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A correlation for forced convective boiling heat transfer of pure refrigerants in a horizontal smooth tube

H. TAKAMATSU,† S. MOMOKI‡ and T. FUJII†

† Institute of Advanced Material Study, Kyushu University, 6-1 Kasugakohen, Kasuga 816, Japan
‡ Department of Mechanical System Engineering, Nagasaki University, 1-4 Bunkyomachi, Nagasaki 852, Japan

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Abstract—An experimental study is reported on the boiling heat transfer of HFC134a, HCFC22, CFC114 and CFC12 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. Based on the supposition of Chen that the total heat flux is represented as the sum of forced convective contribution and nucleate boiling contribution, a correlation equation is proposed for the data in the annular-flow regime. The mean deviation between the calculated and measured heat transfer coefficients is 12.2% for the present experimental data and 9.5% for the data available from literature. The proposed correlation shows that the nucleate boiling is not fully suppressed even in the high-quality region in the case of counter flow, while convective evaporation is dominant in the high-quality region with uniform heat flux condition.

1. INTRODUCTION

CHLOROFLUOROCARBONS (CFCs) have been widely used as the working fluids in the systems of refrigeration, air conditioning and heat pump. However, we have to replace the CFCs by alternative stratospherically-safe refrigerants. In the past few years, experiments of flow boiling heat transfer have been actively made with HFC134a which is a leading candidate for a substitute of CFC12. In the future, other new refrigerants will be used as working fluids. Under these circumstances, it is worth while to accurately predict heat transfer coefficients with some available correlations. Also the correlation for pure refrigerants is required as the base for refrigerant mixtures which will be used as an alternative to CFCs and/or to improve the performance of refrigerators, air conditioners and heat pumps.

During the past few decades, much work has been carried out on the flow boiling of refrigerants inside horizontal smooth tubes; many aspects of boiling heat transfer have been revealed and a lot of correlations for local heat transfer coefficients have been proposed for the design of evaporators. These correlations are classified into the following three types: (a) the heat transfer coefficient ratio of boiling two-phase flow to liquid only flow, α/α_{lo}, is expressed as a function of some dimensionless parameters such as the Boiling number and the Martinelli parameter, (b) the forced convection dominant region and the nucleate boiling dominant region are expressed individually, (c) the heat transfer coefficient is expressed as the sum of forced convection and nucleate boiling equations. The correlations proposed by Rhee and Young [1], Shah [2] and Kandlikar [3] belong to type (a), those of Lavin and Young [4], Dembi *et al.* [5], and Dhar *et al.* [6] to type (b), and those of Gungor and Winterton [7], Jung *et al.* [8] and Yoshida *et al.* [9] to type (c). The correlations of type (a) and (b) have been obtained purely empirically, while those of type (c) are based on the physical model that the total heat flux is represented as the sum of forced convective contribution and nucleate boiling contribution.

The objective of this work is to present a new correlation equation for pure refrigerants. We conducted a series of experiments of flow boiling heat transfer in a horizontal smooth tube with four kinds of pure refrigerants—HCFC22, CFC114, HFC134a and CFC12. Employing a water-heated, double-tube type evaporator, the local heat transfer coefficients were measured for both counter and parallel flows. The proposed correlation has the form of type (c) because it will be applied to refrigerant mixtures.

2. EXPERIMENTS

2.1. *Experimental apparatus and measurements*

A schematic view of the experimental apparatus is shown in Fig. 1. It is made up of four main loops: a refrigerant loop, two water loops and a brine loop. The refrigerant loop consists of a positive-displacement pump (1), a preheater (4), a test section (5), a rear heater (6), a condenser (7), and an auxiliary condenser (8). To exclude the effect of lubricating oil, the pump, instead of a compressor, was used for the circulation of refrigerant. The water loops are used to supply heating and cooling waters to the test section

NOMENCLATURE

c_p	isobaric specific heat capacity [$J\,kg^{-1}\,K^{-1}$]
d_i	inside diameter of test tube [m]
G	mass velocity [$kg\,m^{-2}\,s^{-1}$]
g	gravitational acceleration [$m\,s^{-2}$]
h_{fg}	latent heat of vaporization [$J\,kg^{-1}$]
K	heat-flux-fraction factor, q_{nb}/q
P	pressure [MPa]
Pr	Prandtl number
q	heat flux [$W\,m^{-2}$]
Re	Reynolds number, Gd_i/μ
Re_{lo}	Reynolds number for liquid only flow, $G(1-x)\,d_i/\mu_l$
S	suppression factor
T	temperature [°C]
x	vapor quality.

Greek symbols

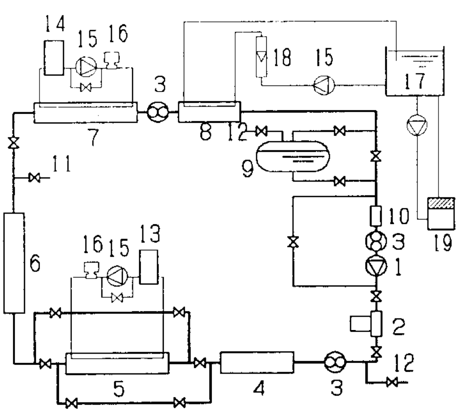
α	heat transfer coefficient [$W\,m^{-2}\,K^{-1}$]
ΔT	wall superheat [K]
λ	thermal conductivity [$W\,m^{-1}\,K^{-1}$]
μ	viscosity [Pa s]
ρ	density [$kg\,m^{-3}$]
σ	surface tension [$N\,m^{-1}$]
X_{tt}	Lockhart–Martinelli parameter, $\{(1-x)/x\}^{0.9}(\rho_v/\rho_l)^{0.5}(\mu_l/\mu_v)^{0.1}$.

Subscripts

b	bulk
cv	convection
l	liquid
lo	liquid only
nb	nucleate boiling
pb	nucleate pool boiling
r	refrigerant
s	heating water (heat source)
sat	saturated
v	vapor
w	wall.

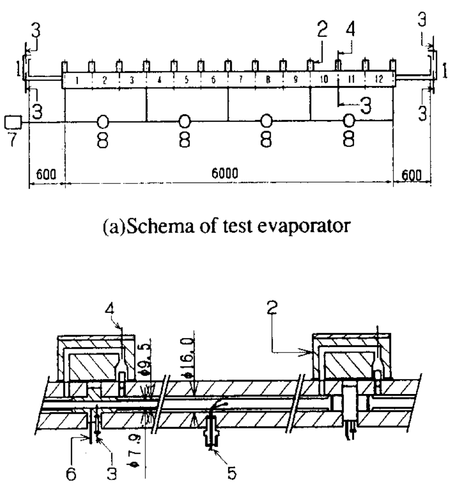
and the condenser, respectively. The brine loop is used to cool the refrigerant at the auxiliary condenser.

The test section shown in Fig. 2 is a double-tube type evaporator in which the refrigerant flows inside the inner tube and the heating water flows in the outer annulus. The inner tube is made of a straight smooth copper tube 6 m long, 7.9 mm inside diameter and 9.5 mm outside diameter. The outer tube is formed by two polycarbonate resin blocks each of which had a half-round ditch 8 mm in radius; so the width of the annular gap for water flow is 3.25 mm. The outer annulus is subdivided along the tube into twelve sub-



1 Circulation pump
2 Mass flow meter
3 Sight glass
4 Preheater
5 Test evaporator
6 Rear heater
7 Condenser
8 Auxiliary condenser
9 Liquid reservoir
10 Strainer
11 Sampling port
12 Charging port
13 Heat source unit
14 Heat sink unit
15 Pump
16 Gear-type flow meter
17 Brine tank
18 Float-type flow meter
19 Chilling unit

Fig. 1. Experimental apparatus.



(a)Schema of test evaporator

(b)Detail of subsection

1 Mixing chamber (refrigerant)
2 Mixing chamber (heating water)
3 Thermocouple (refrigerant)
4 Thermocouple (heating water)
5 Thermocouple (tube surface)
6 Pressure measuring port
7 Absolute pressure transducer
8 Differential pressure transducer

Fig. 2. Test evaporator: (a) schema of test evaporator; (b) detail of subsection.

sections by brass blocks to measure the local heat transfer rate. The effective heated length of each subsection is 0.46 m. An adiabatic entrance region 0.6 m in length is provided before the refrigerant inlet. The direction of refrigerant flow at the test section was switched with six valves so as to obtain the data for both counter- and parallel-flow conditions.

The bulk temperatures of the refrigerant were measured at the mixing chambers which were equipped 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 saturation temperature of refrigerants was confirmed 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 in three serial subsections were measured with an absolute pressure transducer and four differential pressure transducers, respectively (see Fig. 2). In the measurement of flow rates, a mass flow meter was used for the refrigerant and a gear type flow meter for the heating water. The heat transfer rate at each subsection was calculated from the temperature change and the flow rate of heating water.

2.2. *Experimental conditions and data reduction*

A series of experiments was carried out with pure HFC134a, HCFC22, CFC114 and CFC12. All data were taken under steady state conditions. The range of mass velocity was 100, 200, 200 and 350 kg m^{-2} s^{-1}. The reduced pressure varied from 0.13 to 0.23. The mass flow rate of heating water was kept constant at 100 kg h^{-1}. In order to obtain data for different heat fluxes, three sets of experiments were carried out for the counter-flow condition—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. The data for the parallel flow were also obtained only for the condition (a).

The local heat transfer coefficient α was defined by

$$\alpha = q/(T_{wi} - T_{sat}) \tag{1}$$

where q is the local heat flux calculated from the mass flow rate and the temperature drop of water in each subsection, T_{sat} the saturation temperature calculated from the measured pressure, and T_{wi} the average inner wall temperature calculated from the measured outside wall temperature by a radial heat conduction equation. The bulk enthalpy and the vapor quality of refrigerant at each axial location were calculated with a heat balance equation. This calculation was proceeded backward from the exit of the test section if the refrigerant was a superheated vapor at the exit. The physical properties were taken from the thermophysical property tables [10–13]. The error in wall temperature measurement is 0.1 K judged from the calibration of thermocouples. The uncertainty in heat flux due to errors in flow rate and temperatures of heating water is estimated to be 5%. The uncertainty in the saturation temperature depends on the accuracy of both the pressure measurement and the thermophysical tables. Those at $T_{sat} = 25°C$ are estimated to be about 0.10 K for HCFC22, 0.12 K for CFC114, 0.07 K for HFC134a and 0.09 K for CFC12. From these uncertainties, the accuracies of the heat transfer coefficients are determined to be 10–15% for counter-flow conditions and 20–25% for parallel-flow conditions.

2.3. *Experimental results*

Before the tests of flow boiling, several single-phase heated tests were performed for HCFC22 and CFC114. In this case, the heat transfer coefficient was defined by

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

where T_b is the bulk temperature of refrigerant calculated from the bulk enthalpy. The uncertainty in the bulk temperature is estimated to be approximately 0.05 K, so that the accuracy of the heat transfer coefficients is determined to be 11%. Figure 3 shows the relation between $Nu/Pr^{0.4}$ and Re. The relation evaluated with the well-known Dittus–Boelter equation is also shown in Fig. 3. The measured values scatter in the range of 5–35% higher than the Dittus–Boelter equation. The reason for these discrepancies is not clear at present, though the errors in temperatures and effective heat transfer length were examined. In the present work, we obtained the following correlation:

$$\alpha = 0.0116\, Re^{0.89}\, Pr^{0.4} \lambda_l/d_i. \tag{3}$$

Equation (3) agrees with measured single-phase heat transfer coefficients within a maximum deviation of 10%.

Figures 4(a) and (b) show temperature distributions along the test tube from the refrigerant inlet for HCFC22 for the counter and parallel flow, respectively. Illustrated are the temperature of heating water T_s, the measured refrigerant temperature T_r, the

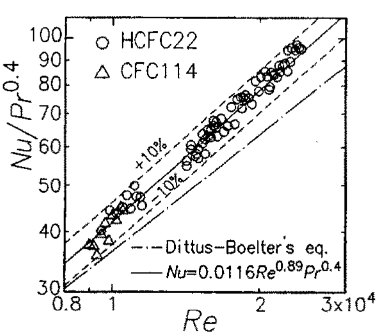


FIG. 3. Single-phase heat transfer coefficients.

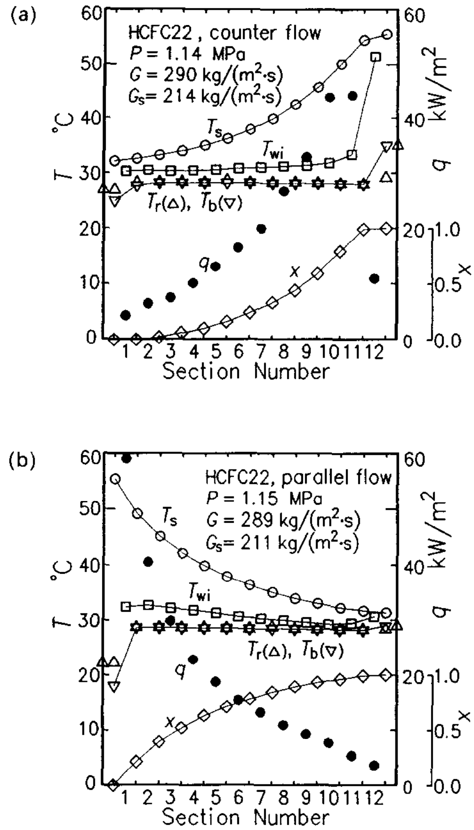


FIG. 4. Experimental results: (a) counter flow; (b) parallel flow.

calculated bulk temperature of refrigerant T_b, and the inner wall temperature T_{wi}. The heat flux q and the vapor quality x are also plotted in Fig. 4. The temperature difference between refrigerant and water increases with the progression of evaporation for the counter flow, while it decreases for the parallel flow. Hence the heat flux increases with quality for the counter flow and decreases for the parallel flow in contrast with the experiments by using electrically heated tubes.

The measured heat transfer coefficients α are plotted against vapor quality x in Fig. 5. The value of α increases with x for the counter flow. In the case of parallel flow, however, α first decreases in the low-quality region due to the reduction in nucleate boiling heat transfer with decreasing heat flux, and then increases slightly due to the enhancement in convective heat transfer caused by the increase in fluid velocity. Figure 5 also shows a comparison between measured heat transfer coefficients and those predicted by previous correlations, i.e. correlations proposed by Yoshida *et al.* [9], Jung *et al.* [8], Dembi *et al.* [5], Dhar *et al.* [6], Shah [2], and Gungor and Winterton [7]. Among these equations, that of Yoshida *et al.* shows better agreement with the present data, but it underpredicts the data in the high-quality region for parallel flow.

3. CORRELATION

In the horizontal flow boiling, the heat transfer characteristics are largely affected by the flow pattern—the annular flow or the stratified flow. Considering the application to the evaporator of air conditioners for rated load operation, the correlation is obtained for the annular-flow region in the present study. The correlated data are in the ranges of mass velocity larger than 200 kg m^{-2} s^{-1} and vapor quality larger than 0.2, since most of the data for mass velocity 100 kg m^{-2} s^{-1} appears to be stratified flow judged from the circumferential distribution of wall temperature. More detailed conditions are listed in Table 1.

As is described previously, we express the total heat

Table 1. Experimental conditions of the correlated data

Fluid	Pressure [MPa]	Mass velocity [kg m^{-2} s^{-1}]	Heat flux [kW m^{-2}]
HCFC22	0.69 ~ 1.15	221 ~ 358	1.9 ~ 75.8
CFC114	0.25 ~ 0.45	218 ~ 350	1.9 ~ 53.7
HFC134a	0.64 ~ 0.70	235 ~ 306	2.7 ~ 85.9
CFC12	0.63 ~ 0.64	305	2.6 ~ 36.8

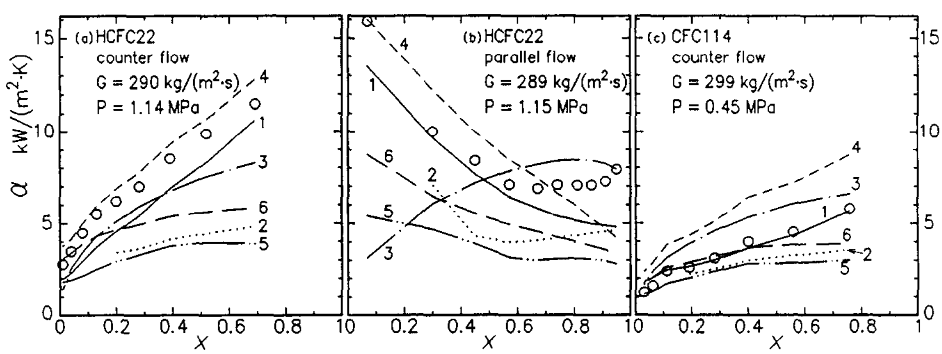


FIG. 5. Variation of heat transfer coefficient with quality (Symbol: Experiment, 1: Correlation of Yoshida *et al.* [9], 2: Jung *et al.* [8], 3: Dembi *et al.* [5], 4: Dhar *et al.* [6], 5: Shah [2], 6: Gungor–Winterton [7]). (a) HCFC22, counter flow; (b) HCFC22, parallel flow; (c) CFC114, counter flow.

flux q as the sum of forced convection contribution q_{cv} and nucleate boiling contribution q_{nb}:

$$q = q_{cv} + q_{nb}. \quad (4)$$

Hence the heat transfer coefficient α is expressed as

$$\alpha = \alpha_{cv} + \alpha_{nb} \quad (5)$$

where α_{cv} is the heat transfer coefficient due to forced convective contribution and α_{nb} is that due to nucleate boiling contribution.

3.1. *Convective heat transfer coefficient* α_{cv}

Following after Chen's correlation [14], the convective evaporation heat transfer coefficient α_{cv} is evaluated by

$$\alpha_{cv} = F\alpha_{lo} \quad (6)$$

where F represents the Reynolds number factor and α_{lo} is the heat transfer coefficient for liquid only flow. Since the present result for single-phase liquid flow is correlated by equation (3), α_{lo} is calculated by

$$\alpha_{lo} = 0.0116\, Re_{lo}^{0.89}\, Pr_l^{0.4} \lambda_l / d_i \quad (7)$$

where

$$Re_{lo} = G(1-x)d_i/\mu_l. \quad (8)$$

The factor F is determined experimentally with plotting α/α_{lo} against $1/X_{tt}$. The result is shown in Fig. 6 for all of the test fluids. The data for the counter flow scatter even in the range of large values of $1/X_{tt}$, i.e. in the high-quality region. This is attributed to the fact that the nucleate boiling is not fully suppressed due to high heat flux in the high-quality region as shown in Fig. 4(a). The data for the parallel flow however fall into a single line for $1/X_{tt} > 10$ and are considered as those of forced convection. As shown in Fig. 6, these data are correlated by

$$F = F_Y^{0.89/0.8} \quad (9)$$

where F_Y is the following correlation which has been proposed by Yoshida *et al.* [9] for their own data:

$$F_Y = 1 + 2X_{tt}^{-0.88}. \quad (10)$$

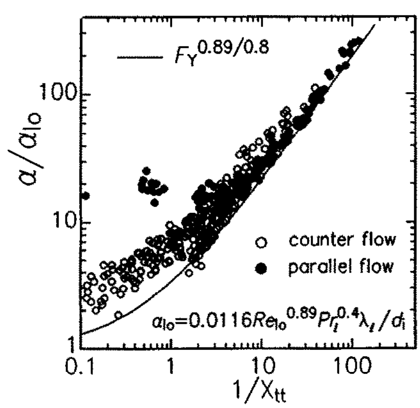


FIG. 6. Relation between α/α_{lo} and $1/X_{tt}$.

Since Yoshida *et al.* [9] calculated the heat transfer coefficient α_{lo} with the Dittus–Boelter equation, F_Y can be considered as

$$F_Y = (Re_{tp}/Re_{lo})^{0.8} \quad (11)$$

where Re_{tp} represents a two-phase Reynolds number. In the present work where α_{lo} is evaluated by equation (7), F is regarded as

$$F = (Re_{tp}/Re_{lo})^{0.89}. \quad (12)$$

Provided that Re_{tp}/Re_{lo} in equations (11) and (12) are identical, equation (9) can be derived.

Finally, the convective heat transfer coefficient α_{cv} is calculated by equations (6)–(10).

3.2. *Nucleate boiling heat transfer coefficient* α_{nb}

3.2.1. *Expression of nucleate boiling heat transfer.* The nucleate boiling heat transfer coefficient α_{nb} is estimated by the correlation equation for nucleate pool boiling, taking into account a degradation in effective wall superheat for nucleate boiling by fluid flow. When we define the suppression factor S as the ratio of the effective wall superheat ΔT_e to the actual wall superheat ΔT:

$$S = \Delta T_e/\Delta T, \quad (13)$$

the nucleate boiling heat transfer coefficient in convective flow becomes

$$\begin{aligned} \alpha_{nb} &= q_{nb}/\Delta T \\ &= (\Delta T_e/\Delta T)(q_{nb}/\Delta T_e) \\ &= S\alpha_{pb}|_{q=q_{nb}}. \end{aligned} \quad (14)$$

Assuming that the nucleate pool boiling heat transfer coefficient α_{pb} is proportional to the heat flux to the nth power, i.e. $\alpha_{pb} = Cq^n$, equation (14) yields

$$\begin{aligned} \alpha_{nb} &= SCq_{nb}^n \\ &= SK^n \alpha_{pb}|_{q=q} \end{aligned} \quad (15)$$

where K is a newly introduced heat-flux-fraction factor defined by

$$K = q_{nb}/q. \quad (16)$$

3.2.2. *Nucleate pool boiling heat transfer coefficient* α_{pb}. In this work, the nucleate pool boiling heat transfer coefficient α_{pb} is calculated by the correlation of Stephan and Abdelsalam [15]

$$\alpha_{SA} = 405\lambda_l \left\{\frac{g(\rho_l - \rho_v)}{2\sigma}\right\}^{0.5} \left(\frac{qLa}{\lambda_l T_{sat}}\right)^{0.745} \times \left(\frac{\rho_v}{\rho_l}\right)^{0.581} Pr_l^{0.533} \quad (17)$$

with an empirical multiplier C_l as

$$\alpha_{pb} = C_l \alpha_{SA}. \quad (18)$$

Equation (17) is obtained by substituting a contact angle of 35° in the original correlation for refrigerants, and La is the following characteristic length known as the Laplace constant:

$$La = \sqrt{\left(\frac{2\sigma}{g(\rho_l-\rho_v)}\right)}. \tag{19}$$

Thus from equations (17) and (18), the exponent n of K in equation (15) becomes 0.745.

3.2.3. *Suppression factor S.* Bennett *et al.* [16] have analyzed the suppression of nucleate boiling in forced convective flow and derived the following equation for the suppression factor S:

$$S = \frac{\lambda_l}{\alpha_{cv}\delta}\left[1-\exp\left(-\frac{\alpha_{cv}\delta}{\lambda_l}\right)\right] \tag{20}$$

where δ represents the thickness of bubble growth region and expressed by

$$\delta = C_d La. \tag{21}$$

Equation (21) has been obtained empirically with a supposition that δ is related to the bubble departure diameter. In this work, we additionally introduce the modified Jacob number

$$Ja^* = \frac{\rho_l c_{pl}}{\rho_v h_{fg}} T_{sat} \tag{22}$$

so as to consider the effect of pressure, so that

$$\delta = C_2 Ja^{*1.25} La \tag{23}$$

where C_2 is an empirically determined coefficient.

Then, substituting equation (23) into equation (20) gives

$$S = (1-e^{-\xi})/\xi \tag{24}$$

where

$$\xi = C_2 Ja^{*1.25} La\alpha_{cv}/\lambda_l. \tag{25}$$

3.2.4. *Heat-flux-fraction factor K.* Combining equations (4) and (16) gives

$$K = \frac{1}{1+\alpha_{cv}/\alpha_{nb}}. \tag{26}$$

Then, inserting equation (15) gives

$$K = \frac{1}{1+\eta K^{-n}} \tag{27}$$

where η is defined by

$$\eta = \frac{\alpha_{cv}}{S\alpha_{pb}}. \tag{28}$$

Equation (27) represents the relation between K and η implicitly. In the case of $n = 0.745$, this relation is approximated explicitly by

$$K = \left(\frac{1}{1+0.875\eta+0.518\eta^2-0.159\eta^3+0.7907\eta^4}\right)^{1/0.745}. \tag{29}$$

3.2.5. *Empirical coefficients C_1 and C_2.* Now, the heat transfer coefficient can be calculated with equations (5)–(10), (15), (17)–(19), (24), (25), (28) and (29) if only the empirical coefficients C_1 and C_2 are determined. The heat transfer coefficient was first calculated with an assumption that the nucleate boiling is not suppressed by the fluid flow, i.e. $S = 1$. Even in this limiting condition, the calculated heat transfer coefficient for HCFC22 became lower than the experimental data by approximately 20% when C_1 is taken as unity. It may be ascribed to the correlation of Stephan and Abdelsalam which underpredicts the nucleate pool boiling heat transfer coefficient for the present conditions. The coefficient C_1 was then determined tentatively to be 1.35 by the comparison between the calculated and experimental heat transfer coefficients.

Comparing calculated heat transfer coefficients for various values of C_2 with measured heat transfer coefficients, we obtained $C_2 = 3.3\times10^{-5}$ which yielded the best agreement with measured heat transfer coefficients. While the similar procedure was repeated with different values of C_1, no better agreement was obtained.

3.3. *Comparison with the present experiment*

Figure 7 shows a comparison between the measured and calculated heat transfer coefficients for all of the test fluids. While the measured values are slightly higher for HCFC22, most of the data are correlated within a deviation of 20%. Table 2 lists the mean deviation MD and the average deviation AD which are respectively defined by

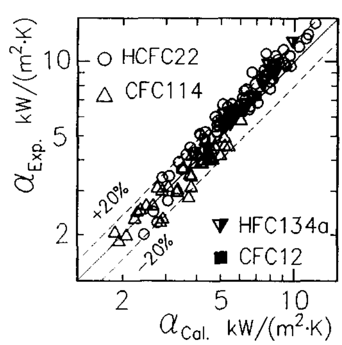


FIG. 7. Comparison between measured and calculated heat transfer coefficients.

Table 2. Deviations between measured and calculated heat transfer coefficients

	n	MD	AD
HCFC22	157	13.8	12.1
CFC114	80	9.4	−4.2
HFC134a	63	12.2	12.0
CFC12	4	5.1	2.8
Total	304	12.2	7.7

$$MD = \frac{1}{n}\sum^{n} \frac{|\alpha_{exp} - \alpha_{cal}|}{\alpha_{cal}} \times 100 \quad [\%] \tag{30}$$

and

$$AD = \frac{1}{n}\sum^{n} \frac{\alpha_{exp} - \alpha_{cal}}{\alpha_{cal}} \times 100 \quad [\%] \tag{31}$$

where n is the number of data. The mean and average deviations are 12.2% and 7.7%, respectively, for all the data. Only the average deviation for CFC114 is negative, because the correlation of Stephan and Abdelsalam underpredicts the nucleate pool boiling heat transfer coefficient for CFC114.

The measured and calculated heat transfer coefficients of HCFC22 are shown against quality in Figs. 8(a) and (b) for the counter flow and the parallel flow, respectively. Also illustrated are changes in the ratio α_{nb}/α—the contribution of nucleate boiling in total heat transfer—the suppression factor S and the value $K^{0.745}$. The ratio α_{nb}/α for the counter flow is approximately 0.5 even in the high-quality region, so that the nucleate boiling is not fully suppressed. In the parallel flow, however, α_{nb}/α decreases with increasing quality and the convection becomes predominant at higher qualities.

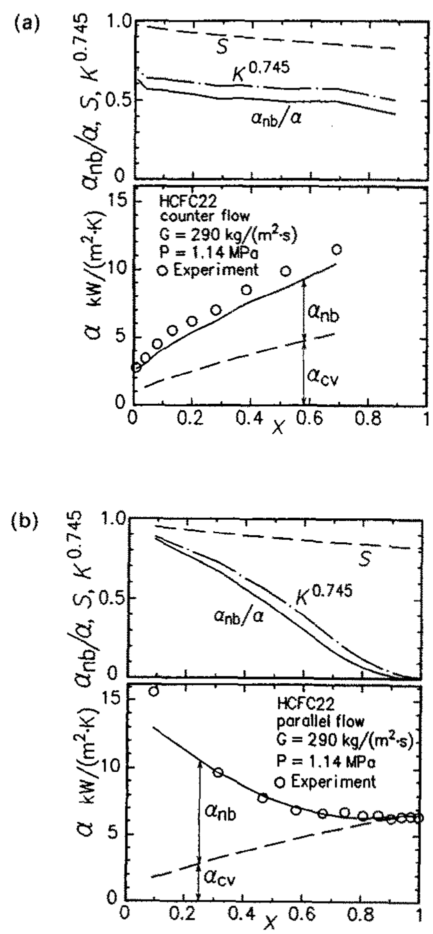


FIG. 8. Calculated result for HCFC22: (a) counter flow; (b) parallel flow.

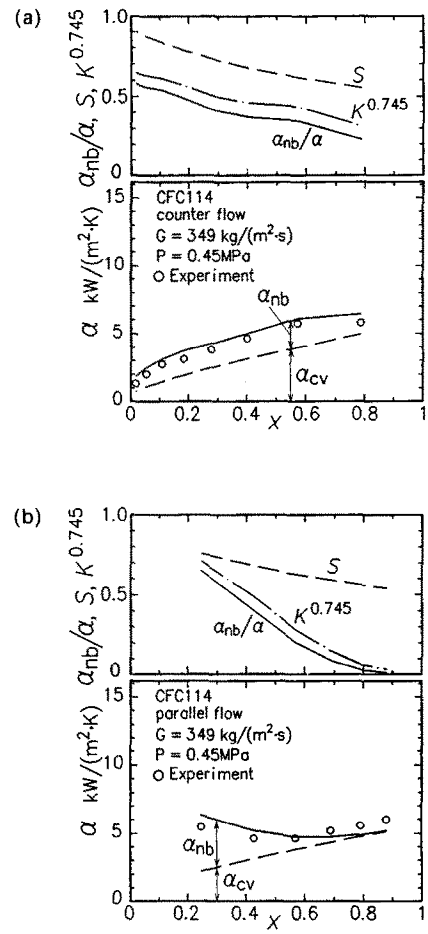


FIG. 9. Calculated result for CFC114: (a) counter flow; (b) parallel flow.

Figures 9(a) and (b) show the results for CFC114. The changes in the calculated values are similar to those of HCFC22, but the suppression factor S is smaller than that for HCFC22.

3.4. *Comparison with the previous experiments*

The present correlation is compared with a lot of previous experiments listed in Table 3. The collected data are those obtained with tubes heated by water or electrically heated tubes with diameters close to the present experiment. The present correlation overpredicts the data up to 15–30% in the mean deviation. This is due to the fact that the single-phase heat transfer coefficient of the present experiment was higher than the Dittus–Boelter equation as described previously. Hence equations (7) and (9) are replaced by

$$\alpha_{lo} = 0.023\, Re_{lo}^{0.8}\, Pr_l^{0.4} \lambda_l / d_i \tag{32}$$

and

$$F = F_Y = 1 + 2X_{tt}^{-0.88} \tag{33}$$

Table 3. Experimental data collected from previous research

No.	Authors	Fluid	P [MPa]	G [kg m^{-2} s^{-1}]	d_i [mm]	n
Data for copper tubes heated with water						
1	Khanpara *et al.* [17]	CFC113	0.33	250 ~ 520	8.8	11
2	Altman *et al.* [18]	HCFC22	1.02 ~ 1.07	280 ~ 560	8.7	6
3	Anderson *et al.* [19]	HCFC22	0.57	310 ~ 350	16.9	9
4	Takahashi *et al.* [20]	HCFC22	0.56 ~ 0.60	390 ~ 400	7.9	3
Data for electrically heated copper tubes						
5	Khanpara *et al.* [17]	CFC113	0.33	590	8.8	3
6	Yoshida [21]	HCFC22	0.59	300 ~ 500	11.0	41
7	Murata–Hashizume [22]	CFC114	0.2	200 ~ 300	10.3	12
Data for electrically heated stainless steel tubes						
8	Yoshida *et al.* [9]	HCFC22	0.59	200 ~ 400	10.6	6
9	Jung [23]	HCFC22	0.4 ~ 0.84	250 ~ 520	9.0	24
10	Jung [23]	CFC12	0.34 ~ 0.35	260 ~ 370	9.0	12
11	Jung [23]	HFC152a	0.36	250 ~ 530	9.0	24
12	Jung [23]	CFC114	0.26 ~ 0.53	260 ~ 520	9.0	27

respectively, when the correlation is compared to the previous experiments.

The comparison is shown in Fig. 10 and Table 4 together with the deviations yielded by the previous correlations. The correlation of Yoshida *et al.* [9] has a large mean deviation of 17.1%. While the deviation against all the data is smaller by the correlation of Jung *et al.* [8], it overpredicts the data of Altman *et al.* [18] and Yoshida *et al.* [21] up to more than 30%. The present correlation predicts the data with a mean deviation of 9.5%, although the correlations are compared with the data for HFC152a and CFC113 and those for $G > 350$ kg m^{-2} s^{-1}. Scattered values of average deviation also suggests that the heating method—heating by water or electrically—does not affect the heat transfer coefficients in the annular-flow regime.

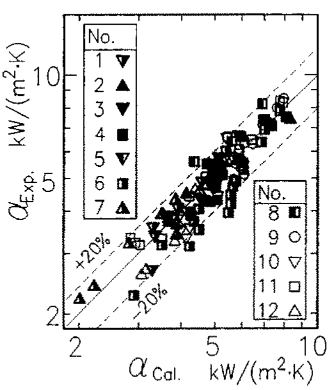


FIG. 10. Comparison between the present correlation and experimental data listed in Table 3.

Figures 11 and 12 compare the present correlations with the experimental heat transfer coefficients obtained by Jung *et al.* [23] and Murata and Hashizume [22], respectively. The changes in three parameters—α/α_{lo}, S and $K^{0.745}$—are also illustrated in the same manner as Figs. 8 and 9. The present correlation shows good agreement with the experiments. The calculated results for these uniform heat flux con-

Table 4. Deviations between the correlations and experimental data listed in Table 3

No.	n	Present work MD	Present work AD	Yoshida *et al.* [9] MD	Yoshida *et al.* [9] AD	Jung *et al.* [8] MD	Jung *et al.* [8] AD
1	11	8.4	−2.2	15.1	−14.6	7.6	6.3
2	6	7.4	−4.5	13.1	13.1	38.2	38.2
3	9	8.4	6.8	8.7	8.7	26.5	26.5
4	5	7.1	−0.8	16.9	−16.9	19.2	19.2
5	3	8.8	−8.8	11.9	−5.2	7.9	7.9
6	41	10.9	−9.9	14.8	−14.8	7.2	−0.8
7	12	10.8	6.6	17.3	−17.2	13.4	9.9
8	6	18.3	18.3	11.8	9.5	33.4	33.4
9	24	13.9	−4.8	21.9	−17.1	15.6	13.6
10	12	12.2	−7.6	29.8	−29.8	10.1	6.1
11	24	4.8	4.2	14.2	−14.2	3.5	−1.6
12	27	6.0	−3.6	19.3	−18.1	10.3	4.5
Total	180	9.5	−2.4	17.1	−13.7	12.1	7.9

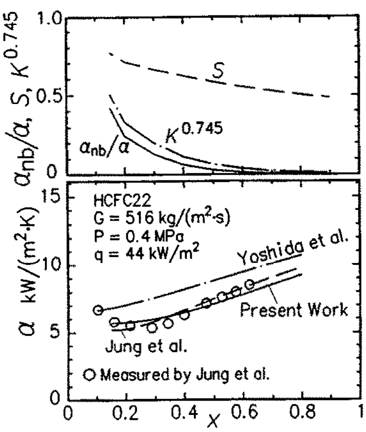


FIG. 11. Calculated result for the experimental data obtained by Jung [23].

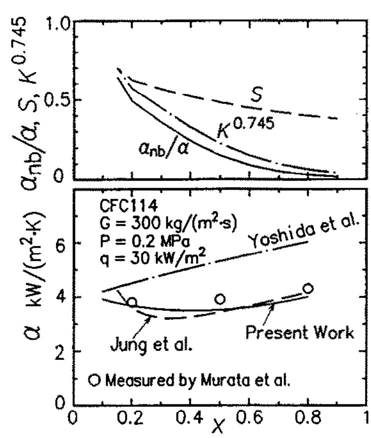


FIG. 12. Calculated result for the experimental data obtained by Murata and Hashizume [22].

ditions reveal that the ratio α_{nb}/α decreases with increasing quality, so that the convective evaporation governs the heat transfer in the high-quality region. Therefore, the previous data obtained by uniformly heated tubes may possibly exclude the data in which the nucleate boiling is not fully suppressed at high qualities, unless the experiments have been carried out specially under high-heat flux and high-quality conditions.

3.5. *Discussion*

In the present work, we used the correlation of Stephan and Abdelsalam [15] for nucleate pool boiling, in which the heat transfer coefficient is expressed as a function of heat flux. This nucleate pool boiling heat transfer coefficient should be calculated reasonably for the heat flux contributed by nucleate boiling which is not given beforehand. The introduction of the new parameter K in this work makes iterative calculations unnecessary when the total heat flux is given. Furthermore, it is reasonably accepted that the contribution of nucleate boiling to the total heat transfer varies with the ratio α_{cv}/α_{nb} as described by equation (26).

Since the nucleate boiling heat flux q_{nb} is expressed by

$$q_{nb} = \alpha_{nb}\Delta T = S\alpha_{pb}|_{q=q_{nb}}\Delta T, \tag{34}$$

combination of equations (17) and (18) gives

$$\alpha_{pb} = 405C_1\lambda_1\left\{\frac{g(\rho_1-\rho_v)}{2\sigma}\right\}^{0.5}\left(\frac{S\alpha_{pb}\Delta T\,La}{\lambda_1 T_{sat}}\right)^{0.745}\times\left(\frac{\rho_v}{\rho_1}\right)^{0.581}Pr_1^{0.533}, \tag{35}$$

so that

$$\alpha_{pb} = \left[405C_1\lambda_1\left\{\frac{g(\rho_1-\rho_v)}{2\sigma}\right\}^{0.5}\left(\frac{S\Delta T\,La}{\lambda_1 T_{sat}}\right)^{0.745}\times\left(\frac{\rho_v}{\rho_1}\right)^{0.581}Pr_1^{0.533}\right]^{1/0.255}. \tag{36}$$

Therefore, even in the problem where the heat flux is required for a given wall superheat ΔT, the heat transfer coefficient is calculated directly with equations (14) and (36) instead of equations (15), (17) and (18).

In the correlations of Gungor and Winterton [7], Jung *et al.* [8] and Yoshida *et al.* [9], the nucleate pool boiling heat transfer coefficient is also expressed as a function of heat flux. However, the nucleate pool boiling heat transfer coefficient has been evaluated for total heat flux. Hence the suppression factor S in their correlations is equivalent to the product of K^n and S in the present work. This is why the factor S in their correlations has been a function of the Boiling number, i.e. the heat flux. On the other hand, Chen [14] has used the correlation of Forster and Zuber [24] for nucleate pool boiling, in which the heat transfer coefficient is expressed by the wall superheat, and evaluated the convective evaporation heat transfer coefficient by $\alpha_{nb} = S\alpha_{pb}$. The suppression factor S therefore has the same definition as the present work. But the iterative calculation is required to estimate the heat transfer coefficient in the problem of the given heat flux.

4. CONCLUSIONS

The experiment was carried out on the boiling of pure refrigerants HFC134a, HCFC22, CFC114 and CFC12 flowing inside a 7.9 mm ID horizontal smooth tube. Using a water-heated, double-tube type evaporator, the local heat transfer coefficients were measured for both counter and parallel flows. A set of correlation equations is proposed based on the supposition that the heat transfer coefficient is expressed as the sum of convective contribution and nucleate boiling contribution. In the equations, the effects of nuclete boiling heat flux and effective wall superheat on nucleate boiling heat transfer are separately expressed by the newly introduced heat-flux-fraction factor K and the suppression factor S, respectively. The conclusions are:

(1) The proposed correlation equations correlates the present experimental data for the annular-flow regime ($G \cong 200$–350 kg m^{-2} s^{-1}, $x > 0.2$) with a mean deviation of 12.2%.

(2) The experimental data available from the previous papers can be expressed with a mean deviation of 9.5% by the revised set of equations in which the empirical correlation equation for convective heat transfer coefficient, equations (7) and (9), are replaced by equations (32) and (33). These revised equations are recommended for the prediction of heat transfer coefficients in evaporators.

(3) The nucleate boiling is not fully suppressed even in the high-quality region for the counter flow, while the convective evaporation becomes predominant with increasing quality for the data obtained by electrically and uniformly heated tubes.

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