

# A correlation for forced convective boiling heat transfer of nonazeotropic refrigerant mixture of HCFC22/CFC114 in a horizontal smooth tube

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**Abstract**—Experiments are carried out on the boiling heat transfer of nonazeotropic refrigerant mixtures of HCFC22/CFC114 flowing inside a 7.9 mm ID horizontal smooth tube. Using a water-heated, double-tube type evaporator, the local heat transfer coefficients are measured for both counter and parallel flows. An empirical correlation equation is proposed for the data in the annular-flow regime, taking into account the mixture effect in the authors' previous correlation equation for pure refrigerants. The mean deviation between the calculated and measured heat transfer coefficients is 8.9% for the present experimental data and 7.1% for the data obtained by Jung. The equation reveals that the convection heat transfer is dominant due to the suppression of nucleate boiling for the mixtures.

## 1. INTRODUCTION

NONAZEOTROPIC refrigerant mixtures (NARMs) first attracted the attention of researchers on refrigeration, air conditioning and heat pump systems because the performance is expected to be improved by realizing the Lorenz cycle. The other merit of using NARMs is to select the best combination of refrigerants for given conditions, i.e. temperatures and pressures. The capacity control of the systems have also been tried by varying the composition of mixtures. In addition, under the situation that HCFC may be replaced by alternative stratospherically safe refrigerants in the future, mixture of limited kinds of available pure refrigerants is a candidate for a substitute. However, the heat transfer of mixtures is inferior to that of pure refrigerant, so that the enhancement of heat transfer in the evaporator and condenser is very important (see ref. [1] for instance).

During the past decade, experiments on the horizontal flow boiling of refrigerant mixtures have been carried out and some correlation equations for the heat transfer coefficient have been proposed. The correlation of Hihara *et al.* [2] for HCFC22/CFC114, that of Mishra *et al.* [3] for HCFC22/CFC12 and that of Singal *et al.* [4] for CFC13/CFC12 are purely empirical equations whose coefficients depend on the specific composition. The correlation equation of Hihara and Saito [5] for HCFC22/CFC114, that of Jung *et al.* for HCFC22/CFC114 [6] and CFC12/HFC152a [7], and that of Murata and Hashizume for CFC114/CFC11 [8] and HFC134a/HCFC123 [9] are based on the idea of Chen [10] and Bennett and Chen [11], in which the flow boiling heat transfer coefficient has been expressed as the sum of nucleate boiling and forced convection. Most of these cor-

relation equations except that of Jung *et al.* [6, 7] are not applicable to the other mixtures because they also include some empirical relations or coefficients which can be used only for the specific combination and composition of refrigerants. This is mainly due to the fact that few correlations for both pure components are available in spite of a lot of earlier works on flow boiling of pure refrigerants.

In the previous paper [12], the authors have proposed a correlation equation for horizontal flow boiling of pure refrigerants. It is applicable to HCFC22, CFC114, HFC134a, CFC12, HFC152a and CFC113. The objective of the present work is to propose a correlation equation for binary refrigerant mixtures, taking into account of the mixture effect in the above equation. The experimental data were obtained for forced convective boiling of refrigerant mixture in a horizontal smooth tube. Employing a water-heated, double-tube type evaporator, the local heat transfer coefficients were measured for both counter and parallel flows. We used HCFC22/CFC114 as the test fluid because accurate information about thermo-physical properties was available and the temperature difference between boiling and dew points is relatively large: about 20°C at the pressure around 0.5 MPa.

## 2. EXPERIMENTS

### 2.1. Experimental apparatus

The experimental apparatus and the procedure employed in this study were the same as those in the previous study (see ref. [12] for details). The refrigerant loop consists of a positive-displacement pump, a preheater, a test section, a rear heater and two condensers. The test section is made of a double-tube

## NOMENCLATURE

$c_p$	isobaric specific heat capacity [J kg <sup>-1</sup> K <sup>-1</sup> ]	$y$	mole fraction of HCFC22
$d_i$	inside diameter of test tube [m]	$\hat{y}$	mass fraction of HCFC22
$G$	mass velocity [kg m <sup>-2</sup> s <sup>-1</sup> ]	$z$	normal distance from wall surface [m].
$g$	gravitational acceleration [m s <sup>-2</sup> ]	Greek symbols	
$h_{lg}$	latent heat of vaporization [J kg <sup>-1</sup> ]	$\alpha$	heat transfer coefficient [W m <sup>-2</sup> K <sup>-1</sup> ]
$K$	heat-flux-fraction factor, $q_{nb}/q$	$\lambda$	thermal conductivity [W m <sup>-1</sup> K <sup>-1</sup> ]
$La$	Laplace constant, equation (16)	$\mu$	viscosity [Pa s]
$P$	pressure [MPa]	$\rho$	density [kg m <sup>-3</sup> ]
$Pr$	Prandtl number	$\sigma$	surface tension [N m <sup>-1</sup> ]
$q$	heat flux [W m <sup>-2</sup> ]	$X_{tt}$	Lockhart–Martinelli parameter, $\{(1-x)/x\}^{0.9}(\rho_v/\rho_l)^{0.5}(\mu_l/\mu_v)^{0.1}$ .
$Re_{lo}$	Reynolds number for liquid only flow, $G(1-x)d_i/\mu_l$	Subscripts	
$S$	suppression factor	b	bulk
$S_{cvm}$	effective-temperature-difference factor which expresses a decrease in convective heat transfer of mixture from that of pure refrigerant, equation (20)	cv	convection
$S_{pbm}$	the factor which expresses a decrease in nucleate pool boiling heat transfer of mixture from that of pure refrigerant, equation (36)	int	vapor–liquid interface
$T$	temperature [°C]	l	liquid
$T_{rc}$	refrigerant temperature calculated with the equation of state [°C]	lo	liquid only
$T_{rm}$	measured refrigerant temperature [°C]	m	mixture
$x$	vapor quality	nb	nucleate boiling
		pb	nucleate pool boiling
		r	refrigerant
		s	heating water (heat source)
		sat	saturated
		v	vapor.

type evaporator, in which the refrigerant flows inside the inner tube and a heating water flows in the outer annulus. The inner tube is made of a straight smooth copper tube 6 m long and 7.9 mm in inside diameter. The outer annulus is subdivided into twelve subsections to measure the local heat transfer rate. The effective heated length of each subsection is 0.46 m.

The bulk temperatures of the refrigerant were measured at the inlet and outlet of the test section. The wall temperatures were measured at the center of each subsection with four thermocouples 0.2 mm in diameter, which were attached at the top, both sides and bottom of the outside surface of the inner tube. The refrigerant temperatures were measured with thirteen thermocouples which were inserted into the inner tube at the ends of subsections. The pressure at the inlet of the test section and the pressure drops during three serial subsections were measured with an absolute pressure transducer and four differential pressure transducers, respectively. The heat transfer rate at each subsection was calculated from the temperature change and the flow rate of heating water. For each test run, the vapor sample was drawn at the exit of the rear heater and the overall composition of mixture was measured with a gas chromatograph.

All data were taken under steady-state conditions. The overall compositions, which denote the mole frac-

tion  $y$  of more volatile component HCFC22, were 0.12, 0.23, 0.5, 0.75 and 0.9, and the mass velocities were 100, 200, 300 and 350 kg m<sup>-2</sup> s<sup>-1</sup>. The reduced pressure varied from 0.13 to 0.16 depending on the composition. The flow rate of the heating water was kept constant at 100 kg h<sup>-1</sup>. In order to obtain data for different heat fluxes, three sets of experiments were carried out—namely (a) the refrigerant was a subcooled liquid at the inlet of the test section and a superheated vapor at the exit, (b) the vapor quality  $x \cong 0.3$  at the inlet and the refrigerant was a superheated vapor at the exit, (c) the refrigerant was a subcooled liquid at the inlet and  $x \cong 0.7$  at the exit. Most of the data were taken under a counter-flow condition, which may be employed in the real system. For the purpose of extending a heat-flux range, the data for a parallel-flow condition were also obtained for the condition (a).

## 2.2. Data reduction

Most of thermodynamic properties of the mixtures, i.e. temperatures, densities, enthalpies and fugacities, were calculated by the modified Benedict–Webb–Rubin equation of state [13, 14]. The other properties including transport properties were calculated from those of components HCFC22 and CFC114 which were taken from refs. [15] and [16], respectively:

the specific heat capacity was evaluated as the average of both components by mass fraction (see refs. [17, 18] for further details), the surface tension by Macleod-Sugden correlation [19], the viscosities of liquid and vapor by the equation of Idoko *et al.* [20] and the method of Wilke [21], respectively, and the thermal conductivity of liquid by the method of Chen *et al.* [22].

In the flow boiling of mixtures the fluid temperature increases with the vapor quality. Assuming a thermodynamic equilibrium in the cross section, the fluid temperature  $T_{rc}$  and the vapor quality  $x$  at each axial location were calculated by a heat balance equation and the equation

$$x = \frac{\tilde{y}_b - \tilde{y}_l}{\tilde{y}_v - \tilde{y}_l} \quad (1)$$

where  $\tilde{y}_b$  is the mass fraction of HCFC22 of the bulk fluid. The mass fraction of the vapor  $\tilde{y}_v$  and liquid  $\tilde{y}_l$  and the enthalpies of the vapor and liquid were estimated by using the equation of state. The pressure at each location was estimated by the measured pressure drops. The above calculations were proceeded backward from the exit of the test section if the refrigerant was a superheated vapor at the exit.

The local heat transfer coefficient  $\alpha$  was defined by

$$\alpha = q/(T_{wi} - T_b) \quad (2)$$

where  $q$  is the local heat flux calculated from the mass flow rate and the temperature drop of water in each subsection, and  $T_{wi}$  the average inner wall temperature evaluated from the measured outside wall temperatures. The measured fluid temperature  $T_{rm}$ , instead of the calculated temperature  $T_{rc}$ , was used for the bulk fluid temperature  $T_b$  (the reason is discussed later). The inaccuracy of  $T_b$  is estimated to be  $0.5^\circ\text{C}$ . The error in wall temperature measurement is  $0.1^\circ\text{C}$  judged from a calibration of thermocouples. The uncertainty in heat flux due to the errors in flow rate and temperatures of heating water is estimated to be 4–5%. From these uncertainties, the accuracies of heat transfer coefficients are determined to be 15% for the counter flow and 25% for the parallel flow.

### 2.3. Experimental results

Figures 1(a) and (b) show measured temperature distributions along the test tube for the counter flow and the parallel flow, respectively. Illustrated are the temperature of heating water  $T_s$ , the measured refrigerant temperature  $T_{rm}$ , the refrigerant temperature calculated by using the equation of state  $T_{rc}$ , and the inner wall temperature  $T_{wi}$ . The heat flux  $q$  and the vapor quality  $x$  are also plotted in Fig. 1. Both in the counter- and parallel-flow conditions,  $T_{rc}$  is about 3 C higher than  $T_{rm}$  for higher quality. The deviation depended on the overall composition and was little affected by flow rates. When the fluid was saturated at the exit of the test section, the value of  $T_{rm}$  at the exit of the last subsection was approxi-

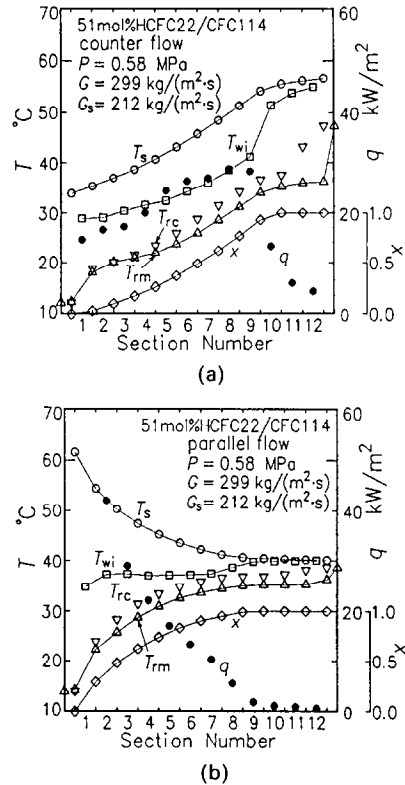


FIG. 1. Experimental results: (a) counter flow; (b) parallel flow.

mately equal to the bulk temperature measured at the exit mixing chamber; the difference was smaller than  $0.3^\circ\text{C}$ . On the other hand, the vapor–liquid equilibrium data from literature on which the parameters in the equation of state in the present work were based show a scatter of about  $\pm 2^\circ\text{C}$  [18]. Therefore the difference between  $T_{rc}$  and  $T_{rm}$  may be attributed to the uncertainty of the equation of state. For this reason, the measured temperature  $T_{rm}$  was used as the bulk fluid temperature  $T_b$  in the present study.

Figure 1 illustrates that  $q$  increases for the counter flow and decreases for the parallel flow with  $x$ . As compared to the results for pure refrigerants in the previous paper [12], the change in  $q$  with  $x$  is gentle for the counter flow and steep for the parallel flow, since the refrigerant temperature increases in the flow boiling of mixtures.

Figure 2 shows a comparison between the measured heat transfer coefficients and the values which are calculated with the authors' correlation equations for pure refrigerants [12] (following equations (3)–(18)) by using thermophysical properties of the mixture. Figures 2(a) and (b) correspond to the results shown in Figs. 1(a) and (b), respectively. The heat transfer coefficients  $\alpha_{BWR}$  which is defined by  $q/(T_{wi} - T_{rc})$  is also plotted in Fig. 2. The measured values of  $\alpha$  are much lower—about 50% in maximum deviation—than the calculated values. The values of  $\alpha_{BWR}$  are higher than  $\alpha$ , and the deviation is much larger for the

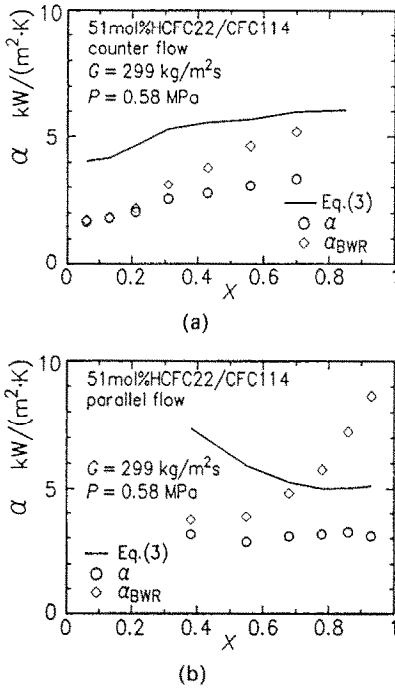


FIG. 2. Variation of heat transfer coefficient with quality: (a) counter flow; (b) parallel flow.

high-quality region. For the parallel flow shown in Fig. 2(b)  $\alpha_{\text{BWR}}$  is unreasonably higher than the calculated values because the temperature difference between wall and refrigerant is considerably small at the high-quality region.

### 3. CORRELATION EQUATIONS FOR PURE REFRIGERANT

In the present report, the application of the authors' correlation equations for pure refrigerant [12] is extended, taking into account of the mixture effect. Hence we first sum up the equations proposed in the previous paper [12].

The heat transfer coefficient  $\alpha$  is expressed as the sum of the convective heat transfer coefficient  $\alpha_{\text{cv}}$  and the nucleate boiling heat transfer coefficient  $\alpha_{\text{nb}}$ :

$$\alpha = \alpha_{\text{cv}} + \alpha_{\text{nb}} \quad (3)$$

The convective heat transfer coefficient  $\alpha_{\text{cv}}$  is given by

$$\alpha_{\text{cv}} = F\alpha_{\text{lo}} \quad (4)$$

where

$$\alpha_{\text{lo}} = 0.023 Re_{\text{lo}}^{0.8} Pr_1^{0.4} \lambda_l / d_i \quad (5)$$

$$F = F_Y = 1 + 2X_{\text{li}}^{-0.88} \quad (6)$$

$$Re_{\text{lo}} = G(1-x)d_i/\mu_l \quad (7)$$

Since the measured heat transfer coefficient for a single-phase liquid flow has been slightly higher than the Dittus-Boelter equation and the new empirical

correlation was obtained in the previous paper [12], equations (5) and (6) are replaced by the following equations in the evaluation of  $\alpha_{\text{cv}}$  for the present test evaporator:

$$\alpha_{\text{lo}} = 0.0116 Re_{\text{lo}}^{0.89} Pr_1^{0.4} \lambda_l / d_i \quad (8)$$

$$F = F_Y^{0.89, 0.8} = (1 + 2X_{\text{li}}^{-0.88})^{0.89, 0.8} \quad (9)$$

The nucleate boiling heat transfer coefficient  $\alpha_{\text{nb}}$  is given by

$$\alpha_{\text{nb}} = K^{0.745} S \alpha_{\text{pb}} \quad (10)$$

The heat-flux-fraction factor  $K$  is the ratio of heat flux due to nucleate boiling to total heat flux,  $q_{\text{nb}}/q$ . The suppression factor  $S$ , which was originally introduced by Chen [10], represents the effect of fluid flow on the nucleate boiling heat transfer as compared to the pool boiling. These values are calculated with the following equations:

$$K = \left( \frac{1}{1 + 0.875\eta + 0.518\eta^2 - 0.159\eta^3 + 0.7907\eta^4} \right)^{1/0.745} \quad (11)$$

$$\eta = \frac{\alpha_{\text{cv}}}{S\alpha_{\text{pb}}} \quad (12)$$

$$S = (1 - e^{-\zeta})/\zeta \quad (13)$$

$$\zeta = 3.3 \times 10^{-5} Ja^* 1.25 La \alpha_{\text{cv}}/\lambda_l \quad (14)$$

$$Ja^* = \frac{\rho_l c_{\text{pl}} T_{\text{sat}}}{\rho_v h_{\text{fg}}} \quad (15)$$

$$La = \sqrt{\left( \frac{2\sigma}{g(\rho_l - \rho_v)} \right)} \quad (16)$$

The nucleate pool boiling heat transfer coefficient  $\alpha_{\text{pb}}$  is obtained by

$$\alpha_{\text{pb}} = 1.35\alpha_{\text{SA}} \quad (17)$$

where  $\alpha_{\text{SA}}$  is the nucleate pool boiling heat transfer coefficient calculated with the correlation of Stephan and Abdelsalam [23]

$$\alpha_{\text{SA}} = 405 \lambda_l \left\{ \frac{g(\rho_l - \rho_v)}{2\sigma} \right\}^{0.5} \times \left( \frac{q La}{\lambda_l T_{\text{sat}}} \right)^{0.745} \left( \frac{\rho_v}{\rho_l} \right)^{0.581} Pr_1^{0.537} \quad (18)$$

It should be noted that the exponent 0.745 in equation (10) is identical with the exponent of heat flux in equation (18).

### 4. CORRELATION EQUATIONS FOR MIXTURES

The correlation equations for the annular-flow regime only will be proposed in the present work, since the equations (3)–(18) for pure refrigerants are based on the data of that regime. Table 1 lists the

Table 1. Experimental conditions for the correlated data

Mole fraction of HCFC22	0.11 ~ 0.12	0.22 ~ 0.23	0.49 ~ 0.51	0.74 ~ 0.75	0.91
Pressure [MPa]	0.40 ~ 0.47	0.50 ~ 0.52	0.47 ~ 0.59	0.54 ~ 0.81	0.61 ~ 0.81
Mass velocity [ $\text{kg m}^{-2} \text{s}^{-1}$ ]	223 ~ 348	221 ~ 357	214 ~ 393	222 ~ 357	218 ~ 310
Heat flux [ $\text{kW m}^{-2}$ ]	1.2 ~ 53.9	0.7 ~ 64.6	1.8 ~ 72.8	1.0 ~ 72.7	2.5 ~ 72.9

experimental conditions of the data for the correlation. The data for the mass velocity  $200 \text{ kg m}^{-2} \text{ s}^{-1}$  and quality smaller than 0.2 are excluded because they are considered to be in the stratified or wavy flow regime.

A physical model of the flow boiling of mixtures in the annular flow is illustrated in Fig. 3(a). Also illustrated are the distributions of temperatures and mole fractions of HCFC22 in the liquid film and the vapor core in Fig. 3(b), and the phase equilibrium diagram in Fig. 3(c). The mole fraction in the liquid phase  $y_l$  and that in the vapor phase  $y_v$  are those of equilibrium at the bulk temperature  $T_b$ . The subscript int denotes the value at the vapor–liquid interface. The theoretical analysis of the evaporation in annular two-phase flow reported by Yoshida *et al.* [24] confirmed that the temperature at the vapor–liquid interface  $T_{int}$  is higher than the bulk temperature. The heat fluxes  $q_{cv}$  and  $q_{nb}$  are those due to convection and nucleate boiling, respectively, and the sum of them is identical with the overall heat flux  $q$ . The heat transfer coefficient for mixtures is expressed by

$$\alpha = \alpha_{cvm} + \alpha_{nbm} \quad (19)$$

where  $\alpha_{cvm}$  is the convective heat transfer coefficient and  $\alpha_{nbm}$  is the nucleate boiling heat transfer coefficient.

#### 4.1. Convective heat transfer coefficient $\alpha_{cvm}$

Since  $T_{int}$  is higher than  $T_b$  as shown in Fig. 3, the effective temperature difference for convective heat transfer in the liquid film decreases. If we define the effective-temperature-difference factor  $S_{cvm}$  by

$$S_{cvm} = \frac{\Delta T_{int}}{\Delta T} = \frac{T_w - T_{int}}{T_w - T_b}, \quad (20)$$

the convective heat transfer coefficient for mixtures can be written as

$$\alpha_{cvm} = \frac{q_{cv}}{\Delta T} = S_{cvm} \frac{q_{cv}}{\Delta T_{int}}. \quad (21)$$

The temperature difference  $\Delta T_{int}$  for the mixture corresponds to  $\Delta T$  for the pure refrigerant. Hence  $q_{cv}/\Delta T_{int}$  in equation (21) is equivalent to the convective heat transfer coefficient  $\alpha_{cv}$  for the pure refrigerant, so that

$$\alpha_{cvm} = S_{cvm} \alpha_{cv} \quad (22)$$

where  $\alpha_{cv}$  is given by equation (4) using thermo-physical properties of the mixture.

The convective heat transfer coefficient  $\alpha_{cvm}$  is obtained experimentally with plotting  $\alpha/\alpha_{lo}$  against  $1/X_{tt}$  where  $\alpha_{lo}$  is the single-phase heat transfer coefficient for liquid only flow given by equation (8). Figure 4 shows the result for both counter and parallel flows for  $y = 0.5$ . The values of  $\alpha/\alpha_{lo}$  for  $1/X_{tt} > 10$  can be expressed by a single line irrespective of the flow direction, showing that they are in the purely convective region, i.e.  $\alpha = \alpha_{cvm}$ . The solid line in Fig. 4 denotes the value of  $F = F_Y^{0.89/0.8}$  calculated with equation (9), which indicates the ratio  $\alpha_{cv}/\alpha_{lo}$  for pure refrigerants. Using the data for  $1/X_{tt} > 10$  in the case of  $y = 0.5$ , the effective-temperature-difference factor  $S_{cvm}$  for convective heat transfer is therefore calculated by

$$S_{cvm} = \frac{\alpha_{cvm}}{\alpha_{cv}} = \frac{\alpha_{cvm}/\alpha_{lo}}{\alpha_{cv}/\alpha_{lo}} = \frac{\alpha/\alpha_{lo}}{F_Y^{0.89/0.8}}. \quad (23)$$

The factor  $S_{cvm}$  is evaluated in the same manner for the other compositions, and finally correlated in the

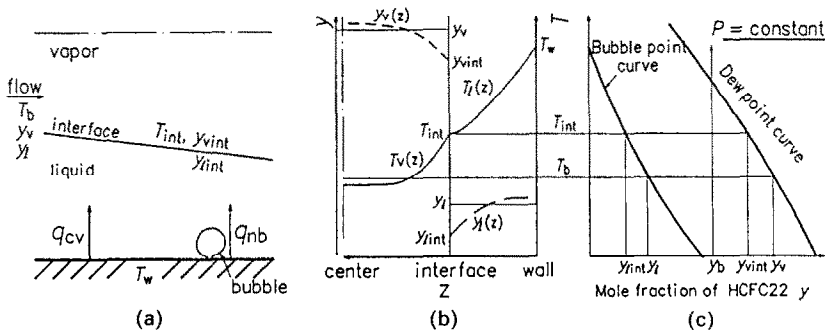


FIG. 3. Physical model of the flow boiling of mixtures in annular flow: (a) physical model; (b) distributions of temperatures and compositions; (c) phase equilibrium diagram.

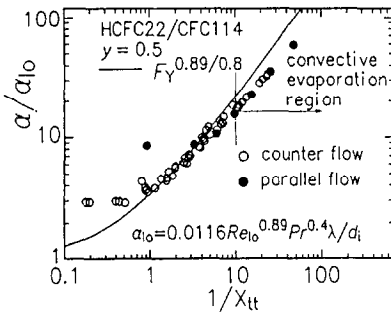


FIG. 4. Relation between  $\alpha/\alpha_0$  and  $1/X_{tt}$ .

form

$$S_{cvm} = \frac{1}{1 + a_0 x^{1+b} (1-x)^{-b} (|y_v - y_l| + a_1 |y_v - y_l|^2)} \quad (24)$$

Coefficients  $a_0$  and  $a_1$  were evaluated with the least squares method for given exponent  $b$  varied from 0 to 2. The best fitted coefficients which yielded minimum values of standard and maximum deviations were  $b = 0.39$ ,  $a_0 = 0.30$  and  $a_1 = 5.20$ . Equation (24) correlates the measured  $S_{cvm}$  with a maximum deviation of about 10% as shown in Fig. 5.

#### 4.2. Nucleate boiling heat transfer coefficient $\alpha_{nbm}$

Similar to the correlation for pure refrigerant, the nucleate boiling heat transfer coefficient  $\alpha_{nbm}$  for mixtures is expressed by

$$\alpha_{nbm} = K^{0.745} S_m \alpha_{pbm} \quad (25)$$

where the values for the mixture are distinguished from the pure fluid by subscript  $m$ .

The effective superheat for the nucleate pool boiling is smaller than that of pure fluid due to the mass transfer resistance in the liquid near the bubble surface [25]. Stephan and Körner [26] have correlated the wall superheat for seventeen kinds of mixtures except refrigerants by

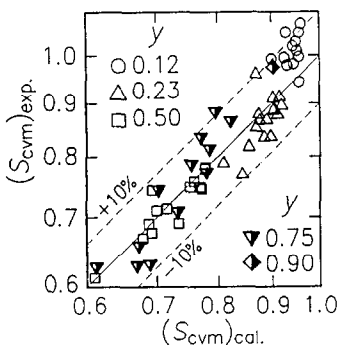


FIG. 5. Correlation of  $S_{cvm}$ .

$$\Delta T = \{1 + A_0(0.88 + 0.12p)|y_v - y_l|\} \Delta T_{id} \quad (26)$$

where  $p$  is the pressure in [bar] and

$$\Delta T_{id} = y_l \Delta T_1 + (1 - y_l) \Delta T_2. \quad (27)$$

The temperature differences  $\Delta T_1$  and  $\Delta T_2$  are those of pure components 1 and 2, respectively, for saturated pool boiling. They are evaluated for the pressure and the heat flux which are the same as those of the mixture. Using the definition of heat transfer coefficients and substituting equation (27) into equation (26) lead to the following expression:

$$\alpha_{pbm} = \frac{\alpha_{pb1} \alpha_{pb2}}{\alpha_{pb2} y_l + \alpha_{pb1} (1 - y_l)} \left( \frac{1}{1 + A_1 |y_v - y_l|} \right). \quad (28)$$

Heat transfer coefficients  $\alpha_{pb1}$  and  $\alpha_{pb2}$  are those of pure components 1 (HCFC22) and 2 (CFC114), respectively, for saturated pool boiling which are calculated with equation (17). Here in equation (28), the multiplier for composition difference in equation (26) is reduced to an empirically fitted coefficient  $A_1$ . It should be noted that, since the heat transfer coefficient  $\alpha_{pbm}$  for pool boiling of mixture is proportional to 0.745 power of heat flux, the exponent of  $K$  in equation (25) has been 0.745.

Next we consider the suppression of nucleate boiling in forced convective flow for the mixture based on the analysis of Bennett *et al.* [27] for pure fluids. We first approximate the liquid temperature profile near the wall surface by

$$T(z) - T_{int} = (T_w - T_{int}) \exp\left(-\frac{\alpha_{cv} z}{\lambda_1}\right) \quad (29)$$

where  $z$  is the normal distance from the wall surface. Equation (29) coincides with the expression for pure fluids if the interface temperature  $T_{int}$  is replaced by  $T_b$ . Defining the superheat by  $T - T_b$ , the average superheat in the bubble layer becomes

$$\begin{aligned} \Delta \bar{T}|_\delta &= \frac{1}{\delta} \int_0^\delta \{T(z) - T_b\} dz \\ &= (T_w - T_b) S_{cvm} \frac{\lambda_1}{\alpha_{cv} \delta} \left[ 1 - \exp\left(-\frac{\alpha_{cv} \delta}{\lambda_1}\right) \right] \\ &\quad + (T_{int} - T_b) \end{aligned} \quad (30)$$

where  $\delta$  is the thickness of bubble growth region. The suppression factor  $S_m$  is defined as the ratio of  $\Delta \bar{T}|_\delta$  to  $\Delta T$ , so that

$$\begin{aligned} S_m &= \Delta \bar{T}|_\delta / \Delta T \\ &= S_{cvm} \frac{\lambda_1}{\alpha_{cv} \delta} \left[ 1 - \exp\left(-\frac{\alpha_{cv} \delta}{\lambda_1}\right) \right] \\ &\quad + (1 - S_{cvm}). \end{aligned} \quad (31)$$

Assuming that  $\delta$  is estimated by the same equation as for pure refrigerants, the resulting equation for  $S_m$  becomes

$$S_m = S_{cvm} \frac{1}{\xi} (1 - e^{-\xi}) + (1 - S_{cvm}) \quad (32)$$

where  $\xi$  is given by equation (14).

The definition of  $K$  in equation (25) is the same as pure refrigerants: defined as the ratio of heat flux due to nucleate boiling to total heat flux. It is therefore evaluated by equation (11) with the value  $\eta$  given by the following expression for mixtures:

$$\eta = \frac{\alpha_{cvm}}{S_m \alpha_{pbm}} \quad (33)$$

Finally, the heat transfer coefficient for mixtures can be calculated with equations (4), (7)–(9), (11), (14)–(19), (22), (24), (25), (28), (32) and (33) for given coefficient  $A_1$ . The coefficient  $A_1$  is determined to be 4.8 from the comparison of measured heat transfer coefficients with those calculated for various values of  $A_1$ .

## 5. COMPARISON BETWEEN CORRELATION EQUATIONS AND EXPERIMENTS

### 5.1. Comparison with the present experiment

Figure 6 shows a comparison between measured heat transfer coefficients and those predicted by the correlation of Jung *et al.* [7]. While the experimental data for  $y = 0.12$ – $0.75$  agree well with the correlation, those for  $y = 0.9$  are more than 50% higher than calculated values. The mean deviation for all the data defined by

$$MD = \frac{1}{n} \sum \frac{|\alpha_{exp} - \alpha_{cal}|}{\alpha_{cal}} \times 100 \quad [\%] \quad (34)$$

is 19.1% and the average deviation defined by

$$AD = \frac{1}{n} \sum \frac{\alpha_{exp} - \alpha_{cal}}{\alpha_{cal}} \times 100 \quad [\%] \quad (35)$$

is 15.6%.

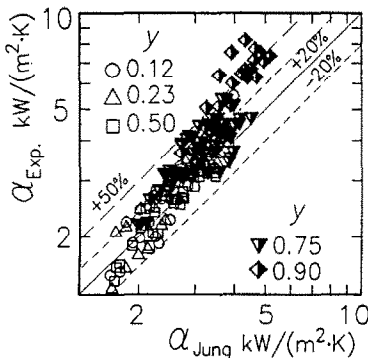


FIG. 6. Comparison between measured heat transfer coefficients and those calculated with the correlation of Jung *et al.* [7].

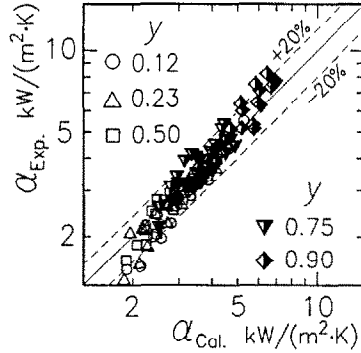


FIG. 7. Comparison between the present correlation and the measured heat transfer coefficients.

Figure 7 shows a comparison of the present correlation with measured heat transfer coefficients. Most of the data are correlated within a deviation of 20%. Table 2 lists the mean deviation MD and the average deviation AD. The average deviation varies with the mole fraction from negative to positive values, because the correlation for pure refrigerant overpredicts for CFC114 and underpredicts for HCFC22. The mean deviation, however, is 8.9% for all the data and this is considerably smaller than the correlation of Jung *et al.* [7].

The measured and calculated heat transfer coefficients are shown against the quality in Figs. 8(a) and (b) for the counter flow and the parallel flow, respectively. Also illustrated are changes in  $S_{cvm}$  and  $S_{pbm}$ . The factor  $S_{pbm}$  is calculated by

$$S_{pbm} = \alpha_{pbm} / \alpha_{pb} \quad (36)$$

where  $\alpha_{pbm}$  is obtained by equation (28) for mixtures, and  $\alpha_{pb}$  is estimated by equation (17) for pure refrigerants using thermophysical properties of the mixture. In the figures, the difference  $y_2 - y_1$  is also shown for reference. For both flow directions  $S_{pbm}$  is considerably smaller than  $S_{cvm}$ ; the pool boiling heat transfer coefficient decreases down to approximately one-half of that for pure refrigerant. It reveals that this severe degradation in pool boiling heat transfer causes considerably low heat transfer coefficient for mixtures as described in Fig. 2. As a result, the convective heat transfer is predominant for mixtures contrary to the results for pure refrigerants [12].

Table 2. Deviations between measured and calculated heat transfer coefficients

$y$	$n$	MD [%]	AD [%]
0.11 ~ 0.12	36	9.5	-9.1
0.22 ~ 0.23	37	10.9	-9.7
0.49 ~ 0.51	45	7.0	-6.2
0.74 ~ 0.75	48	6.4	2.3
0.91	22	9.1	7.5
all	188	8.9	-2.5

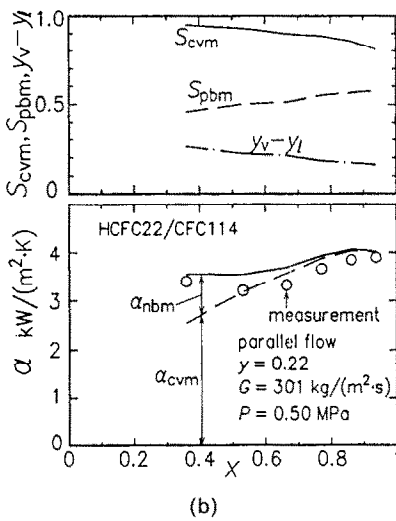
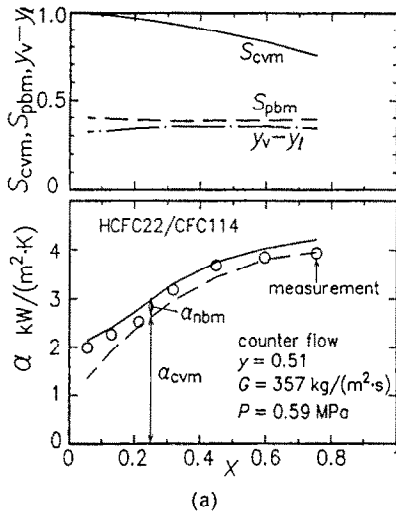


FIG. 8. Calculated results: (a)  $y = 0.51$ , counter flow; (b)  $y = 0.22$ , parallel flow.

5.2. Comparison with the other experiment

In Fig. 9, the present correlation is compared to the experimental data for the HCFC22/CFC114 mixture which are taken from the list in ref. [28]. The ranges

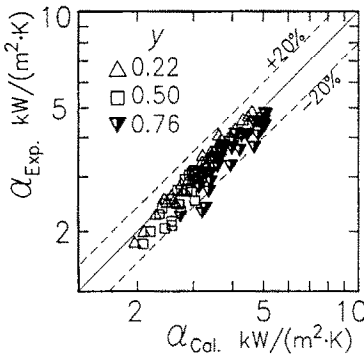


FIG. 9. Comparison between the present correlation and the experimental data obtained by Jung [28].

of the conditions are: inside diameter of the test tube 9 mm, mole fractions of HCFC22 approximately 0.22, 0.5 and 0.76, reduced pressures 0.08 (corresponding to around 0.3 MPa for  $y = 0.22$  and 0.4 MPa for  $y = 0.76$ ), mass velocities 350 to 520  $\text{kg m}^{-2} \text{s}^{-1}$ , and heat fluxes 17 to 45  $\text{kW m}^{-2}$ . The values taken from the list to calculate heat transfer coefficients with present equations are the overall composition, the pressure, the mass velocity of the refrigerant, the heat flux, the quality and the compositions of vapor and liquid phases. The thermophysical properties are estimated by the method used in the present study. In the prediction with the present equations, as equations (8) and (9) for  $\alpha_{cv}$  include some specific heat transfer characteristics of the present test section, they are replaced by equations (5) and (6). This revised set of correlation equations predicts most of the experimental data within a deviation of 20%. The mean and average deviations are 7.1% and -6.1%, respectively, for all the data whose mass velocities are larger than 350  $\text{kg m}^{-2} \text{s}^{-1}$ .

6. CONCLUSIONS

The local heat transfer coefficients were measured for the boiling of HCFC22/CFC114 mixtures (mole fractions of HCFC22 were 0.12, 0.23, 0.5, 0.75 and 0.9) flowing inside a 7.9 mm ID horizontal smooth tube. Using a water-heated, double-tube type evaporator, the test was carried out for both counter and parallel flows. The conclusions are:

(1) The refrigerant temperature calculated with the equation of state is approximately up to 3°C higher than the measured temperature. It is mainly attributed to the uncertainty of the equation of state, and thereupon a much more precise equation is needed for studies of heat transfer with mixtures. In the present study, the measured temperature is used for the bulk temperature of refrigerants.

(2) The heat transfer coefficient for mixtures is much lower—approximately 50% in maximum deviation—than the values which are estimated with our correlation equations for pure refrigerants by using thermophysical properties of the mixture.

(3) Based on the correlation equations for pure refrigerants, a set of correlation equations is proposed for refrigerant mixtures. It correlates the present data for the annular-flow regime ( $G \cong 200\text{--}300 \text{ kg m}^{-2} \text{ s}^{-1}$ ,  $x > 0.2$ ) with a mean deviation of 8.9%.

(4) The experimental data of Jung [28] can be expressed with a mean deviation of 7.1% by the revised set of equations in which the empirical correlation for convective heat transfer coefficient, equations (8) and (9), are replaced by equations (5) and (6). These revised equations are recommended for the prediction of heat transfer coefficients in evaporators.

(5) The present correlation reveals that the convection heat transfer is predominant due to the suppression of nucleate boiling for mixtures.



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