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A study of flow boiling heat transfer with refrigerant mixtures

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Abstract—Mixture effects are studied on horizontal flow boiling heat transfer with both azeotropic and non-azeotropic refrigerant mixtures. More than 2000 local heat transfer coefficients are obtained with the azeotropic R12/R152a mixture and compared against the previously measured data with the non-azeotropic R22/R114 mixture. In a convective evaporation region, small mass transfer resistance is found for mixtures. The variation of physical properties due to mixing is responsible for almost all of the heat transfer degradation. In a partial boiling region, however, severe degradation of heat transfer with mixtures, similar to that in nucleate pool boiling heat transfer with mixtures, is found. A suppression of nucleate boiling at lower qualities due to loss of wall superheat with mixtures is responsible for this degradation. An analysis is developed to predict a transition quality by using Hsu's onset of nucleate boiling theory. The prediction agreed well with observed transition qualities for both pure and mixed refrigerants. Correlations, based on the supposition of Chen and using only phase equilibrium data to consider mixture effects, are developed with mean deviations of 7.2 and 9.6% for pure and mixed refrigerants.

1. INTRODUCTION

DURING the past few decades, much work has been carried out toward an understanding of boiling heat transfer with pure fluids. At present, many aspects of boiling heat transfer with pure components are well explored and reasonably good correlations have been developed for the design of efficient heat exchange equipment. With mixtures, boiling heat transfer is more complicated. Due in part to this complexity, the heat transfer problems in mixture systems have received less attention than those in pure component systems. The lack of such design data as heat transfer coefficients, especially in flow boiling heat transfer, is now one of the limitations in the design of efficient heat exchangers for use with mixtures [1, 2].

This work is a continuation of our previous study regarding the heat transfer characteristics of a non-azeotropic R22/R114 refrigerant mixture [3]. The R22/R114 mixture has a very large boiling temperature difference between the pure components as well as a large composition difference between the liquid and vapor phases in a two-phase region.

The previous study revealed that nucleate boiling was fully suppressed for both pure and mixed refrigerants beyond certain transition qualities. For pure fluids, the transition quality from a partial boiling to a convective evaporation region was shown to largely depend upon such fluid properties as liquid thermal conductivity and heat of evaporation. For mixtures of R22 and R114, nucleate boiling was suppressed at much lower qualities than those for pure fluids under the same flow conditions. The phenomenon of loss of wall superheat associated with mixtures was thought to be the most probable cause for the suppression of nucleate boiling at lower qualities.

The focus of the previous investigation was on heat transfer in the convective evaporation region (nucleation fully suppressed), typically qualities more than 30%. In this region, the heat transfer coefficients of the R22/R114 mixture were up to 36% lower than the ideal values calculated by a linear mole fraction weighting of pure component values (ideal mixing rule). The variation of physical properties due to mixing accounted for 80% of the total heat transfer degradation; the remainder was believed to be from mass transfer resistance in the convective evaporation region.

The dimensionless heat transfer coefficient and Martinelli's parameter, h_{tp}/h_{lo} and X_{tt}, were introduced to consider the variations in test condition and physical properties. The slope of Chen's F factor [4] was very similar to that of the h_{tp}/h_{lo} curve for pure refrigerants with its magnitude being smaller by 20–30%. The magnitude of the h_{tp}/h_{lo} curve for the R22/R114 mixture was up to 10% smaller than that for pure refrigerants with similar slope. The effects of the variables affecting the heat transfer in the evaporative region were properly considered by using the dimensionless parameters (h_{tp}/h_{lo} and X_{tt}). Thus, the difference in magnitudes of the h_{tp}/h_{lo} curves between pure and mixed refrigerants was inferred to be due to mass transfer resistance, an unaccounted variable.

For the R22/R114 mixture, a circumferential composition variation of up to 0.07 mole fraction in the annular liquid film was measured between the top

NOMENCLATURE

D	mass diffusivity [$m^2 s^{-1}$] or diameter [m]
F	heat transfer enhancement factor in equation (3)
G	mass flux [$kg\ m^{-2} s^{-1}$]
h	heat transfer coefficient [$W\ m^{-2} K^{-1}$]
h_{fg}	latent heat of vaporization [$kJ\ kg^{-1}$]
k	thermal conductivity [$W\ m^{-1} K^{-1}$]
$\dot{m}$	mass flow rate [$g\ s^{-1}$]
N	factor due to nucleate boiling
p	pressure [kPa]
q	heat flux [$W\ m^{-2}$]
R	universal gas constant [$Nm\ kmol^{-1} K^{-1}$] or bubble radius [m]
S	suppression factor in equation (3)
T	temperature [K or °C]
ΔT	wall superheat [°C]
X	liquid phase composition based on mole
x	quality
Y	vapor phase composition based on mole
y	distance from the wall [m].

Greek symbols

α	thermal diffusivity [$m^2 s^{-1}$]
μ	viscosity [Pa s]
ρ	density [$kg\ m^3$]
σ	surface tension [$N\ m^{-1}$]
ϕ	property and quality combination in equation (4).

Dimensionless numbers

Bo	boiling number, $q/(Gh_{fg})$
Pr	Prandtl number of liquid, $C_p\mu/k$
Re	Reynolds number, GD/μ
X_{tt}	Martinelli parameter, equation (5).

Subscripts

b	bottom or bulk
bub	bubble
c	critical
cal	calculated
cec	convective evaporation contribution
exp	experimentally determined
f	fluid
i	interface or ideal
l	liquid
lo	liquid only
m	mixture
max	maximum
nb	nucleate boiling
nbc	nucleate boiling contribution
ONB	onset of nucleate boiling
p	pure component
sb	sublayer
SNB	suppression of nucleate boiling
tp	two phase
v	vapor
w	wall.

and bottom of the heated tube. This explained the corresponding circumferential wall temperature variation seen with the R22/R114 mixture and with the R13B1/R152a mixture investigated by Ross *et al.* [5]. It was recommended that experiments with an azeotropic mixture be made in an attempt to explain the role of mass transfer resistance in flow boiling heat transfer.

Of primary concern in this study is the heat transfer characteristics of refrigerant mixtures in horizontal annular flow. Based upon the measured heat transfer data with both non-azeotropic and azeotropic mixtures, R22/R114 and R12/R152a, the role of mass transfer resistance on flow boiling heat transfer will be studied and compared to that in nucleate pool boiling heat transfer. An analytical method to predict the transition quality will be developed utilizing the onset of nucleate boiling theory, which will be followed by a correlation development. The nucleate boiling and convective evaporation contributions comprising the total heat transfer are separately considered. Finally, the mixture effects are taken into consideration individually in both contributions.

Figure 1 illustrates the temperature–composition diagram for the binary mixture, R12/R152a, which forms an azeotrope (R500) at roughly 60 mole% R12 at room temperature. In this study, composition means overall composition based on the mole fraction of the more volatile component. At the azeotrope the compositions of the liquid and vapor phases are identical; thus, the azeotrope behaves essentially like

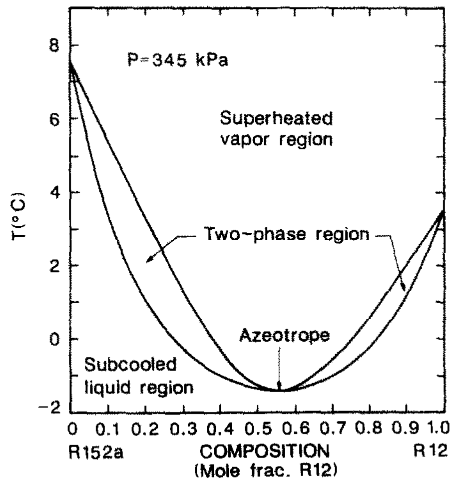

FIG. 1. Temperature–composition diagram for azeotropic R12/R152a mixture.

a pure fluid. Except for the azeotrope, however, a constant pressure evaporation or condensation would occur over a range of temperatures, known as the gliding temperature effect, as it would for a non-azeotropic mixture.

For an azeotropic mixture, it is reasonable to assume that the mixture effects on heat transfer would be absent at the azeotropic composition. On the other hand, at other compositions they would play a similar role as those associated with non-azeotropic mixtures. This can be illustrated by considering the results of nucleate pool boiling heat transfer with an azeotropic R22/R115 mixture [6] (Fig. 2). The nucleate pool boiling heat transfer coefficients for the R22/R115 mixture are lower than the ideal values except for the azeotrope, at which they are about the same as the ideal values.

In order to explain the reduction in nucleate pool boiling heat transfer with mixtures, many theoretical models for the bubble growth have been proposed. The resulting equations for the bubble radius for mixtures, R_m, fall into the form, first derived by Scriven [7]

$$R_m = \frac{R_p}{1-(C_{pl}/h_{fg})(\alpha/D)_l^{0.5}(Y-X)(dT/dX)_{bub}}. \quad (1)$$

The denominator in equation (1) is always greater than 1 for non-azeotropic mixtures since $(Y-X)$ and $(dT/dX)_{bub}$ have opposite signs; for an azeotrope the denominator is 1.

The implication of equation (1) is that the bubble growth rate for a mixture is lower than that for a pure fluid having otherwise identical physical properties. This is due to mass transfer resistance and loss of wall superheat via the mass diffusivity, D, and the composition difference between the two phases, $(Y-X)$. Many nucleate boiling correlations for mixtures have been proposed which account for the mixture effects by applying certain correction factors often including such terms as $(Y-X)$ and D.

To the authors' knowledge, there has been no flow boiling heat transfer measurements using azeotropic mixtures. Thus, the present study may reveal the fundamental aspects of mixture effects on flow boiling heat transfer by comparing the results obtained with both non-azeotropic and azeotropic mixtures from the same experimental setup.

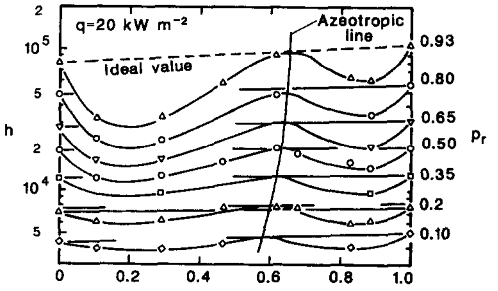

FIG. 2. Nucleate pool boiling heat transfer coefficients of azeotropic R22/R115 mixture (adapted from ref. [6]).

2. EXPERIMENTS

Since the same experimental apparatus used in the previous study [3] was employed during this study, a detailed description will not be given here. The reader is referred to refs. [3, 8] for the complete information. In this paper, only experimental parameters and data verification will be given.

All data were taken under steady-state conditions. The pressure at the outlet of the test section was kept at a reduced pressure of 0.08, corresponding to 330 and 360 kPa for pure R12 and R152a. The primary experimental parameters were overall composition, mass flow rate, heat flux, and quality. The overall compositions were 0, 21, 60 (azeotrope, R500), 89, and 100 mole% R12. Mass flow rates were within $\pm 5\%$ of the target values of 16, 23, 33, and 46 g s^{-1}, equivalent to 250–720 kg s^{-1} m^{-2} in terms of mass flux. Heat fluxes were 10, 17, 26, 36, and 45 kW m^{-2}. The quality ranged up to 95%.

The validity of the experimental setup was checked by comparing measured single phase heat transfer coefficients with existing correlations. Ten tests for pure and mixed refrigerants were performed with the subcooled liquid heated throughout the test section. The results were compared with the Dittus–Boelter [9] and Petukhov equations [10]. Both equations differed from each other only within 2% in correlating the present results with $\pm 5\%$ mean deviation. Due to the simplicity and popularity of the Dittus–Boelter equation, equation (2), it has been used throughout this study

$$h_l = 0.023\frac{k_l}{d}\left(\frac{Gd}{\mu_l}\right)^{0.8}\left(\frac{C_{pl}\mu_l}{k_l}\right)^{0.4}. \quad (2)$$

The good agreement with the accepted correlations indirectly substantiated the measurements of the power, mass flow rate, wall temperature, and pressure.

3. RESULTS AND DISCUSSION

A series of tests were performed with the R12/R152a mixture at several overall compositions. For the analysis of the data and comparison with the predictive methods, all thermodynamic properties for both pure and mixed refrigerants were calculated by the Carhahan–Starling–DeSantis equation of state [11]. Transport properties for pure refrigerants were taken from Johns [12] and Phillips and Murphy [13]. There are often large discrepancies in transport property values among different sources; special care was taken to select the best set of transport properties (see ref. [8] for further details).

For the calculation of transport properties of mixtures, the mixing rules suggested by Reid *et al.* [14] and Kandlikar *et al.* [15] were adopted. A detailed

Table 1. Physical properties of pure refrigerants at a reduced pressure of 0.08

Property	Unit	R22	R114	R12	R152a
Molecular weight	kg kmol^{-1}	86.47	170.92	120.91	66.05
Pressure	kPa	400	260	330	360
Saturated temperature	°C	−6.47	31.25	2.1	8.64
Liquid density	kg m^{-3}	1304	1435	1389	938
Vapor density	kg m^{-3}	17.1	19.3	19.2	11
Liquid specific heat	J kg^{-1} °C^{-1}	1155	1003	940	1644
Heat of evaporation	J kg^{-1}	211 000	124 100	150 800	300 000
Liquid thermal conductivity	W m^{-1} °C^{-1}	0.101	0.0626	0.0748	0.112
Liquid viscosity	Pa s	2.32×10^{-4}	3.36×10^{-4}	2.58×10^{-4}	2×10^{-4}
Vapor viscosity	Pa s	1.17×10^{-5}	1.19×10^{-5}	1.19×10^{-5}	9.6×10^{-6}
Surface tension	N m^{-1}	0.0128	0.01	0.0117	0.0116
Prandtl number		2.66	5.38	3.25	2.93

procedure for determining thermophysical properties of pure and mixed refrigerants is given elsewhere [8]. Table 1 lists only the properties of four pure refrigerants at a reduced pressure of 0.08. Since the properties of mixtures are a function of both composition and quality under the same pressure, it would be inappropriate to list them. Thus, the reader is referred to refs. [8, 11–15] for further details.

3.1. *Heat transfer characteristics of azeotropic R12/R152a mixtures*

Figure 3 illustrates the flow boiling heat transfer coefficients for a 21% R12/79% R152a mixture. For a given mass flow rate, the heat transfer coefficients at various heat fluxes are distinctly different at low qualities, merging into a single line beyond certain transition qualities. In Fig. 3, the heat flux dependence of heat transfer coefficients is emphasized by the dotted lines only for the flow rate of 32–33 g s^{-1}; it, however, is present at all other flow rates considered and for other R12/R152a mixtures at various compositions including pure components (this phenomenon is more clearly shown in Figs. 9 and 11). The same behavior was also observed for the R22/R114 mixture [3].

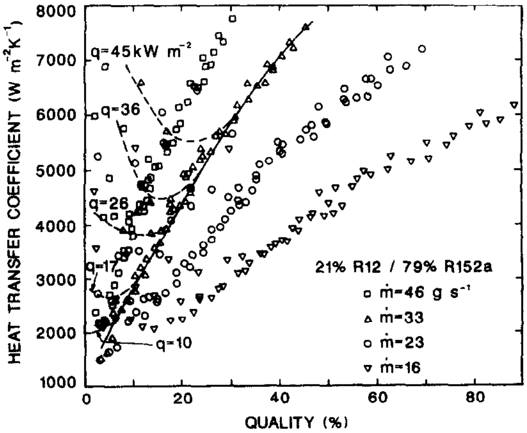

FIG. 3. Flow boiling heat transfer coefficients for 21% R12/79% R152a at $p_r = 0.08$ as a function of quality for various mass flow rates. (Heat flux dependence at low qualities is shown only for $\dot{m} = 32$–34 g s^{-1}.)

The above mentioned results indicate that there are two distinct heat transfer regions in flow boiling of refrigerants in the pressure range of 300–400 kPa. The first is a 'partial boiling' region occurring at low qualities in which heat transfer coefficients are a strong function of heat flux. Both the forced convective evaporation and nucleate boiling mechanisms were found to be responsible for the heat transfer in this region [4]. The rapid suppression of the latter even leads to a temporary reduction of the heat transfer coefficients with increasing quality as shown in Fig. 3, which was observed also with other refrigerants [3, 5]. The second is a 'convective evaporation' region beyond the transition quality where heat transfer coefficients are independent of heat flux.

Recently, Jensen and Bensler [16] have measured the heat transfer coefficients with R113 flowing in an 8.1 mm diameter, 1.2 m long stainless steel tube. They attributed the same phenomenon of decreasing heat transfer coefficients with increasing quality in the low quality region to a thermal entry length effect. The present results obtained from a long test section, however, do not support Jensen and Bensler's view of this behavior.

Instead, the present results advocate a conventional concept of the suppression of nucleate boiling with increasing quality in annular flow. As quality is increased in annular flow, the effective wall superheat decreases due to a thinner liquid film (less thermal resistance) and an enhanced convection caused by high vapor velocity. Thus, the number of active nucleation sites decreases till a transition quality is reached. Beyond the transition quality, the effective wall superheat is below the threshold value required for bubble nucleation on the wall.

Chen [4] proposed that there were two major contributions in flow boiling heat transfer, namely nucleate boiling and convective evaporation contributions, h_{nbc} and h_{cec}

$$h_{\mathrm{tp}} = h_{\mathrm{nbc}} + h_{\mathrm{cec}} = Sh_{\mathrm{nb}} + Fh_{\mathrm{lo}} \quad (3)$$

where h_{nb} is a nucleate boiling heat transfer coefficient originally taken from Forster and Zuber [17] and h_{lo} is a single-phase heat transfer coefficient for liquid only flow, which is obtained by replacing G in equation (2) with $G(1-x)$.

Under the suppression of nucleate boiling, it was shown [3] that h_{tp} is directly proportional to $\dot{m}^{0.8}$ or $G^{0.8}$. The heat transfer coefficients for both pure and mixed refrigerants of R12 and R152a at qualities above the transition points follow a similar trend of $h_{tp} \propto \dot{m}^{0.8}$, confirming that nucleate boiling is suppressed.

Of special interest is the prediction of the transition quality. In the purely convective region, the heat transfer process might be modelled by considering the single-phase heat transfer alone with a certain enhancement factor [5]. The prediction of the transition quality requires the accurate value of the two-phase heat transfer enhancement factor, F in equation (3), to determine the temperature difference in the purely evaporative flow. Thus, the data in the evaporative region will be considered first to determine the accurate F factor. Then, the nucleate boiling contribution will be determined by subtracting the convective evaporation portion from the measured h_{tp}.

Figure 4 illustrates the heat transfer coefficients of the R12/R152a mixture in the evaporative region as a function of composition for various qualities. The heat transfer coefficients for R152a are shown to be approximately 70% higher than those for R12. In the evaporative region, it was shown [3] that h_{tp} can be expressed as

$$h_{tp} \propto k_l^{0.6}(C_{pl}/\mu_l)^{0.4}(1-x)^{0.8}F(X_{tt}) = \phi F(X_{tt}) \quad (4)$$

where

$$\phi = k_l^{0.6}(C_{pl}/\mu_l)^{0.4}(1-x)^{0.8}.$$

For similar fluids, the F factor in equation (4) could be well correlated by using a reduced pressure and quality [3]. Consequently, under the same reduced pressure and quality, h_{tp} in equation (4) can be predicted by evaluating only the property group, $k_l^{0.6}(C_{pl}/\mu_l)^{0.4}$. Under a reduced pressure of 0.08, the property group for R152 is found to be 76% greater than that for R12, indicating a good agreement with experimental data.

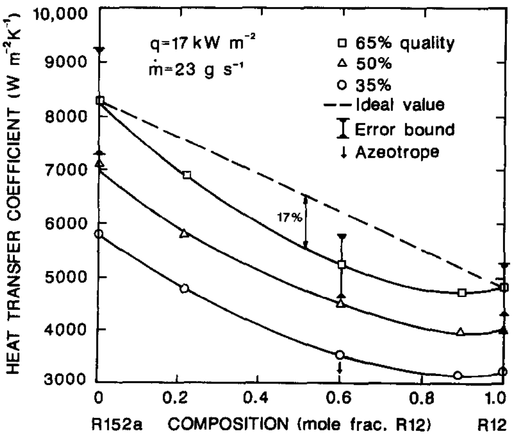

FIG. 4. Heat transfer characteristics of pure and mixed refrigerants of R12 and R152a in the convective evaporation region at $p_r = 0.08$.

The maximum degradation of measured heat transfer coefficients from the ideal value, 17%, occurs in the composition range of 0.45–0.55. Following the same procedure presented in ref. [3], $\phi^*F(X_{tt})$, a direct measure of the heat transfer coefficient in the evaporative region as in equation (4), of the R12/R152a mixture is evaluated. The results are shown in Fig. 5 at a reduced pressure of 0.08. In evaluating transport properties of the mixture, local equilibrium liquid and vapor phase compositions were used rather than overall compositions since the compositions of both phases vary with quality.

As composition changes $\phi^*F(X_{tt})$ varies smoothly and does not rise to the ideal values near the azeotrope. This is in contrast with the nucleate pool boiling heat transfer results shown in Fig. 2. The maximum degradation of 16% from the ideal values is predicted at roughly 0.5 mole fraction R12. This indicates that in the evaporative region the variation (actually degradation) of the physical properties due to mixing alone accounts for almost all of the heat transfer degradation. In other words, had the mixture been treated as a pure fluid with the properties calculated by proper mixing rules, there would have been negligible heat transfer degradation in the evaporative region.

The good agreement between the experimental and analytical results indicates that for the R12/R152a mixture mass transfer resistance is negligible in the evaporative region. The results in Fig. 4 also support this finding. Had the R12/R152a mixture possessed a strong mass transfer resistance, the experimental data in Fig. 4 would have gone up close to the ideal values near the azeotrope as shown in Fig. 2. The measured heat transfer coefficients vary smoothly over the entire composition range in close agreement with the prediction in Fig. 5.

The present result with the R12/R152 mixture is in contrast with the previous result with the non-azeotropic R22/R114 mixture [3]. The previous study revealed that mass transfer resistance was responsible for roughly 20% of the total degradation in the heat transfer coefficients of the R22/R114 mixture in the evaporative region.

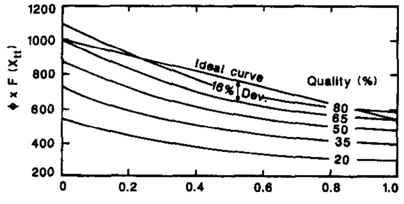

FIG. 5. Variation of $\phi^*F(X_{tt})$ for an R12/R152a mixture at $p_r = 0.08$.

The discrepancy may be explained as follows: mass transfer resistance for the R22/R114 mixture would be much larger than that for the R12/R152a mixture because of the larger composition difference between the phases, $(Y-X)$ (i.e. $(Y-X) = 0.45$ for R22/R114 vs 0.11 for R12/R152a). Thus, it can be tentatively concluded that in the absence of nucleate boiling the heat transfer coefficients of mixtures having $(Y-X) < 0.11$ (such as R12/R152a) could be predicted well by treating these fluids as equivalent pure fluids without consideration of mass transfer resistance.

Ross *et al.* [5] discovered that for the mixture of R13B1/R152a the wall temperature at the top was higher than that at the bottom in annular flow, the reverse of the behavior seen with pure fluids. The same phenomenon was observed for the R22/R114 mixture in ref. [3]. It was experimentally verified [3] that the circumferential composition variation was responsible for this unique behavior. The R12/R152a mixture at all compositions, however, exhibited a circumferential wall temperature variation similar to that for a pure fluid even though it was a mixture.

The results from sampling the local annular liquid film (as described in refs. [3, 8]) indicated no circumferential composition variation for the R12/R152a mixture at various compositions including the azeotrope. This is probably due to the fact that the volatility difference between the pure components was not strong enough to cause the circumferential composition variation. In this regard, it may be interesting to perform experiments with mixtures having $(Y-X)$ of roughly 0.2 in an attempt to find a maximum value of $(Y-X)$ below which mixtures exhibit no circumferential composition variation.

3.2. *Determination of two-phase enhancement factor F*

Since the present results indicate the suppression of nucleate boiling for pure and mixed refrigerants of R12 and R152a, a plot of dimensionless heat transfer coefficients, h_{tp}/h_{lo}, against a reciprocal of the Martinelli parameter, $1/X_{tt}$, is made in order to take into account the effects of various test conditions and a physical property variation. Figures 6 and 7 show the results for pure and mixed refrigerants of R12 and R152a. In these figures, h_{tp} is a measured heat transfer coefficient and X_{tt} is given as

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1}. \tag{5}$$

Also shown in these figures are Chen's F factor [4] and the curve fit of Kenning and Hewitt's nondimensionalized water data [18] obtained under the suppression of nucleate boiling. The dimensionless heat transfer coefficients for pure refrigerants are scattered at low qualities (the partial boiling region, indicated by the area enclosed by dashed lines); they fall into a single line for the convective evaporation region as previously observed for the R22/R114 mixture [3]. Chen's F factor underpredicts the present data for pure components by 20–30% with the larger deviation observed at higher qualities. Its slope, however, runs parallel to the present data.

The present data for pure R12 and R152a, the data for R22 and R114 [3], the data for steam–water by Kenning and Hewitt [18], and Aounallah *et al.* [19] under the suppression of nucleate boiling agree very well up to 65% quality which is the highest quality obtained by Kenning and Hewitt [18]. This may imply that a new F factor may be proposed and applied for the heat transfer prediction under the suppression of nucleate boiling for other fluids since water and refrigerants, which are very different fluids, have the same characteristics in this region.

Bennett and Chen [20] observed that Chen's F factor underpredicted their data obtained with ethylene glycol the Prandtl number of which is 6. They made a so-called 'Prandtl number correction' to account for a variation in fluid properties. The proposed correlation for the dimensionless heat transfer coefficient under the suppression of nucleate boiling is

$$\frac{h_{tp}}{h_{lo}} = F\,Pr^{0.296} \tag{6}$$

where F was originally given by Chen [4].

The dimensionless heat transfer coefficients in the purely convective evaporation region for R22, R114 (Fig. 8 in ref. [3]), R12, and R152a (Pr ranging from 2.6 to 5.4) in Fig. 6 lie on a single line and thus do not support the 'Prandtl number correction'. Furthermore, Kenning and Hewitt's [18] and Aounallah *et al.*'s [19] water data ($Pr = 1.4$) also lie very close to the present data. This fact strongly indicates that the dimensionless heat transfer coefficient, h_{tp}/h_{lo}, in the evaporative region should be independent of Pr.

There is other evidence supporting this view. Bennett and Chen's data were obtained from a very short test section and a thermal entry length effect was introduced as discussed in ref. [18]. Thus, the Prandtl number correction may have been used to account for the thermal entry length effect. An analytical study by Kunz and Yerazunis [21] showed that under evaporative conditions, the two-phase heat transfer coefficient was independent of Pr. They varied Pr

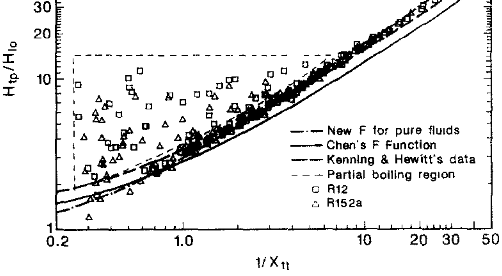

FIG. 6. Two-phase heat transfer multiplier as a function of $1/X_{tt}$ for pure R12 and R152a at $p_r = 0.08$.

from 10^{-3} to 10^4 and observed that Chen's F factor was always up to 20% below their prediction regardless of Pr.

Based on experimental evidence and the observations of other investigators, a revised F factor was determined by a regression analysis of the data with pure fluids in the evaporative region. The resulting F factor had a correlating coefficient of 0.999

$$F_p = 2.37\left(0.29+\frac{1}{X_{tt}}\right)^{0.85}. \qquad (7)$$

This newly determined F factor shown in Fig. 6 predicts the data for pure fluids in the convective evaporation region with a mean deviation of ±5%.

Figure 7 shows that for the R12/R152a mixture, nucleate boiling is not well suppressed at low qualities and the 'partial boiling' region is extended to high qualities, which is similar to the trend seen with pure fluids. However, the test results for the R22/R114 mixture [3] revealed that the scatter of data due to nucleate boiling is limited to qualities below 15%. Beyond this quality all data lie on a single line. Ross *et al.* [5] also observed a similar behavior with the R13B1/R152a mixture.

The discrepancy may be explained by considering the bubble growth dynamics. Due to a preferential stripping of a more volatile component near the bubble interface, the local equilibrium temperature increases. Consequently, the effective wall superheat decreases, which is termed loss of wall superheat. With a smaller effective wall superheat, nucleate boiling would be suppressed more rapidly with mixtures than with pure fluids under the same flow condition.

The amount of wall superheat reduced with mixtures may be approximated by an easily measurable quantity such as the composition difference in two phases, $(Y-X)$. Since the R22/R114 and R13B1/R152a mixtures have values of $(Y-X)$ four times larger than those of the R12/R152a mixture, it is reasonable that nucleate boiling would be suppressed at lower qualities for the R22/R114 and R13B1/R152a mixtures than the R12/R152a mixture.

Our previous study [3] also showed that the ratio, h_{tp}/h_{lo}, of the R22/R114 mixture in the evaporative region fell approximately 10% below that of the pure fluids. For the R12/R152a mixtures, however, the ratio lies very close to that of the pure components as shown in Figs. 6 and 7. The variation in the physical properties is already considered by using the dimensionless parameters. Hence, both present and previous results indicate that mass transfer resistance, an unaccounted factor, is small for flow boiling heat transfer with mixtures under the suppression of nucleate boiling. This conclusion supports the findings of Berntsson *et al.* [22] and Shock [23].

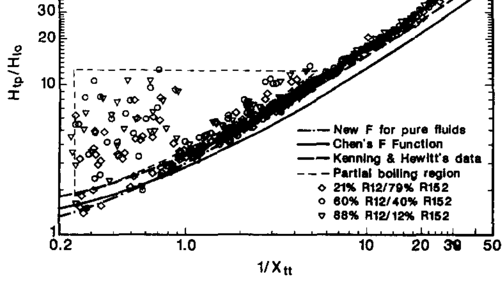

FIG. 7. Two-phase heat transfer multiplier as a function of $1/X_{tt}$ for a mixture of R12 and R152a at $p_r = 0.08$.

In order to take this mixture effect into account in the convective evaporation region even if it is small, the F factor for mixtures, F_m, is obtained by using only the phase equilibrium data, $(Y-X)$

$$F_m = F_p C_{me} \qquad (8)$$

where $C_{me} = 1-0.35|Y-X|^{1.56}$, $0.9 < C_{me} \leqslant 1$, is a correction factor which considers mass transfer resistance in the convective evaporation region.

4. PREDICTION OF TRANSITION QUALITY

There have been few studies on the suppression of nucleate boiling in the literature and it is not yet fully understood. Since the study of suppression of nucleate boiling requires theory of onset of nucleate boiling, a description of the bubble growth and a wall superheat equation to sustain nucleate boiling on a heating surface will be briefly given.

4.1. *Onset of nucleate boiling*

The equation describing the superheat required for a stable bubble is obtained by combining the pressure difference equation across a curved surface, the Clausius–Clapeyron equation, and the ideal gas law. The derivation of the bubble equilibrium equation, equation (9), is available elsewhere [24]

$$T_v - T_{sat} = \Delta T_{sat} = \frac{2\sigma R T_{sat}^2}{M h_{fg} p_l r^*} = \frac{2\sigma T_{sat}}{h_{fg}\rho_v r^*}. \qquad (9)$$

For bubble nucleation from a conical site on a heated surface, Hsu [25] postulated that the temperature of the liquid surrounding the top of the bubble should exceed that necessary for the nucleus to remain in equilibrium. A linear temperature profile was assumed in the laminar sublayer based on the hypothesis that the bubble nuclei develop within the laminar sublayer

$$T_{sb} = T_w - qy/k_l. \qquad (10)$$

To further simplify the analysis, the present study adopts the assumption, made by Collier [24], that a hemispherical bubble is formed on the cavity and no distortion of the isotherms occurs near the bubble cap in the sublayer.

One way of illustrating Hsu's criterion is shown in Fig. 8. If a wide range of cavity sizes is available on

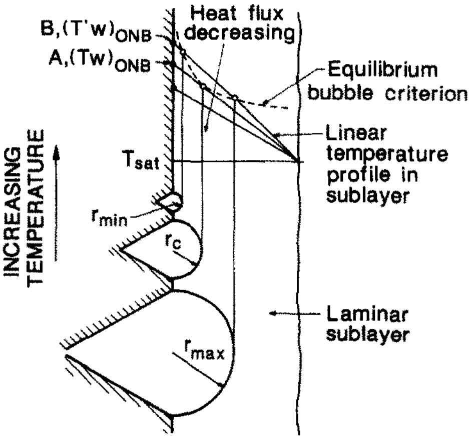

FIG. 8. Bubble nucleation from a conical cavity (taken from ref. [25]).

the surface, the first cavities generating bubbles would be those corresponding to the point of tangency of equations (9) and (10), r_c in Fig. 8. When equation (9) is just tangent to equation (10), then nuclei of a critical radius, r_c, would grow, which is expressed as

$$\frac{dT_{sb}}{dy} = \frac{dT_v}{dy}. \tag{11}$$

Davis and Anderson [26] carried out an analytical solution of equation (11) and obtained a critical radius, r_c, and the superheat required to initiate nucleate boiling for two cases.

Case (1). The heating surface has a sufficiently wide range of active cavity sizes and the corresponding wall superheat is

$$(T_w - T_{sat})_{ONB} = \left(\frac{8\sigma q_{ONB} T_{sat}}{h_{fg} k_l \rho_v}\right)^{0.5}. \tag{12}$$

Case (2). Available cavity size on the surface is limited and hence the maximum cavity size available, r_{max}, on the surface is assumed and substituted for r_c in equation (13)

$$(T_w - T_{sat})_{ONB} = \frac{2\sigma T_{sat} v_{fg}}{h_{fg} r_c} + \frac{q_{ONB} r_c}{k_l}. \tag{13}$$

At first, Davis and Anderson applied case (1) and discovered that the wall superheat calculated by equation (12) did not agree with the measured value. After examining the heating surface under a microscope, they discovered that the available cavity size on the surface was limited to 1 μm, they obtained a good agreement by applying case (2) with an assumed maximum radius of $r_{max} = 1$ μm.

4.2. *Suppression of nucleate boiling in annular flow*

Suppression of nucleate boiling in annular flow bears similar characteristics to those of the onset of nucleate boiling except that it occurs in the opposite direction with decreasing wall superheat. As the flow pattern becomes annular, the main heat transfer mechanism becomes heat conduction across the liquid layer. As quality is increased further, the liquid layer becomes thinner and hence the heat transfer coefficient increases due to less thermal resistance. Thus, the wall superheat available in the liquid layer for a given heat flux decreases with increasing quality or mass flux. When the wall superheat available falls below that required to form bubbles, nucleate boiling is fully suppressed.

For the prediction of transition quality, the two cases discussed above were considered. For case (1) the suppression wall superheats, ΔT_{SNB}, are calculated at a reduced pressure of 0.08 for a heat flux of 26 kW m^{-2}; they are 1.4, 2.1, 1.75, and 1.35°C for R22, R114, R12, and R152a, respectively. These values are much lower than the measured wall superheats at the point of suppression of nucleate boiling, which are 7.65, 10.5, 8.9, and 7.9°C, respectively. Consequently, this result implies that there is not a sufficiently wide range of active cavity sizes available on the present heating surface, which corresponds to the findings of Davis and Anderson [26]. Thus, case (2) is applicable for the prediction of the transition quality with an assumed maximum cavity size on the surface.

Recently, Hino and Ueda [27] performed experiments with R113 flowing inside a stainless steel tube of the same kind as the present test section. They measured the wall superheat corresponding to the inception of nucleate boiling. The wall superheats obtained from their experiments are much greater than those predicted by equation (12). Thus, they used equation (13) with an assumed maximum cavity radius, r_{max}, ranging from 0.22 to 0.34 μm, and obtained a good agreement.

The procedure for the prediction of transition quality by applying case (2) is as follows: the wall superheat in the absence of nucleate boiling is determined from the newly determined F factor given in equation (7). Combining equations (7) and (13) yields the suppression wall superheat as a function of quality (note that the F factor contains a quality)

$$\Delta T_{SNB} = \frac{2\sigma T_{sat} k_l}{h_{fg} \rho_v r_{max}(k_l - F h_{lo} r_{max})}. \tag{14}$$

The suppression heat flux, q_{SNB}, is obtained by combining equations (7) and (14)

$$q_{SNB} = F_p h_{lo} \Delta T_{SNB}. \tag{15}$$

Finally, the transition quality is the quality at which the applied heat flux is equal to the suppression heat flux under a given flow condition.

Figure 9 illustrates the comparison between the predicted transition qualities and the experimentally observed values for pure fluids at different mass flow rates. A value of $r_{max} = 0.28$ μm was found to predict the transition quality very well and it also falls in the range found by Hino and Ueda [27].

The method for the prediction of transition quality is now extended to mixtures. For mixtures, the wall

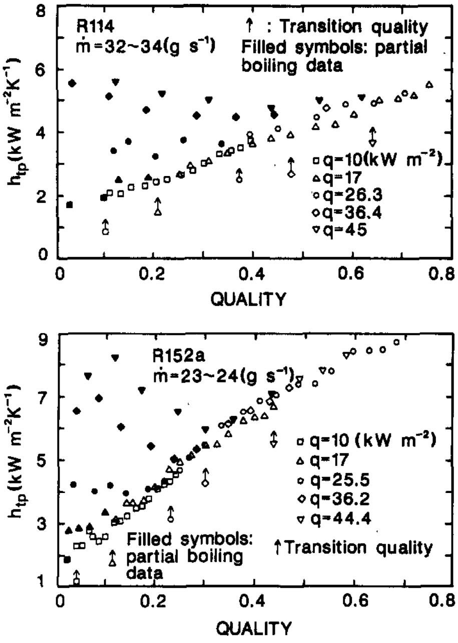

FIG. 9. Comparison between the predicted and experimentally observed transition quality for pure refrigerants.

superheat equation has to be modified due to the phenomenon of loss of available superheat. Most previous investigators used equation (1) to consider the mixture effect. The reciprocal of the denominator in equation (1) is an analytically developed correction factor to consider the mixture effects. In terms of the temperature difference, equation (1) can be rewritten as

$$\frac{\Delta T_m}{\Delta T_i} = 1-(C_{pl}/h_{fg})(\alpha/D)^{0.5}(Y-X)(\mathrm{d}t/\mathrm{d}X)_{bub} \quad (16)$$

where

$$\Delta T_i = X_1 \Delta T_1 + X_2 \Delta T_2.$$

Equation (16) states that for mixtures, the resulting temperature difference (not a temperature driving potential) for a given heat flux is greater than the ideal value since the right-hand side of equation (16) is greater than 1. Consequently, the heat transfer coefficient of mixtures is lower than the ideal values.

There have been a few empirical correlations to approximate equation (16). They, however, have such limitations as requiring either a value of mass diffusivity of liquid, D, which is rarely found, or empirical constants which depend upon mixtures [28]. In order to avoid these difficulties, Thome [28] has proposed a method of using only phase equilibrium data to consider the mixture effects. Following the same line of reasoning as Thome [28], Ünal [29] developed a correlation of equation (16) for mixtures

$$\frac{\Delta T_m}{\Delta T_i} = C_{UN} = [1+(b_2+b_3)(1+b_4)](1+b_5) \quad (17)$$

where

$$b_2 = (1-X)\ln\left(\frac{1.01-X}{1.01-Y}\right)+X\ln\left(\frac{X}{Y}\right) + |Y-X|^{1.5}$$

$$b_3 = 0 \quad \text{for } X \geq 0.01$$

$$b_3 = (Y/X)^{0.1}-1 \quad \text{for } X < 0.01$$

$$b_4 = 152(p/p_{cmvc})^{3.9}$$

$$b_5 = 0.92|Y-X|^{0.001}(p/p_{cmvc})^{0.66}$$

$$X/Y = 1 \quad \text{for } X = Y = 0$$

and p_{cmvc} is the critical pressure of the more volatile component.

Figure 10 illustrates Ünal's constants which take into account the mixture effects, equation (17) for the R22/R114 and R12/R152a mixtures. The R22/R114 and R12/R152a mixtures have up to 80 and 25% larger values of ΔT_m than the ideal values, ΔT_i. This implies that the reduction in a wall superheat would be as much as 44 and 28% for the R22/R114 and R12/R152a mixtures, respectively.

The same approach for the prediction of transition quality for pure fluids is applied for mixtures. Equation (14) is used with the properties calculated using the mixing rules and the suppression heat flux is multiplied by the correction factor, equation (17), as done by Ross *et al.* [5]. By doing this, the predicted suppression heat flux is forced to be significantly higher. Thus, it takes into account that with mixtures nucleate boiling is suppressed at lower qualities. Figure 11 shows the results for mixtures. A fairly good agreement is again obtained with mixtures.

5. CORRELATION

5.1. *Correlation for pure fluids*

Figure 12 illustrates a typical trend of the two-phase heat transfer coefficients for pure fluids for a given mass flow rate at various heat fluxes. Figure 12 was actually obtained by approximating the measured data with R12. In the partial boiling regime (point 1),

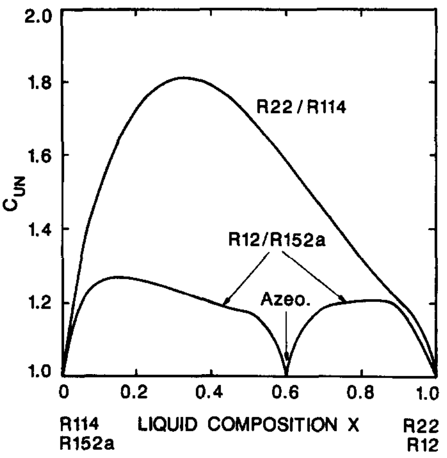

FIG. 10. Ünal's constants to consider mixture effects in nucleate pool boiling.

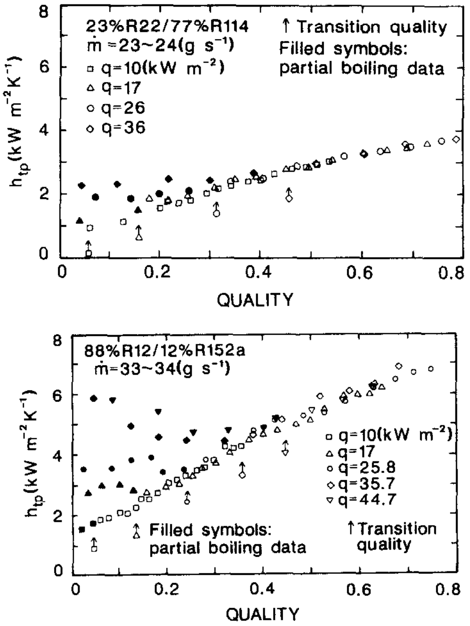

FIG. 11. Comparison between the predicted and experimentally observed transition quality for mixed refrigerants.

h_{nbc} is obtained by subtracting h_{cec} from h_{tp} and h_{cec} is determined using equation (7)

$$h_{\mathrm{nbc}} = h_{\mathrm{tp}} - h_{\mathrm{cec}} \tag{18}$$

where

$$h_{\mathrm{cec}} = F_{\mathrm{p}} h_{\mathrm{lo}}.$$

As quality increases further, h_{nbc} decreases and above the transition quality (point 2), h_{cec} becomes equal to h_{tp}.

Since the present results support Chen's supposition that the two-phase heat transfer coefficient is additive of the two contributions, the original form of equation (3) by Chen [4] is retained with some modifications. First of all, Forster and Zuber's nucleate pool boiling heat transfer correlation [17] is replaced by that of Stephan and Abdelsalam [30]

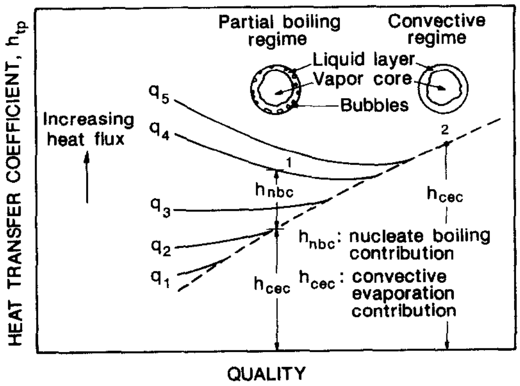

FIG. 12. Nucleate boiling and convective evaporation contributions in flow boiling heat transfer.

$$h_{\mathrm{SA}} = 207 \frac{k_{\mathrm{l}}}{bd} \left(\frac{q\,bd}{k_{\mathrm{l}} T_{\mathrm{sat}}} \right)^{0.745} \left(\frac{\rho_{\mathrm{v}}}{\rho_{\mathrm{l}}} \right)^{0.581} Pr_{\mathrm{l}}^{0.533} \tag{19}$$

where $bd = 0.0146\beta[2\sigma/(g(\rho_{\mathrm{l}} - \rho_{\mathrm{v}}))]^{0.5}$ with a contact angle $\beta = 35^{\circ}$.

The second modification is in h_{nbc}. For a given quality in the partial boiling regime, h_{nbc} is a strong function of heat flux as shown in Figs. 3, 9, 10, and 12. It is also a function of mass flow rate for a given heat flux. Consequently, h_{nbc} becomes a function of quality, heat flux, and mass flow rate. The nucleate boiling contribution is usually normalized by a nucleate pool boiling heat transfer coefficient, h_{nb}. Then, the factor due to a nucleate boiling effect, N, obtained by dividing h_{nbc} by h_{nb}, represents how strong the effect of nucleate boiling is in flow boiling as compared to that in nucleate pool boiling. This factor N becomes a function of quality, heat flux, and mass flow rate

$$N = h_{\mathrm{nbc}}/h_{\mathrm{SA}} = fn(x, q, \dot{m}). \tag{20}$$

The functional dependence of N on x, q, and $\dot{m}$ can be best represented by employing dimensionless parameters such as the Martinelli parameter and boiling number, X_{tt} and Bo

$$N = fn(X_{tt}, Bo) \tag{21}$$

where N in the present study differs from other similar factors such as the other suppression factors, S, by Chen [4] and by Gungor and Winterton [31] in that N is a strong function of heat flux while the other S factors are independent of heat flux. The present results, however, clearly reveal a strong dependence of N on heat flux.

A regression analysis was carried out to obtain the functional dependence of the factor N on X_{tt} and Bo. First, an attempt was made by using only X_{tt} and mass flux G, which are the only variables in the S factors by Chen, and Gungor and Winterton. The resulting correlation coefficient was 0.3–0.5, indicating a poor fit. When the heat flux dependence of N was included by including Bo, the resulting correlation coefficient using all the pure fluids (R22, R114, R12, R152a) and azeotropic (R500) data in the partial boiling regime became 0.97. This indicates an excellent fit considering the accuracy of Stephan and Abdelsalam's nucleate pool boiling correlation (±10% for refrigerants). The resulting N factor is

$$N = 4048 X_{tt}^{1.22} Bo^{1.13} \quad \text{for } X_{tt} < 1. \tag{22}$$

The final correlation for pure refrigerants and azeotrope becomes

$$h_{\mathrm{tp}} = h_{\mathrm{nbc}} + h_{\mathrm{cec}} = N h_{\mathrm{SA}} + F_{\mathrm{p}} h_{\mathrm{lo}} \tag{23}$$

where N, h_{SA}, F_{p}, and h_{lo} are defined as in equations (22), (19), (7), and (2) with $G(1-x)$ replacing G.

For mixtures, Ünal's correlation [29] for the nucleate pool boiling heat transfer coefficient is again utilized. For a given pressure, the nucleate pool boiling heat transfer coefficients of pure components 1

Table 2. Comparison of percentage deviation between the various correlations and the present data for pure fluids

Fluid	Number of data points	Chen Mean	Chen Ave.	Bennett and Chen Mean	Bennett and Chen Ave.	Shah Mean	Shah Ave.	Gungor and Winterton Mean	Gungor and Winterton Ave.	Ross Mean	Ross Ave.	Present work Mean	Present work Ave.
R22	378	16.7	14.3	35.6	34.4	13.0	−11.0	15.0	3.62	11.8	3.1	5.77	−0.7
R114	198	12.8	6.16	42.5	40.2	11.3	−10.3	13.6	4.6	22.1	19.6	9.3	3.0
R12	286	15.3	9.0	36.0	32.8	12.9	−10.5	17.3	4.35	15.2	6.6	7.6	0.5
R152a	376	17.4	14.4	38.6	37.0	12.3	−10.3	21.4	10.8	15.3	5.7	7.4	−3.3
R500	350	14.8	8.0	35.8	32.7	13.9	−12.7	18.3	3.9	15.5	6.7	7.82	−1.7
All	1588	15.7	7.8	37.3	35.0	12.8	−11.0	17.5	5.64	15.3	7.2	7.2	−0.89

$$\text{Mean dev.} = \frac{1}{n}\sum_{1}^{n} ABS\left[\frac{(h_{\text{cal}} - h_{\text{exp}}) \times 100}{h_{\text{exp}}}\right].$$

$$\text{Ave. dev.} = \frac{1}{n}\sum_{1}^{n}\left[\frac{(h_{\text{cal}} - h_{\text{exp}}) \times 100}{h_{\text{exp}}}\right].$$

and 2, h_1 and h_2, are calculated by Stephan and Abdelsalam's correlation [30]. The nucleate pool boiling heat transfer coefficient of a mixture, h_{UN}, is then determined by

$$\frac{h_{\text{UN}}}{h_{\text{i}}} = \frac{1}{C_{\text{UN}}} \tag{24}$$

where C_{UN} is given in equation (17) and the ideal heat transfer coefficient, h_{i}, is

$$h_{\text{i}} = \frac{1}{\dfrac{X_1}{h_1} + \dfrac{X_2}{h_2}} = \frac{h_1 h_2}{h_2 X_1 + h_1 X_2}. \tag{25}$$

Equation (24) indicates that the heat transfer coefficient of a mixture is always less than the ideal value since C_{UN} is always greater than 1 for mixtures as shown in Fig. 10.

As shown earlier, nucleate boiling is suppressed at lower qualities for mixtures than for pure components under a given condition. Thus, the easier suppression of nucleate boiling for mixtures is taken into account by dividing the factor due to nucleate boiling for pure fluids, N, by C_{UN}. The final two-phase heat transfer correlation for mixtures becomes

$$h_{\text{tp}} = \frac{N}{C_{\text{UN}}} h_{\text{UN}} + C_{\text{me}} F_{\text{p}} h_{\text{lo}} \tag{26}$$

where C_{me} is defined in equation (8).

The distinctive feature of equation (26) for mixtures is that it becomes equation (23) for pure fluids and azeotropes since C_{UN} and C_{me} equal 1 for those fluids.

Table 2 gives a comparison between the various correlations and the present results for pure fluids and azeotropes. The correlations by Chen [4], Shah [32], and Gungor and Winterton [31] have mean deviations of 13–18% against the present data. This indirectly ensures the reliability of the present data for pure fluids.

As shown in Fig. 6, the convective contribution in Chen's correlation underpredicts the present results up to 20–30%. The whole correlation, equation (3), however, always overpredicts the present data up to 15% (average deviation of Chen's correlation), which indicates that the nucleate boiling contribution in Chen's equation is overpredicted by up to 40%. Toral *et al.* [33], however, observed that the nucleate boiling contribution in Chen's correlation underestimated their results as much as 60%. These results strongly suggest that Forster and Zuber's nucleate pool boiling equation [17] in Chen's correlation is not universal for all fluids and special care has to be taken in its use.

Another point to be noted is that Bennett and Chen's correlation, which is exactly the same as Chen's except for the Prandtl number correction, overpredicts the present results by up to 42% indicating that the Prandtl number correction, equation (6), does not improve the fit but rather overpredicts the convective contribution.

Even though Shah's correlation [32] has a low mean deviation of 12.8%, his correlation always underpredicts the present results (average deviation of −10%). This comes from the fact that the convective contribution in Shah's correlation is underestimated and this is chosen to be the total heat transfer coefficient instead of adding the two contributions as in Chen's. This implies that the criterion in Shah's correlation of choosing one of the two contributions to be the total heat transfer coefficient is not valid in the partial boiling regime.

The present correlation shows a good agreement with the present data with a mean deviation of 7.2% for all pure fluids. The success of the present correlation stems from the excellent fit of Stephan and Abdelsalam's nucleate pool boiling heat transfer correlation for pure refrigerants and the proper heat flux dependence of N in suppressing the nucleate boiling contribution. Finally, the success of the correlation supports the supposition of Chen [4] that the two contributions are additive.

Table 3 lists the comparisons for mixtures. All correlations developed for pure fluids overpredict the

Table 3. Comparison of percentage deviation between the various correlations and the present data for mixtures

Fluid	Number of data points	Chen		Shah		Gungor and Winterton		Ross		Present work	
		Mean	Ave.	Mean	Ave.	Mean	Ave.	Mean	Ave.	Mean	Ave.
23% R22/77% R114	233	44.5	44.5	15.8	4.42	41.8	40.7	13.8	−9.1	6.8	1.9
48% R22/52% R114	216	62.3	62.3	19.8	12.6	60.4	60.4	13.0	−6.75	8.9	3.64
77% R22/23% R114	207	48.7	48.7	17.4	7.3	45.0	44.7	12.9	−10.2	9.6	7.2
21% R12/79% R152a	313	20.5	19.0	14.8	−8.3	25.0	14.8	25.4	−25.4	9.0	−5.96
88% R12/12% R152a	292	20.5	16.8	16.8	−9.16	21.3	11.8	28.2	−23.0	13.1	−0.03
All	1261	36.7	35.5	16.7	0.0	36.6	31.6	19.7	−16.1	9.6	0.68

present data for mixtures up to 60%. This is due to the fact that these correlations do not provide the right suppression factor to take into account that nucleate boiling effects are more easily suppressed for mixtures than for pure fluids. This indicates that the ideal mixing rule, merely based on the overall composition of the mixture, cannot predict the right heat transfer coefficient for the mixture.

The correlations of Chen, and Gungor and Winterton, which are additive, yielded a large deviation of 36% while those of Shah and Ross, which select the larger of the two heat transfer contributions, had a lower deviation of 16–19%. Since for mixtures nucleation is suppressed at lower qualities than for pure fluids, the nucleate boiling contribution is very small for mixtures. That is why only the convective contributions in Ross' and Shah's correlations, which were chosen as the total heat transfer, yielded smaller deviations.

The present correlation using only phase equilibrium data has a mean deviation of 9.6% which is within the range of experimental error. Considering the complicated boiling phenomenon with mixtures and the accuracies of Ünal's method and Stephan and Abdelsalam's correlation, this correlation is remarkable. This result indicates that the methods to account for the mixture effects in nucleate pool boiling are also equally applicable with the same accuracy in convective boiling.

Finally, Fig. 13 shows the predicted h_{tp} for the R22/R114 mixture at 350 kPa for 26 kW m^{-2} and 23 g s^{-1}. At low qualities h_{tp} decreases with increasing quality for both pure and mixed refrigerants, as observed in the present study. As composition increases toward the more volatile component, h_{tp} degrades substantially at low qualities because in the partial boiling regime the nucleate boiling effect is quite strong for pure fluids while it is very weak for mixtures. Thus, in the partial boiling regime, similar mixture effects found in nucleate pool boiling heat transfer exist in flow boiling heat transfer with mixtures and are responsible for the severe heat transfer degradation seen with mixtures in this regime.

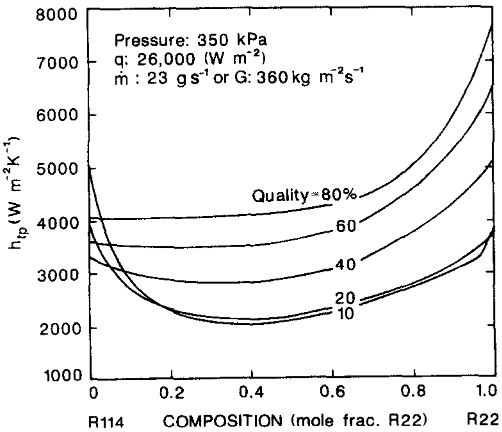

FIG. 13. Predicted two-phase heat transfer coefficients for the R22/R114 mixture at 350 kPa for 26 kW m^{-2}.

At high qualities, however, nucleate boiling is suppressed even for pure fluids due to enhanced heat transfer and hence the primary heat transfer mechanism is evaporation at the interface for both pure and mixed refrigerants. In this regime, the phenomenon of loss of wall superheat is not present since bubbles are not formed and the physical property variation due to mixing is responsible for almost all of the heat transfer degradation seen with mixtures.

6. CONCLUSIONS AND RECOMMENDATIONS

Based on the study of the mixture effects on horizontal flow boiling heat transfer with both non-azeotropic and azeotropic mixtures, R22/R114 and R12/R152a, the following conclusions and recommendations can be drawn.

(1) A full suppression of nucleate boiling was observed for both pure and mixed refrigerants. Two distinct heat transfer regions exist in the annular flow regime. In the partial boiling region, a strong heat flux dependence of h_{tp} was observed while in the convective evaporation region h_{tp} for various heat fluxes merged into a single line depending only upon flow parameters such as quality.

(2) In the convective evaporation region, the variation of physical properties due to mixing was responsible for almost all of the heat transfer degradation from the ideal values; the small remainder was presumably due to mass transfer resistance.

(3) In the partial boiling region at low qualities, the nucleate boiling effect is quite strong for pure fluids while it is suppressed for mixtures due to loss of wall superheat and mass transfer resistance. In this

region, the heat transfer coefficients of mixtures are severely degraded and the treatments to account for the mixture effects in nucleate pool boiling heat transfer seemed to be valid also in flow boiling heat transfer.

(4) While a circumferential wall temperature variation in annular flow was observed for the R22/R114 mixture, it did not occur for the R12/R152a mixture. For the R12/R152a mixture, the local compositions around the circumference were measured to be identical, indicating that the mixtures with a small volatility difference would not have such a unique behavior of circumferential wall temperature variation even if they are mixtures.

(5) A method for the prediction of transition quality from a partial boiling to a convective evaporation region was developed. When it is extended to mixtures, the method requires additionally only phase equilibrium information. The prediction yielded good agreement with the present data for pure and mixed refrigerants when an $r_{\max}$ of 0.28 μm was assumed. The sensitivity of $r_{\max}$ on the suppression heat flux was not studied and may be of interest for further studies.

(6) Data for pure fluids were compared against several well-known correlations; a mean deviation of up to 40% was observed. The proposed correlation for pure fluids, based on the additive supposition of Chen and incorporating a heat flux dependent suppression factor, yielded a mean deviation of 7.2% for pure fluids and azeotrope data.

(7) For mixtures, all well-known correlations overpredicted the present data. The newly developed mixture correlation was based on Ünal's method using only phase equilibrium data in conjunction with Stephan and Abdelsalam's nucleate boiling correlation. It predicted the data with a mean deviation of 9.6%.

Since the major part of the evaporator coils would be in the convective evaporation region as this study reveals, experiments with a forced mixing device such as a swirl generator inside the coils are recommended since a considerable heat transfer enhancement is reported by employing this device, which would overcome the drawback of using mixtures. A visualization study is also recommended to see if bubbles exist on the surface in the convective region. This would help understand the fundamentals of flow boiling heat transfer and lead to a proper prediction method for transition quality.

Experiments with mixtures having a larger volatility difference than that of R22/R114 such as R22/R11 are needed to validate the present methods to account for the mixture effects. Experiments with a mixture having $(Y-X) = 0.15$–0.25 are proposed to determine the limit of $(Y-X)$ below which mass transfer resistance becomes negligible in the convective evaporation region.

Finally, evaporators in most refrigeration equipment are exposed to a constant temperature boundary condition rather than the constant heat flux condition employed in the present study. Since the gliding temperature effect with mixtures can be best exploited in countercurrent heat exchangers, experiments with a tube to tube countercurrent evaporator are recommended to simulate the constant boundary condition. This would also enable the measurements at high qualities in which the present test rig cannot be operated all the time due to the risk of burn out.

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ETUDE DU TRANSFERT THERMIQUE LORS DE L'EBULLITION EN ECOULEMENT DE MELANGES REFRIGERANTS

Résumé—On étudie des effets de mélange sur le transfert thermique lors de l'ébullition pour un écoulement horizontal de mélanges réfrigérants azéotropiques ou non. Plus de 2000 coefficients locaux de transfert de chaleur sont obtenus avec le mélange azéotropique R12/R152a et ils sont comparés avec les données expérimentales antérieures pour le mélange non azéotropique R22/R114. Dans une région d'évaporation convective, on trouve une petite résistance au transfert de masse pour les mélanges. La variation des propriétés physiques dues au mélange est responsable de la plus grande part de la dégradation du transfert de chaleur. Néanmoins, dans une région d'ébullition partielle, on constate une forte dégradation du transfert de chaleur avec des mélanges, semblable à celle obtenue dans le cas de l'ébullition nucléée en réservoir pour les mélanges. Une suppression de l'ébullition nucléée aux faibles qualités, due à la baisse de la surchauffe de la paroi avec les mélanges, est responsable de cela. On développe une analyse pour prédire une qualité de transition en utilisant la théorie de Hsu sur l'apparition de l'ébullition nucléée. La prédiction s'accorde bien avec les qualités de transition observées à la fois pour les réfrigérants purs et en mélange. Des formules, basées sur l'hypothèse de Chen et utilisant des données sur l'équilibre d'une phase pour considérer les effets de mélange, sont développées avec des écarts moyens de 7,2 et 9,6% pour les réfrigérants purs et en mélange.

EINE UNTERSUCHUNG ZUM WÄRMEÜBERGANG BEIM STRÖMUNGSSIEDEN VON KÄLTEMITTEL-GEMISCHEN

Zusammenfassung—Es werden Mischungseinflüsse auf den Wärmeübergang beim Strömungssieden von azeotropen und nicht-azeotropen Kältemittel-Gemischen im horizontalen Rohr untersucht. Es wurden mehr als 2000 lokale Wärmeübergangskoeffizienten mit dem azeotropen Gemisch R12/R152a erhalten und mit den früher gemessenen Werten des nicht azeotropen Gemisches R22/R114 verglichen. Im konvektiven Strömungssiedebereich wird für Gemische ein geringer Widerstand im Massentransport festgestellt. Die Variation der physikalischen Eigenschaften aufgrund der Mischung ist für fast alle Verminderungen des Wärmeübergangs verantwortlich. Im partiellen Siedebereich wird jedoch ähnlich wie beim Behältersieden von Gemischen eine starke Verminderung des Wärmeübergangs festgestellt. Die Unterdrückung des Blasensiedens bei geringem Dampfgehalt aufgrund der Abnahme der Wandüberhitzung ist dafür verantwortlich.

ИССЛЕДОВАНИЕ ТЕПЛООБМЕНА С ОХЛАЖДАЮЩИМИ СМЕСЯМИ ПРИ КИПЕНИИ В ПОТОКЕ

Аннотация—Исследуется влияние азеотропных и неазеотропных смесей хладагентов на горизонтальный теплоперенос при кипении в потоке. Получено более 2000 значений локального коэффициента теплопереноса для азеотропной смеси R12/R152a и проведено их сравнение с ранее полученными данными для неазеотропной смеси R22/R114. В области конвективного испарения обнаружено небольшое сопротивление массопереносу смесей. Ухудшение теплопереноса почти полностью объясняется изменением физических свойств при смешении. Однако, в области частичного кипения найдено сильное ухудшение теплообмена со смесями, аналогичное теплообмену при пузырьковом кипении в большом объеме. Это ухудшение вызвано подавлением пузырькового кипения при более низком весовом паросодержании, обусловленном снижением перегрева стенки смесями. С использованием теории Сюй о возникновении пузырькового кипения разработан метод расчета переходного весового паросодержания. Рассчитанные значения хорошо согласуются с экспериментальными данными по переходным паросодержаниям как для чистых хладагентов, так и для их смесей. Зависимости, выведенные на основе предположений Чена и с применением только данных по фазовому равновесию для описания эффектов смеси дают результаты со средним отклонением 7,2 и 9,6% для чистых и смешанных хладагентов.