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ABSTRACT

An experimental study for heat transfer coefficient and pressure drop with two-phase flow phase change was conducted with R-1234yf in horizontal circular small tubes. The experimental facilities have been used to take data under various flow conditions with intensive study. The test section is made of stainless steel tubes with inner tube diameters of 1.5 mm and 3.0 mm, the length of 1000 mm and 2000 mm each. The small tubes were uniformly heated by engaging an electric current directly to the single tubes; all components were well insulated to prevent heat losses. Local heat transfer coefficients and pressure drop were obtained for a heat flux of 5-40 kW m-2, a mass flux of 50-600 kg m-2 s-1, saturation temperatures of 0, 5, and 10℃ and quality of up to 1.0. The effects of mass flux, heat flux, saturation temperature, and inner tube diameter on heat transfer coefficient and pressure drop are reported. Nucleate boiling heat transfer contribution was predominant, especially at low quality region, and laminar flow appeared in the evaporative small tubes. A new boiling heat transfer coefficient correlation for R-1234yf was developed.

1. INTRODUCTION

Recently, R-1234yf has been paid attention by many researchers and manufacturers as a next-generation refrigerant to replace R134a in mobile air conditioners. Although both R-1234yf and R-134a have no Ozone Depletion Potential, but the Global Warming Potential of R-134a within 100 years is 1340 while the R-1234yf's is 4. In addition, R-1234yf has an atmospheric lifetime of only 11 days, compared to the one of R-134a-13 years. Hence, parameters of R-1234yf can reach the standards of EU Regulation.
Moreover, to present, almost research of R-1234yf have focused on its thermal dynamic properties, the comparison between R-1234yf and R-134a and its applications. There is very limited two-phase heat transfer data measured during vaporization available in the literature. In this paper, an experimental study for two-phase flow heat transfer was conducted with R-1234yf during vaporization in horizontal small tubes. A new boiling heat transfer coefficient correlation for R-1234yf was also developed.

2. EXPERIMENTAL APPARATUS AND METHODS

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.

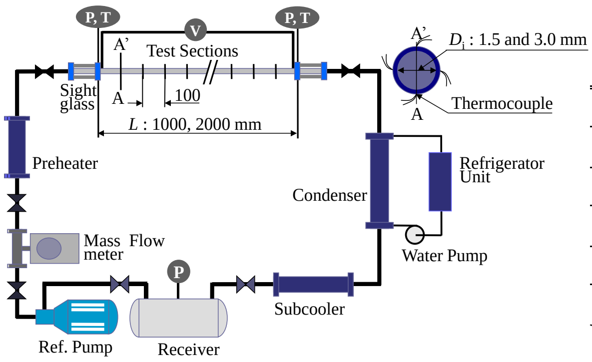

Figure 1: Experimental test facility

Table 1: Experimental conditions

Refrigerants	R-1234yf
Test Section	Horizontal stainless steel small tubes
Inner Diameter (mm)	1.5, 3.0
Tube Length (mm)	1000, 2000
Mass Flux (kg/m^{2}s)	50 –600
Heat Flux (kW/m^2)	5 - 40
Vapor Quality	0.0 - 1.0
Inlet T_{sat} (^{o}C)	0, 5, 10

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 sub cooler, and then it was supplied to the receiver.

The test sections composed of stainless steel smooth tubes with inner diameters of 1.5 mm and 3.0mm and a heated length of 1000 and 2000 mm respectively. 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 thermocouple. 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 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 sub cooled length for reduction data accuracy. The sub cooled length was calculated using the following equation to determine the initial point of saturation.

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

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

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

3. EXPERIMENTAL RESULTS AND DISCUSSION

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. Fig. 2 also illustrates that the pressure drop increases as the heat flux increases. It is presumed that the increasing heat flux results in a higher vaporization, which increases the average fluid vapor quality and flow velocity. The pressure drop in the 1.5 mm tube is higher than that in the 3.0 mm tube. The 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 higher pressure drops.

Figure 3 shows the effect of mass flux 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. 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. 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.

Figure 4 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.

The effect of saturation temperature on heat transfer coefficient is depicted in Figure5. The heat transfer coefficient increases with an increase in saturation temperature, which is due to a more active nucleate boiling.

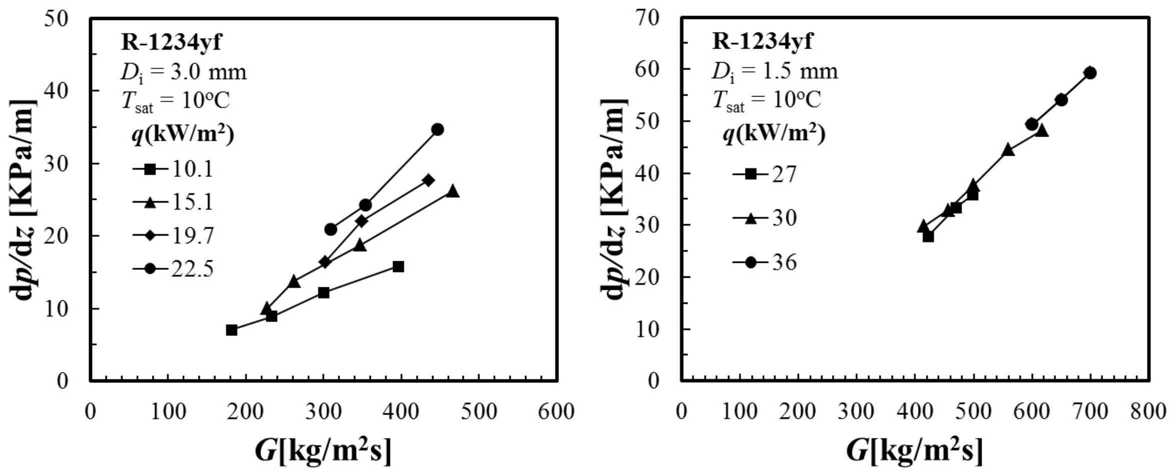

Figure 2 : The effect of mass flux and heat flux on pressure drop

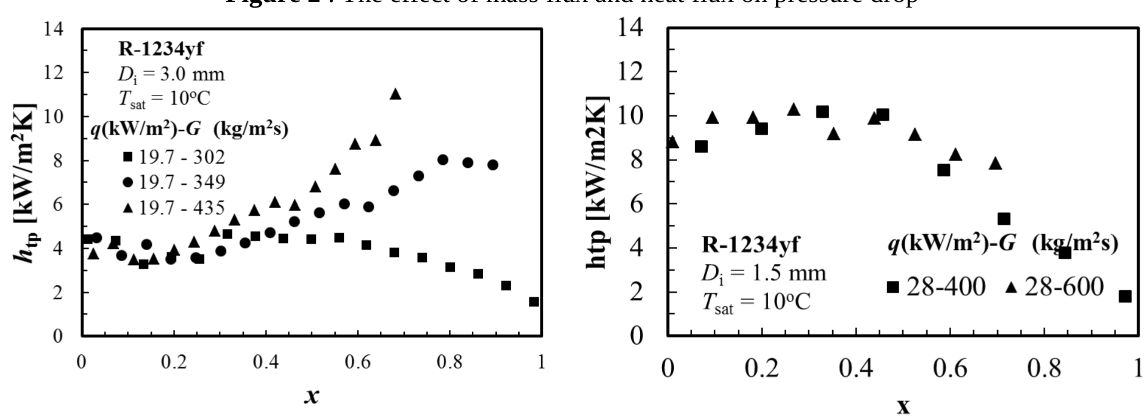

Figure 3 : The effect of mass flux on heat transfer coefficient.

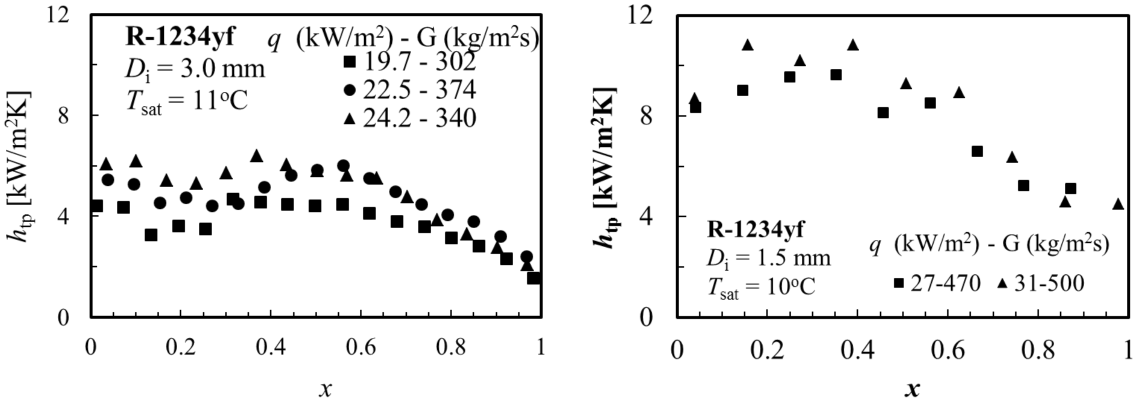

Figure 4 : The effect of heat flux on heat transfer coefficient.

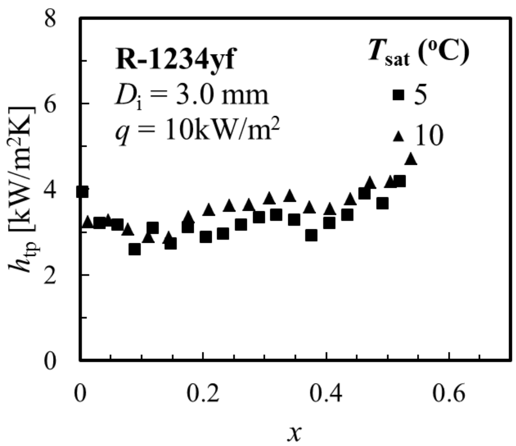

Figure 5 : The effect of heat flux on heat transfer coefficient.

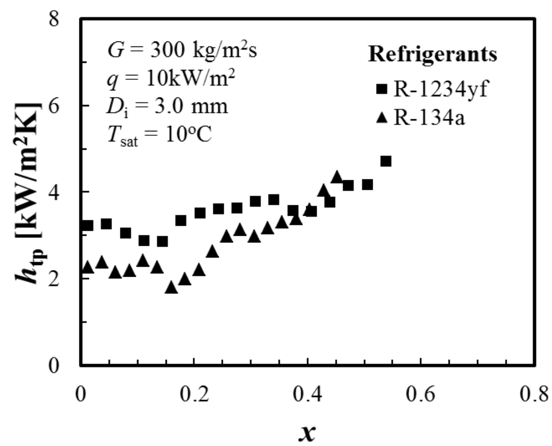

Figure 6 : Comparison of the heat transfer coefficient of R-134a and R-1234yf.

Figure 6 shows a comparison of the heat transfer coefficient of R-134a and R-1234yf at saturation temperature of 10℃. The average experimental heat transfer coefficient of R-1234yf is about 20% higher than that of R-134a. The higher heat transfer coefficient of R-1234yf occurs because of its high boiling nucleation. R-1234yf has a lower surface tension and density ratio than R-134a.

The pre-dry out heat transfer coefficients of the present study were compared with six correlations for boiling heat transfer coefficient as shown in Table 2. For the overall data, Gungor-Winterton(1987) correlation gave the best prediction among the others.

4. DEVELOPMENT OF A NEW CORRELATION

4.1 Modification of factor *F*

Heat transfer of flow boiling is mainly governed by two important mechanisms, namely nucleate boiling and forced convective heat transfer. In two-phase flow boiling heat transfer, the nucleate boiling heat transfer contribution is suppressed by the two-phase flow. Therefore, the nucleate boiling heat transfer contribution may be correlated with the nucleate boiling suppression factor, S. Another contribution of convective heat transfer may be correlated with a liquid single phase heat transfer. The F factor is introduced as a convective two-phase multiplier to account for enhanced convection due to co-current two-phase flow. A superposition model of the heat transfer coefficient may be written as follows:

$$h_{tp} = Fh_{lo} + Sh_{pb} \tag{6}$$

Table 2 : Deviation of the heat transfer coefficient comparison between the experimental and the prediction with existing correlations.

Deviation (%)	**Gungor-Winterton(1987)**	**Takamatsu *et al.*(1993)**	**Wattelet *et al.*(1994)**	**Zhang *et al.*(2004)**	**Jung *et al.*(1989)**	**Kandlikar (1990)**
Mean	15.37	36.57	17.71	29.05	19.76	31.39
Average	3.68	24.77	4.11	17.11	11.37	28.98

Chen (1966) 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 reported the factor F to be a function of X_{tt} which must be evaluated again physically for flow boiling heat transfer in mini-channels with a laminar flow condition, due to the effect of the small diameter.
The factor F is a convective two-phase multiplier that accounts for enhanced convection due to the co-current flow of liquid and vapor. A new factor F, was developed with a regression method using our experimental data and can be expressed as follows:

$$F = \mathrm{MAX}\left[\left(0.065\left(\phi_{\mathrm{f}}^{2}\right)^{0.8} + 0.955\right), 1\right] \tag{7}$$

The liquid heat transfer is defined by the Dittus–Boelter correlation:

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

4.2 Nucleate Boiling Contribution

For evaporation in a small tube, the suppression is lower than that in a conventional tube. The nucleate boiling heat transfer for the experimental data was predicted using the Cooper (1984) correlation. For a surface roughness of 1.0 µm, the correlation is given as follows:

$$h_{\mathrm{pb}} = 55P_{\mathrm{r}}^{0.12}\left(-0.4343\ln P_{\mathrm{r}}\right)^{-0.55}M^{-0.5}q^{0.67} \tag{9}$$

To consider laminar flow in small tubes, the Martinelli parameter X_{tt} is replaced by a 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:

$$S = 38.2825\left(\phi_{\mathrm{f}}^{2}\right)^{-0.182}Bo^{0.353} \tag{10}$$

4.3 Heat transfer coefficient comparison

A modified heat transfer coefficient correlation was developed using a regression method with 239 experimental data points the R-1234yf in the test section with inner tube diameter of 1.5 mm. The heat transfer coefficient comparison from experiment, h_{exp}, and prediction, h_{pred}, using the modified heat transfer coefficient correlation is shown in Figure 7.

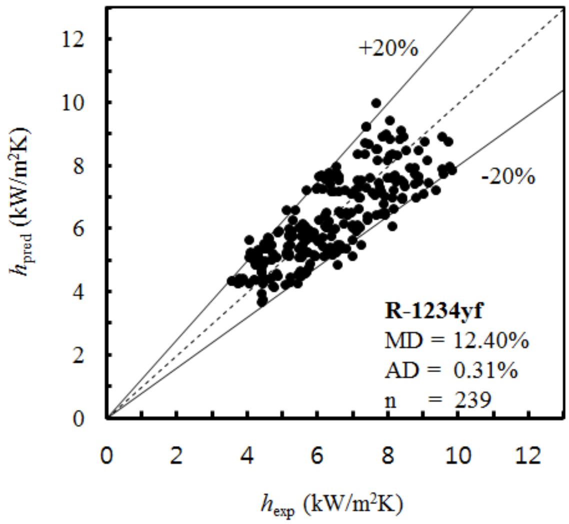

Figure 7: Heat transfer coefficient comparison between the present experimental data (h_{exp}) and the prediction with the newly modified correlation (h_{pred})

5. CONCLUSION

Convective boiling heat transfer experiments were performed in horizontal small tubes with R-1234yf. The results are summarized as follows:

- Mass flux and vapor quality had an insignificant effect on the heat transfer coefficient in the low quality region.
- A strong dependence of the heat transfer coefficients on heat flux appeared in the low quality region.
- The heat transfer coefficient of R-1234yf is about 20% higher than that of R-134a.
- The modified correlations of the multiplier factor on the convective boiling contribution *F*, and on the nucleate boiling suppression factor *S*, were developed using a laminar flow consideration. The new boiling heat transfer correlation based on a superposition model for R-1234yf in small tube with inner diameter of 1.5mm showed 12.40% mean deviation and 0.31% average deviation.

NOMENCLATURE

A	area	(m^2)	AD	Average Deviation,
c_p	Specific heat	(kJ kg^{-1} K^{-1})	B_O	Boiling number
D	Diameter	(m)	f	Friction factor
G	Mass flux	(kg m^{-2} s^{-1})		
g	Acceleration due to gravity	(m s^{-2})		
L	Tube length	(m)	MD	Mean Deviation,
h	heat transfer coefficient	(kW m^{-2}K^{-1})		$\mathrm{MD} = \left(\frac{1}{n}\right)\sum_{1}^{n}\left|\left(\left(\mathrm{d}p_{\mathrm{pred}} - \mathrm{d}p_{\mathrm{exp}}\right)\times 100 / \mathrm{d}p_{\mathrm{exp}}\right)\right|$
I	enthalpy	(kg kJ^{-1})	C	Chisholm parameter
P	Pressure	(kPa)		
q	Heat flux	(kW m^{-2})	Re	Reynolds number, $\mathrm{Re} = GD/\mu$
T	Temperature (K)		W	Mass flow rate (kg s^{-1})
We	Weber number,	$\mathrm{We} = G^2 D/\rho\sigma$	X	Lockhart-Martinelli parameter
x	Vapor quality	(-)		
z	Length	(m)		

Greek letters

α	Void fraction	μ	Viscosity (Pa·s)
ρ	Density (kg m^{-3})	σ	Surface tension (N m^{-1})
ϕ^2	Two-phase frictional multiplier	(dp/dz)	Pressure gradient (N m^{-2}m^{-1})
(dp/dz F)	Pressure gradient due to friction (N m^{-2}m^{-1})		

Subscripts

crit	Critical point	exp	Experimental value
f	Saturated liquid	g	Saturated vapor
i	Inner tube	lo	Liquid only
o	Outer tube	pb	Pool boiling
pred	Predicted value	r	Reduced
sat	Saturation	sc	Subcooled
t	Turbulent	tp	Two-phase
v	Laminar	w	Wall
n	Number of data		

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