in small channels, nucleate boiling is predominant. The high nucleate boiling heat transfer occurs because of the physical properties of the refrigerants, namely surface tension and pressure, and the geometric effect of small channels. The effect of mass flux on the heat transfer coefficient appears at moderate-high vapor quality in NH_3 and C_3H_8, wherein the effect is higher with increasing vapor quality. A higher mass flux results in greater heat transfer coefficient at moderate-high vapor quality due to the increasing convective boiling heat transfer contribution.

Figure 3(a) to (c) shows that a strong dependence of the heat transfer coefficients on heat flux appears at the low quality region for NH_3, C_3H_8, and CO_2, respectively. At the low quality region, the heat transfer coefficients increased with increasing heat flux. Nucleate boiling is known 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

Table 1 Experimental conditions

Working refrigerant	NH_3		C_3H_8		CO_2	
Test section	Horizontal smooth stainless steel minichannels					
Quality	Up to 1.0					
Inner tube diameter (mm)	1.5	3.0	1.5	3.0	1.5	3.0
Tube length (mm)	1000	2000	2000	2000	2000	2000
Mass flux (kg/m^{2}s)	50 – 500	50 – 500	10 – 400	50 – 400	300 – 600	200 – 600
Heat flux (kW/m^2)	10 – 40	10 – 80	5 – 20	5 – 25	10 – 30	20 – 30
Inlet T_{sat} (ºC)	0 – 10	0 – 10	0 – 12	0 – 11	0 – 11	0 – 10

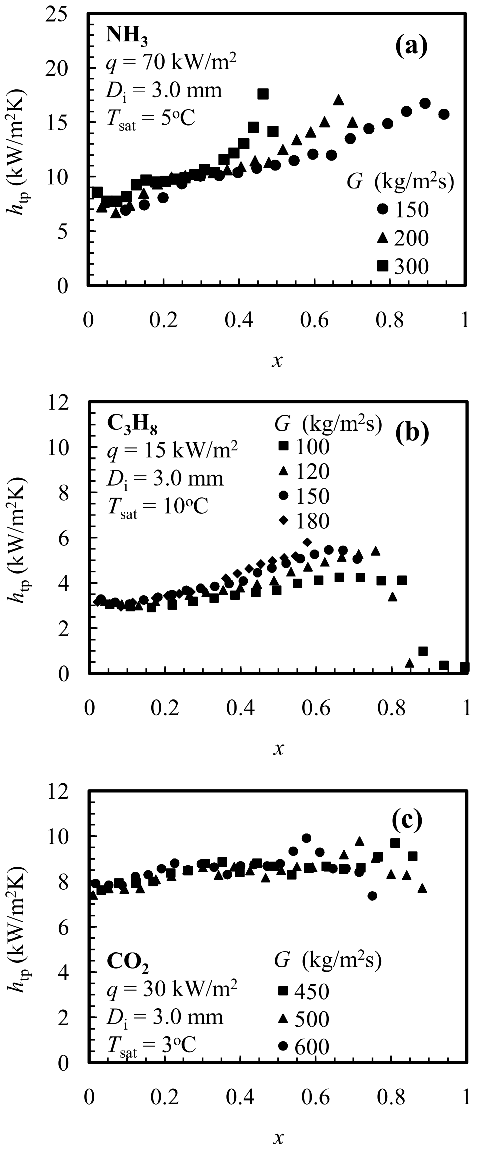

Figure 2 The effect of mass flux on heat transfer coefficients: (a) NH_3, (b)C_3H_8, and (c)CO_2

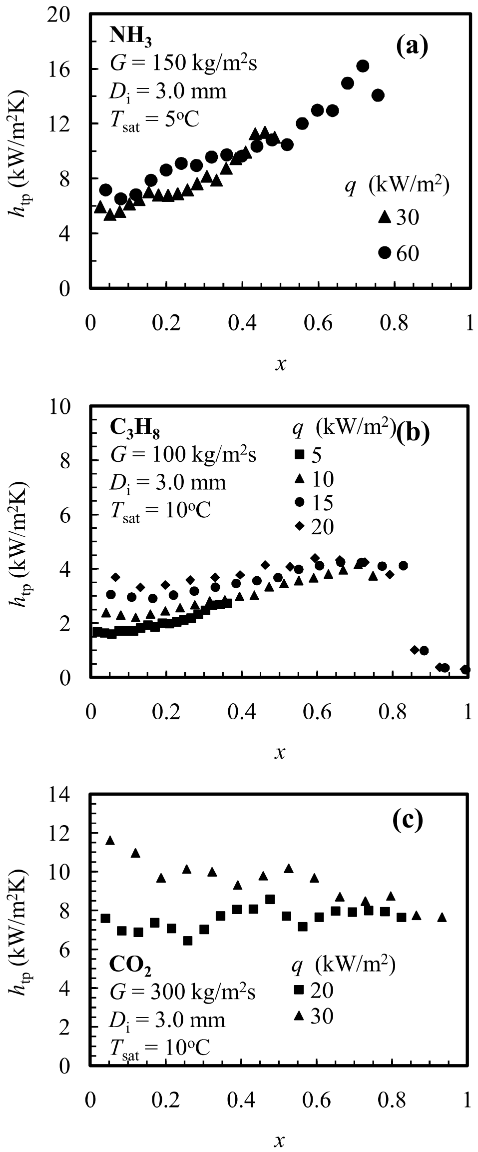

Figure 3 The effect of heat flux on heat transfer coefficients: (a) NH_3, (b)C_3H_8, and (c)CO_2

transfer. Nucleate boiling is suppressed at high quality where the effect of heat flux on heat transfer coefficient becomes lower. As the heat flux increases, the evaporation is more active and the dry-out quality becomes lower.

Figure 4(a) to (c) shows the effect of saturation temperature on heat transfer coefficient for NH_3, C_3H_8, and CO_2, respectively. The heat transfer coefficient increases with an increase in saturation temperature, which is due to a more active nucleate boiling.

Figure 5 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.

Figures 6 show the comparisons of the heat transfer coefficients of C_3H_8 and CO_2 at some experimental conditions. The heat transfer coefficient of CO_2 was higher than that of the other working refrigerants during evaporation under all test conditions. The higher heat transfer coefficient of CO_2 is believed to be due to its high boiling nucleation. CO_2 has much lower surface tension and applies much higher pressure than the

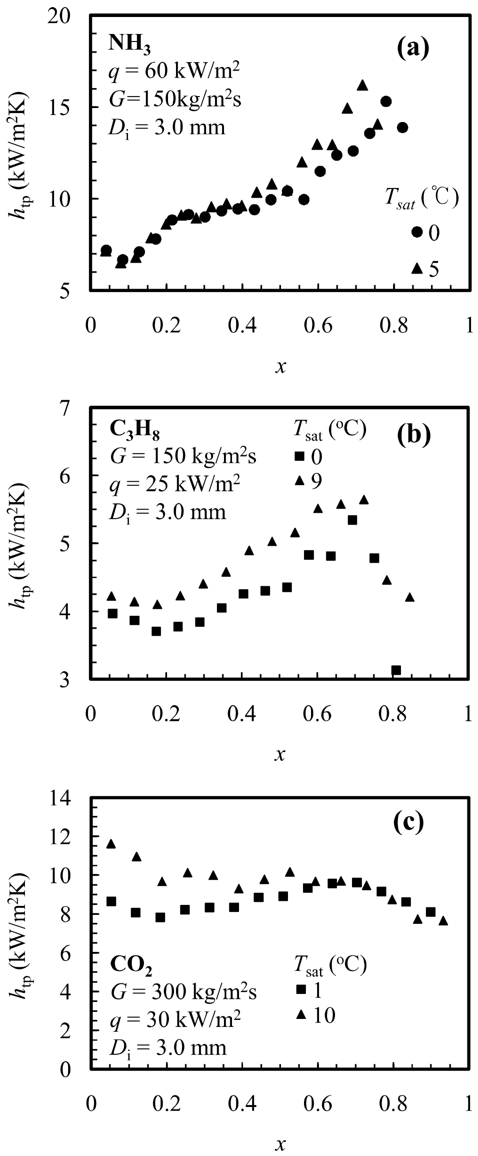

Figure 4 The effect of saturation temperature on heat transfer coefficients: (a) NH_3, (b)C_3H_8, and (c)CO_2

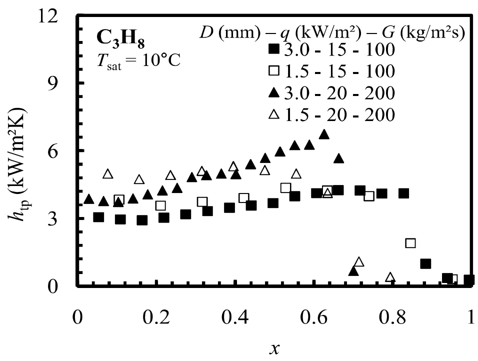

Figure 5 The effect of inner tube diameter on heat transfer coefficient for C_3H_8

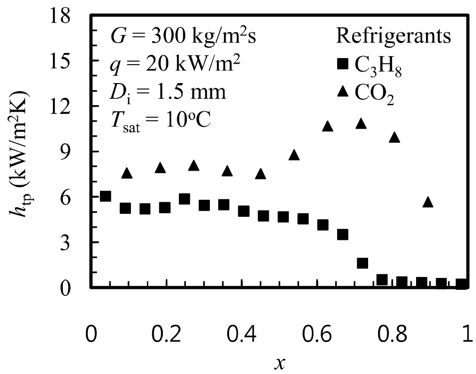

Figure 6 Heat transfer coefficient comparison of the present working refrigerants.

other working refrigerants. The comparisons of the physical properties of the present working refrigerants are given in Table 2. CO_2 has a much lower viscosity ratio μ_f/μ_g than the other working refrigerants, which means that the liquid film of CO_2 can break easier than those of the other refrigerants. CO_2 has also a much lower density ratio ρ_f/ρ_g than the other working refrigerants, which leads to a lower vapor velocity, which in turn causes less suppression of 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 3. For the overall data, Shah's [6] correlation gave the best prediction among the others. Shah's [7] correlation was developed using conventional refrigerant in a conventional channel.

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.* [13] 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

Table 2 Physical properties of NH_3, C_3H_8, and CO_2 at 10°C

Refrigerant	P(MPa)	ρ_f(kg/m^3)	ρ_g(kg/m^3)	ρ_f/ρ_g	μ_f(10^{-6} Pa s)	μ_g(10^{-6} Pa s)	μ_f/μ_g	σ(10^{-3} N/m)
NH_3	0.615	623.64	4.868	128.32	153.03	9.36	16.35	29.589
C_3H_8	0.636	515	13.8	37.32	113.8	8.151	13.96	8.85
CO_2	4.497	861.7	134.4	6.41	86.37	15.46	5.59	2.77

Table 3 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	25.6	48.31	28.45	27.31	26.37	53.92	40.56
Average Deviation	1.48	48.18	21.13	19.12	11.09	-9.91	27.88

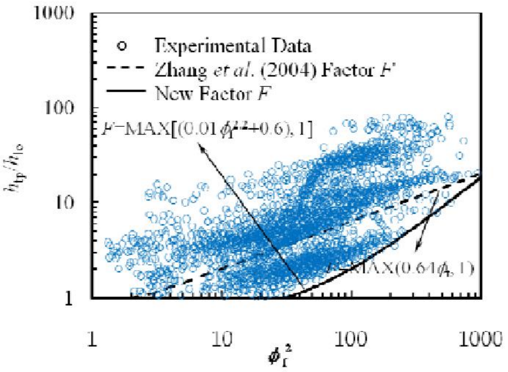

Figure 7 Two-phase heat transfer multiplier as a function of ϕ_f^2

ϕ_f^2 is a general form for four conditions according to Chisholm [14]

$$\phi_f^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{6}$$

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

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

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.

$$F = 0.01\phi_f^{2.2} + 0.6 \tag{8}$$

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 [15]. For a surface roughness that is set equal to 1.0 μm, his correlation is given as:

$$h = 55P_r^{0.12}\left(-0.4343\ln P_r\right)^{-0.55} M^{-0.5} q^{0.67} \tag{9}$$

Chen [11] 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

$$S = 0.2205\left(\phi_f^2\right)^{0.1932} Bo^{-0.0333} \tag{10}$$

Heat transfer Coefficient Comparison

The new heat transfer coefficient correlation is developed using a regression method with 1640 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 13.17% and an average deviation of -0.76%.

CONCLUDING REMARKS

Convective boiling heat transfer experiment was performed in horizontal minichannels with NH_3, C_3H_8, and CO_2. Mass flux, heat flux, inner tube diameter, and saturation temperature have an effect on heat transfer coefficient. The geometric effect of the small channels contributed to the higher boiling nucleation.

The physical properties of the refrigerant and 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

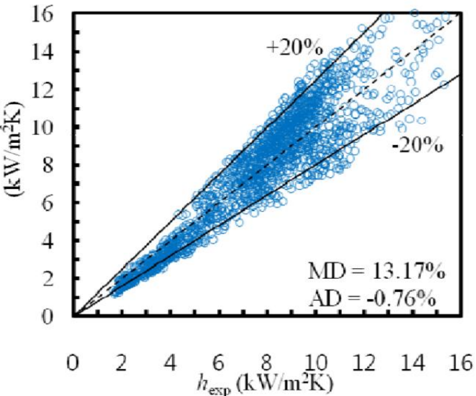

Figure 8 Comparison of the experimental and predicted heat transfer coefficients using the newly correlation

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 13.17% mean deviation and -0.76% average deviation.

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