Prediction of Forced Convective Boiling Heat Transfer Coefficient of Pure Refrigerants and Binary Refrigerant Mixtures Inside a Horizontal Tube

Eul Cheong Hong

School of Mechanical and Aerospace Engineering, Seoul National University, Seoul 151-742, Korea

Jee Young Shin

Department of Mechanical Engineering, Dong-Eui University, Busan 614-714, Korea

Min Soo Kim*, Kyungdoug Min, Sung Tack Ro

School of Mechanical and Aerospace Engineering, Seoul National University, Seoul 151-742, Korea

Forced convective boiling heat transfer coefficients were predicted for an annular flow inside a horizontal tube for pure refrigerants and nonazeotropic binary refrigerant mixtures. The heat transfer coefficients were calculated based on the turbulent temperature profile in liquid film and vapor core considering the composition difference in vapor and liquid phases, and the nonlinearity in mixing rules for the calculation of mixture properties. The heat transfer coefficients of pure refrigerants were estimated within a standard deviation of 14% compared with available experimental data. For nonazeotropic binary refrigerant mixtures, prediction of the heat transfer coefficients was made with a standard deviation of 18%. The heat transfer coefficients of refrigerant mixtures were lower than linearly interpolated values calculated from the heat transfer coefficients of pure refrigerants. This degradation was represented by several factors such as the difference between the liquid and the overall compositions, the conductivity ratio and the viscosity ratio of both components in refrigerant mixtures. The temperature change due to the concentration gradient was a major factor for the heat transfer degradation and the mass flux itself at the interface had a minor effect.

Key Words : Evaporative Heat Transfer Coefficient, Horizontal Annular Flow, Nonazeotropic Refrigerant Mixture, Heat Transfer Degradation, Mass Transfer

Nomenclature

b_L^+ : Dimensionless thickness of viscous sublayer

$\bar{c}$: Molar density

c : Specific heat

D : Inner diameter of tube

D_{eff} : Mass diffusion coefficient

G : Mass flux

h : Heat transfer coefficient

* Corresponding Author,
E-mail : minskim@snu.ac.kr
TEL : +82-2-880-8362; **FAX** : +82-2-883-0179
School of Mechanical and Aerospace Engineering,
Seoul National University, Seoul 151-742, Korea. (Manuscript **Received** November 21, 2002; **Revised** February 14, 2003)

h_D : Mass transfer coefficient

i : Enthalpy

j : Molar flux

k : Thermal conductivity

L : Length

M : Molecular mass

m : Liquid film thickness

m^+ : Dimensionless liquid film thickness
$(m^+ = m u_\tau/\nu, \ u_\tau = \sqrt{\tau_w/\rho})$

m_i'' : Interfacial mass flux

Pr : Prandtl number

q'' : Heat flux

R : Inner radius of tube

R^+ : Dimensionless radius $(R^+ = R u_\tau/\nu)$

r : Radial distance from the axis of tube

r^+ : dimensionless radial distance from the axis of tube ($r^+ = ru_\tau/\nu$)

t : Temperature

t^+ : Dimensionless temperature

u : Velocity

u^+ : Dimensionless velocity ($u^+ = u/u_\tau$)

w : Mass fraction

x : Quality

y : Distance from inner tube wall ($y = R - r$)

y^+ : Dimensionless distance from inner tube wall ($y^+ = yu_\tau/\nu$)

y_G : Mole fraction in vapor phase

y_L : Mole fraction in liquid phase

y_o : Overall mole fraction

z : Axial distance

Greek symbols

α : Thermal diffusivity

δ_H : Equivalent laminar-film thickness for heat transfer

δ_M : Equivalent laminar-film thickness for mass transfer

ε_M : Eddy diffusivity for momentum transfer

ε_H : Eddy diffusivity for heat transfer

κ : Mixing length constant

μ : Dynamic viscosity

ν : Kinematic viscosity ($\nu = \mu/\rho$)

ρ : Density

τ : Shear stress

Subscripts

1 : More volatile component

2 : Less volatile component

c : Center of the vapor core

G : Vapor phase

i : Vapor-liquid interface

L : Liquid phase

mix : Mixture

s : Saturation

t : Total quantity in refrigerant mixture

W : Inner wall

1. Introduction

The phase-out of chlorofluorocarbon (CFC) and hydrochlorofluorocarbon (HCFC) refrigerants which have strong influence on global warming and ozone depletion drives a shift to use new alternative refrigerants in thermal systems including heat pumps and refrigerators. As alternatives, refrigerant mixtures are also suggested since they have a potential for a higher efficiency (Kruse, 1981 ; Domanski and Didion, 1993 ; Didion and Bivens, 1990). Since the first development of these refrigerants, the application to thermal systems has been an issue to achieve required capacity and higher efficiency. Major and minor modifications of refrigeration and heat pump systems have been made in the long history of technology development. However, the system with new alternative refrigerants is found to have some problems and is sometimes reported to have a lower efficiency when the refrigerant is used as a drop-in replacement in CFC or HCFC machines. The performance rating is not always good because the system is not optimized. A new system design should be assisted by precise estimation of thermodynamic, transport and other chemicophysical properties of new refrigerants. In order to redesign heat exchangers, heat transfer characteristics of new alternative refrigerants are also needed.

The main objective of this study is to present the analytical method to predict a horizontal flow boiling heat transfer coefficients of pure refrigerants and binary refrigerant mixtures. It has been reported by many researchers that the evaporative heat transfer coefficient of binary refrigerant mixtures are lower than linearly interpolated value based on the pure component data (Collier and Thome, 1994 ; Hihara et al., 1989 ; Jung et al., 1989 ; Hong et al., 1995). The heat transfer degradation of refrigerant mixtures has been analyzed with a simplified method in this paper.

2. Representation of Evaporative Heat Transfer Coefficient

In commercial heat pumps and refrigerators, subcooled liquid refrigerant is expanded through an expansion device. The refrigerant after an expansion process is normally in two phase. Evaporator inlet quality is about 0.1−0.2, above which convective boiling develops and nucleate

boiling is known to be suppressed.

In this work, a focus was put on the forced convective boiling heat transfer during the evaporation process because it is a dominant mechanism of heat transfer for conventional evaporators. Annular flow model is introduced in this work since it is a well-known flow pattern in the evaporator over the quality ranges from about 0.1 up to almost unity (Collier and Thome, 1994).

For pure refrigerants, most of the vapor and liquid phases during evaporation remain at the same saturation temperature if the pressure drop is neglected. The liquid close to the tube wall is superheated above the saturation temperature when the tube wall is heated. In the case of binary nonazeotropic refrigerant mixtures, it should be noted that the saturation temperature (or equilibrium temperature) changes during evaporation.

In numerous experimental researches, the evaporative heat transfer coefficient is defined as

$$h = \frac{q''_w}{t_w - t_s} \qquad (1)$$

where the saturation temperature, t_s, is calculated from the energy balance and the equation of state at the measured pressure. Inner wall temperature, t_w, is normally measured in the experiment. It is conjectured that the vapor-liquid interface temperature and the vapor temperature are same as the saturation temperature of the pure refrigerants (see Fig. 1).

For radial heat transfer across a turbulent liquid film at a given quality neglecting axial heat transfer, we can write

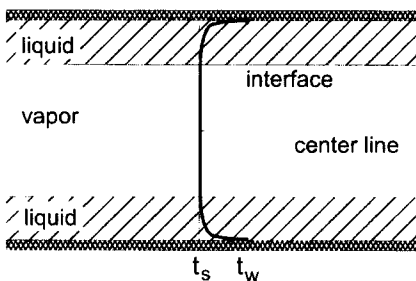$$\frac{R}{r} q''_w = q'' = \rho_L C_L (\alpha + \varepsilon_H) \frac{dt}{dr} \qquad (2)$$

Fig. 1 Schematic temperature profile during evaporation of pure refrigerant

where eddy diffusivity for heat transferis defined as $\varepsilon_H = \varepsilon_M / \mathrm{Pr}_t$. Pr_t is set to 0.9 throughout the considered region (Kays and Crawford, 1993). Eddy diffusivity for momentum is calculated according to the empirical equation proposed by Reichardt for the entire region outside the viscous sublayer (Kays and Crawford, 1993). Introducing a dimensionless temperature expressed by

$$t^+ = \frac{(t_w - t) \sqrt{(\tau_w/\rho_L)}}{q''_w / \rho_L C_L} \qquad (3)$$

and integrating Eq. (2), temperature distribution is derived for liquid layer in an annular flow inside a horizontal tube in Eq. (4). Non-dimensional distance of the viscous sublayer in the temperature field is selected as 13.2, and the von Karman mixing length constant, κ, is set as 0.4 for turbulent wall-bounded flow.

$$t^+ = 13.2 \, \mathrm{Pr}_L - \frac{\mathrm{Pr}_t}{\kappa} \left\{ \ln \left[\frac{(r_L^+/R_L^+)^6}{(1 - (r_L^+/R_L^+)^2)(0.5 + (r_L^+/R_L^+)^2)^2} \right] \right.$$
$$\left. - \ln \left[\frac{((R_L^+ - 13.2)/R_L^+)^6}{(1 - ((R_L^+ - 13.2)/R_L^+)^2)(0.5 + ((R_L^+ - 13.2)/R_L^+)^2)^2} \right] \right\} \qquad (1)$$

From the definition of the dimensionless temperature in Eq. (3) and the conjecture that the the interface temperature is same as the saturation temperature of the pure refrigerants, the evaporative heat transfer coefficient is represented as follows :

$$h = \frac{q''_w}{t_w - t_s} = \frac{C_L \sqrt{\tau_w \rho_L}}{t_i^+} \qquad (5)$$

Once the shear stress at the inner tube wall and the liquid film thickness of liquid layer at a given quality are known, the heat transfer coefficient can be obtained. In order to calculate these shear stress and liquid film thickness, the relations between the mass flow rates and the velocities of the refrigerant in liquid and vapor phases are used (Hashizume et al., 1985), which are shown in Eqs. (6a) and (6b), respectively.

$$\frac{G(1-x)R}{2\mu_L} = \int_0^{m_L^+} \left(1 - \frac{y_L^+}{R_L^+} \right) u_L^+ dy_L^+ \qquad (6a)$$

$$\frac{GxR}{2\mu_G} = \int_{m_G^-}^{R_G^-} \left(1 - \frac{y_G^+}{R_G^+} \right) u_G^+ dy_G^+ \qquad (6b)$$
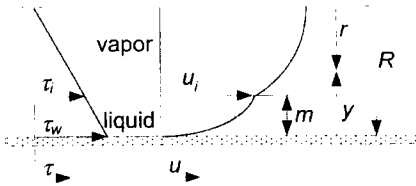
Fig. 2 Annular flow model of an forced convective boiling process

Annular flow model in this work is illustrated in Fig. 2. The entrainment of liquid droplets into the vapor core is neglected and the vapor-liquid interface is assumed to be smooth. It is also assumed that the circumferential liquid film thickness at any location is constant. Fully developed turbulent velocity profile is used to describe radial velocity distribution. Introducing the mixing-length theory for a single-phase flow (Kays and Crawford, 1993), dimensionless turbulent velocity, u^+, is expressed for three different regions.

For viscous sublayer in the liquid film, the dimensionless turbulent velocity is represented by

$$u_L^+ = y_L^+ - \frac{(y_L^+)^2}{2R_L^+} \qquad (7)$$

for $0 \le y_L^+ \le b_L^+$ and dimensionless thickness of viscous sublayer, b_L^+ is chosen as 11.6. For fully turbulent region in the liquid film for $b_L^+ \le y_L^+ \le m_L^+$,

$$u_L^+ = u_L^+\big|_{y_L^+=11.6} + 2.5 \ln \left[y_L^+ \frac{1.5(2-y_L^+/R_L^+)}{1+2(1-y_L^+/R_L^+)^2} \right] \\ -2.5 \ln \left[b_L^+ \frac{1.5(2-b_L^+/R_L^+)}{1+2(1-b_L^+/R_L^+)^2} \right] \qquad (8)$$

For fully turbulent region in the vapor core, the same nondimensional turbulent velocity profile is used for $m_G^+ \le y_G^+ \le R_G^+$.

$$u_G^+ = u_{G,i}^+ + 2.5 \ln \left[y_G^+ \frac{1.5(2-y_G^+/R_G^+)}{1+2(1-y_G^+/R_G^+)^2} \right] \\ -2.5 \ln \left[m_G^+ \frac{1.5(2-m_G^+/R_G^+)}{1+2(1-m_G^+/R_G^+)^2} \right] \qquad (9)$$

Using the velocity profiles expressed in Eqs. (7)-(9), we can find both the shear stress at the wall and the liquid film thickness which satisfies Eq. (6) and the heat transfer coefficient can be

calculated by Eq. (5). The velocity profile at the vapor core is proportional to the logarithm of dimensionless distance (Gill et al., 1964) (see Eq. (9)).

For refrigerant mixtures, the same procedure is applied to calculate the heat transfer coefficients. In this case, transport properties of refrigerant mixtures are calculated by using nonlinear mixing rules (Reid et al., 1987).

3. Calculation of Heat Transfer Coefficient and Discussion

3.1 Verification of the calculation of evaporative heat transfer coefficients

Calculated local heat transfer coefficients are compared with several experimental data sets (Hihara et al., 1989; Jung et al., 1989; Murata and Hashizume, 1993; Ross et al., 1987). For pure refrigerants (R22, R12, R114, R123, and R152a), average root-mean-square deviation of the calculated heat transfer coefficients from the experimental data is 14%. For binary refrigerant mixtures (R-22/114, R-22/12, R-134a/123, and R-32/134a), the average deviation is 18%. In this work, the experimental data of which the qualities are over 0.2 are selected because nucleate boiling effect for low quality region is not taken into account in this analysis.

3.2 Evaporation of nonazeotropic binary refrigerant mixture

When a nonazeotropic binary refrigerant mixture evaporates, the composition in the vapor phase is different from that in the liquid phase. In this section, the effect of this composition difference on heat transfer is analyzed quantitatively by using appropriate mixing rules for the calculation of mixture properties.

For a given overall composition, y_o, the compositions of both phases are y_L and y_G. In order to figure out the effect of composition difference on the prediction of heat transfer coefficients, two methods to estimate mixture properties are considered; one method using the overall composition, y_o, and the other using the compositions of each phase. Although the first method is not
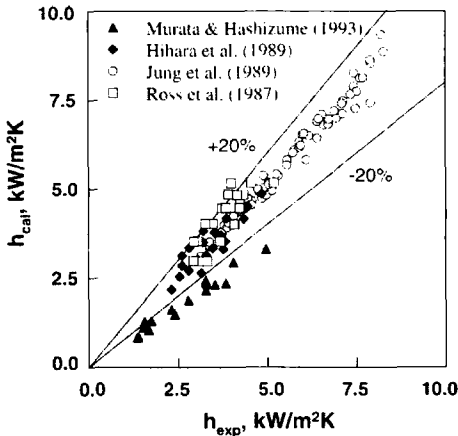
realistic, it is introduced in order to investigate the effect of the composition difference in both

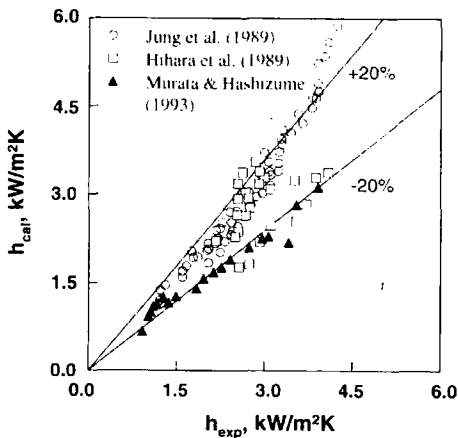**Table 1** Four cases to calculate properties of refrigerant mixtures

Case	Composition	Mixing rule
I	y_0	linear
II	y_0	nonlinear
III	y_L, y_G	linear
IV	y_L, y_G	nonlinear

(a) Hihara et al. (1989): R-22, R-12, R-114 ; Murata & Hashizume (1993): R-123 ; Jung et al. (1989): R-22 ; Ross et al. (1987): R-152a


(b) Hihara et al. (1989): R-22/12, R-22/114; Murata & Hashizume (1993): R-134a/123 ; Jung et al. (1989): R-22/114

Fig. 3 Comparison of the predicted heat transfer coefficients with experimental data from various sources

phases (y_L and y_G).

The effect of nonlinear mixing rules of transport properties on the heat transfer coefficients are also considered in this study. For the calculation of dynamic viscosity of mixtures for liquid and vapor phases, Grunberg & Nissan's equation and Wilke's equation are used respectively (Reid et al., 1987). The thermal conductivity of mixtures is calculated by Filippov's equation (Reid et al., 1987). Four cases to calculate the mixture properties are summarized in Table 1.

3.3 Heat transfer degradation

Heat transfer coefficients of R-22/114 and R-22/12 are shown in Fig. 4 as a function of mole fraction of the more volatile component, R-22. The heat transfer coefficients of refrigerant mixtures are lower than a linearly interpolated value of those of both pure refrigerants (straight line in Fig. 4), which is often used as a reference without
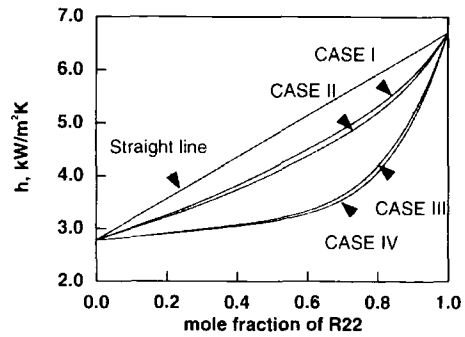
(a) R-22/114
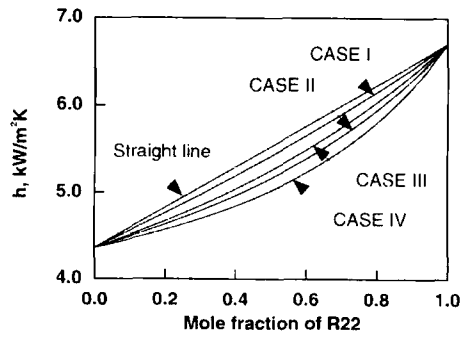
(b) R-22/12

Fig. 4 Predicted heat transfer coefficients for binary refrigerant mixtures with respect to composition for $G = 300$ kg/m²s, $P = 300$ kPa, $x = 0.7$, and tube diameter$=10$ mm

any physical background. The heat transfer degradation is defined as the difference between the linearly interpolated value and the predicted or experimental heat transfer coefficient. The degradation for R-22/12 mixture is small compared with that for R-22/114.

Because the major heat transfer resistance exists in the viscous sublayer in a turbulent flow, heat transfer coefficient shown in Eq. (5) can be rewritten as follows neglecting the contribution of the second term in Eq. (4).

$$h = \frac{c_L \sqrt{\tau_w \rho_L}}{t_i^+} \approx \frac{c_L \sqrt{\tau_w \rho_L}}{13.2 \mathrm{Pr}_L} = \frac{k_L \sqrt{\tau_w \rho_L}}{13.2 \mu_L} \qquad (10)$$

In Eq. (10), it is acknowledged that the heat transfer coefficient is affected mainly by thermal conductivity, dynamic viscosity, density of the liquid phase and the shear stress at the wall. In order to look into the heat transfer degradation, the property, ϕ (such as thermal conductivity or dynamic viscosity) of case I (using y_o), and case III (using y_L) with linear mixing rules are represented in Eq. (11) and the heat transfer coefficients of cases I, III are expressed in Eq. (12) for binary refrigerant mixtures.

$$\begin{aligned} \phi_{CASE\ I} &= \phi_1 y_o + \phi_2 (1 - y_o), \\ \phi_{CASE\ III} &= \phi_1 y_L + \phi_2 (1 - y_L) \end{aligned} \qquad (11)$$

$$\begin{aligned} h_{CASE\ I} &= \frac{\sqrt{\tau_w \rho_L}}{13.2} \left\{ \frac{k_{L,CASE\ I}}{\mu_{L,CASE\ I}} \right\}, \\ h_{CASE\ III} &= \frac{\sqrt{\tau_w \rho_L}}{13.2} \left\{ \frac{k_{L,CASE\ III}}{\mu_{L,CASE\ III}} \right\} \end{aligned} \qquad (12)$$

Considering only the cases of smaller liquid mole fraction than overall mole fraction ($y_L < y_o$), the degradation of heat transfer relative to the case I is represented as follows.

$$\frac{h_{CASE\ I} - h_{CASE\ III}}{h_{CASE\ I}} = \frac{\left(\frac{k_{L1}}{k_{L2}} - \frac{\mu_{L1}}{u_{L2}}\right)\left(\frac{1}{y_L} - \frac{1}{y_o}\right)}{\left(\frac{k_{L1}}{k_{L2}} + \frac{1 - y_o}{y_o}\right)\left(\frac{\mu_{L1}}{\mu_{L2}} + \frac{1 - y_L}{y_L}\right)} \qquad (13)$$

Relative degradation of heat transfer is affected by the liquid conductivity ratio, the liquid viscosity ratio of two components in the binary refrigerant mixture and the difference between y_L and y_o. By differentiating Eq. (13) with respect to the above quantities, it could be derived generally

that the heat transfer degradation becomes greater with larger k_{L1}/k_{L2}, smaller μ_{L1}/μ_{L2} and bigger difference between y_L and y_o in any binary refrigerant mixtures. As pointed out by Kedzierski et al., the difference between the vapor and liquid mass fraction represents the potential of concentration gradients within the liquid phase, which will affect the temperature distribution (Kedzierski et al., 1992).

For example, the thermal conductivity ratios for R-22/114 and R-22/12 mixtures are 1.35 and 1.25, respectively for $G = 300$ kg/m²s, $P = 300$ kPa, $x = 0.7$, and $y_o = 0.5$. The dynamic viscosity ratios are 0.442 for R-22/114 mixture and 0.815 for R-22/12 mixture, respectively. The difference between the overall composition and the liquid composition is bigger for R-22/114 mixture. Consequently, the heat transfer degradation for R-22/114 is larger than that for R-22/12.

3.4 Effect of mass transfer

When a nonazeotropic binary refrigerant mixture evaporates, concentration gradients exist near the vapor-liquid interface due to the preferential evaporation of more volatile component from the liquid phase close to the interface. This phenomenon makes the liquid composition near the interface, y_{Li}, less than the composition of liquid phase, y_L. Also the concentration gradients in the vapor phase exist in a similar way. This concentration gradient induces a motion of fluid

(a) more volatile   (b) less volatile
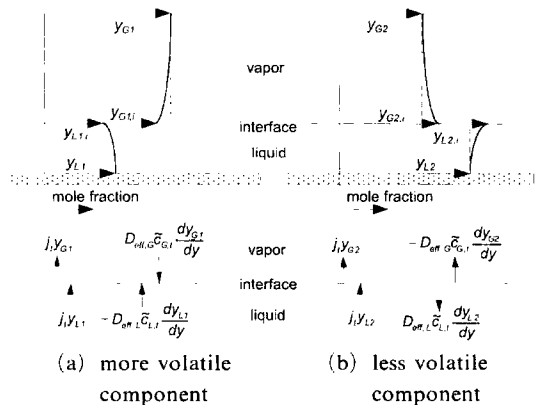   component   component

Fig. 5 Typical concentration gradient near liquid-vapor interface

to obtain the equilibrium. This mass transfer has an effect on the temperature distribution which has a direct influence on heat transfer (Kedzierski et al., 1992). As a result, interfacial temperature changes above the vapor temperature and this situation is illustrated in Fig. 5 and 6.

Total mass flux over the tube length L should be known. If the qualities and enthalpies are assigned at locations 1 and 3 in Fig. 7, the tube length is calculated according to the energy balance equation (14) and then the total mass flux at the interface, m_i'', is obtained by the mass balance equation (15) in the liquid phase for the control volume in Fig. 7.

$$G \frac{\pi D^2}{4}(i_3 - i_1) = \pi D L q_w'' \qquad (14)$$

$$G \frac{\pi D^2}{4}\{(1-x_1)-(1-x_3)\}=\pi(D-2m_2)Lm_i'' \qquad (15)$$

The molar flux of component 1 in a binary refrigerant mixture passing through the vapor-liquid interface shown in Fig. 5 is represented by (Price and Bell, 1974)
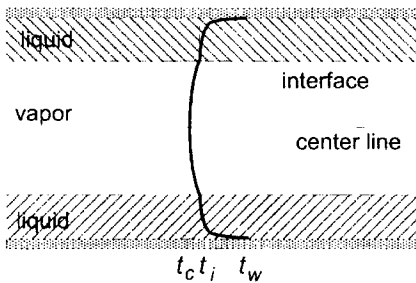

Fig. 6 Typical temperature profile of binary refrigerant mixture during evaporation
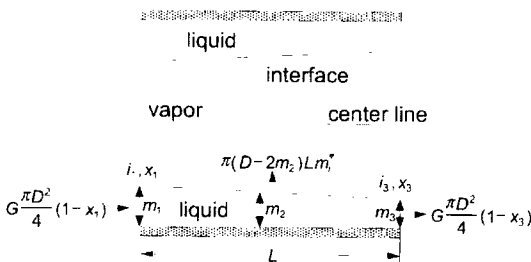

Fig. 7 Control volume for mass transfer analysis in liquid layer

$$j_1 = j_t y_{G1} - D_{eff,G}\tilde{c}_{G,t}\frac{dy_{G1}}{dy}$$
$$= j_t y_{L1} - D_{eff,L}\tilde{c}_{L,t}\frac{dy_{L1}}{dy} \qquad (16)$$

where

$$j_t = j_1 + j_2, \quad j_1 = \frac{m_{1i}''}{M_1}, \quad j_2 = \frac{m_{2i}''}{M_2} \qquad (17)$$

The total mass flux in Eq. (15) is rewritten in terms of molar flux as

$$m_i'' = j_1 M_1 + (j_t - j_1) M_2 \qquad (18)$$

The following relation is derived using the equivalent-laminar-film model (Butterworth and Hewitt, 1977) where the concentration gradient is limited within the equivalent layer of thickness of δ_m and the other vapor core region has the same composition with no concentration gradient.

$$\frac{j_t \delta_m}{D_{eff,G}\tilde{c}_{G,t}} = \ln \frac{y_{G1,c} - j_1/j_t}{y_{G1,i} - j_1/j_t} \qquad (19)$$

where

$$\delta_m = \frac{D_{eff,G}}{h_D} \qquad (20)$$

In order to apply Eq. (16) at the vapor-liquid interface, the following molar transfer by concentration gradient is used with known velocity profile in the liquid film.

$$D_{eff,L}\tilde{c}_{L,t}\frac{\partial y_{L1}}{\partial y}\Big|_t = \tilde{c}_{L,t}\frac{\partial y_{L1}}{\partial z}\int_0^m u\,dy \qquad (21)$$

The derivative, $\frac{\partial y_{L1}}{\partial z}$, is calculated from the change of equilibrium concentration in the liquid phase over the length L shown in Fig. 7. Solving Eqs. (18), (19), and (16) simultaneously, the molar fluxes, j_1, j_t, are obtained and mass fluxes at the interface, m_{1i}'', m_{2i}'', are easily calculated from Eq. (17).

Temperature profile of binary refrigerant mixture will be different from that of pure refrigerant because of the interfacial mass transfer as shown in Fig. 5. Sensible heat flux into the vapor core from the interface is represented as the sum of; a) the vapor sensible heat flux from the interface due to the bulk flow, b) the heat conducted through the equivalent-laminar film (Price and Bell, 1974).

$$q''_G = (t_i - t)\{ m''_{1i}c_{G1} + m''_{2i}c_{G2} \} - k_{G,t}\frac{dt}{dy} \quad (22)$$

Integration of this equation through the equivalent-laminar film in the vapor phase gives

$$q''_G = \frac{C}{1 - e^{-C}} h_G(t_i - t_c) = h'_G(t_i - t_c) \quad (23)$$

where

$$C = \frac{m''_{1i}c_{G1} + m''_{2i}c_{G2}}{h_G}, \quad h_G = \frac{k_{G,t}}{\delta_H} \quad (24)$$

The heat transfer coefficient of binary refrigerant mixture with mass transfer effect, h_{eff}, is represented as follows using the heat transfer coefficient without mass transfer effect, h_L, obtained in the previous section (Collier and Thome, 1994).

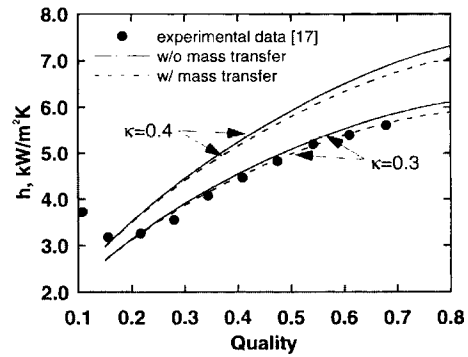$$\frac{1}{h_{eff}} = \frac{1}{h_L} + \frac{q''_G/q''_L}{h'_G} \quad (25)$$

where q''_G/q''_w is calculated from the energy balance for the control volume shown in Fig. 7.

Modified heat transfer coefficients with mass transfer effect of R-22/114 are shown in Fig. 8. It is shown that heat transfer is degraded by the effect of mass transfer. The major influence on the heat transfer degradation is attributed to the temperature change due to the concentration gradient near the interface.
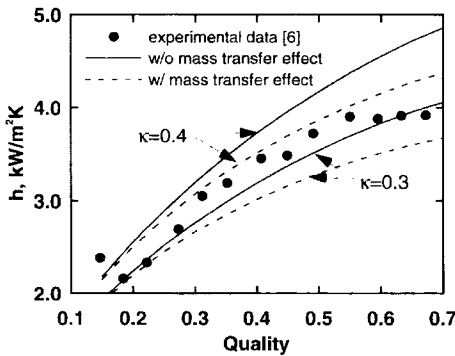
Figure 8 also shows the predicted heat transfer coefficients and the experimental data of Jung et al. (1989). The deviations in this case are about 16% without mass transfer consideration and around 10% when mass transfer effect is taken
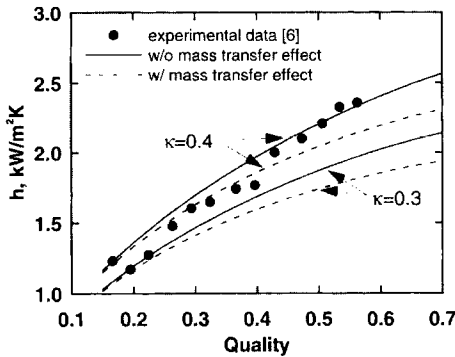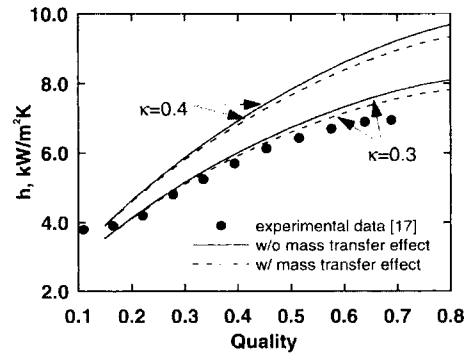

(a) R-22/114, $y_0 = 0.480$, $G = 246.5$ kg/m²s, $q''_w = 10$ kW/m²

(b) R-22/114, $y_0 = 0.485$, $G = 498.5$ kg/m²s, $q''_w = 26$ kW/m²

Fig. 8 Comparison of the predicted heat transfer coefficients with experimental data for R-22/114 mixture by Jung et al. [6] with respect to quality

(a) R-32/134a, $y_0 = 0.415$, $G = 424.0$ kg/m²s, $q''_w = 25$ kW/m²

(b) R-32/134a, $y_0 = 0.415$, $G = 583.0$ kg/m²s, $q''_w = 30$ kW/m²

Fig. 9 Comparison of the predicted heat transfer coefficients with experimental data for R-32/134a mixture by Shin et al. [17] with respect to quality

into account. For low mass and heat flux condition, the heat transfer coefficients are well predicted. As the mass and heat flux increase, the heat transfer coefficients are overpredicted. This phenomenon can be explained by the entrainment of droplets into the vapor core. As stated by Gill et al. (1964), the velocity profile is changing due to the suppression of turbulent eddies by the droplets of entrained liquid and they found that the reduced mixing length is expected when the entrainment is not negligible (Gill et al., 1964). If the reduced mixing length constant in the vapor core is used, the temperature difference between the interface and the vapor core is expected to increase, thus the predicted heat transfer coefficients will be smaller as in Figs. 8 and 9. This study could be evidence that the entrainment is not negligible as the mass flow rate increases in the real experiment. For another set of experimental data given by Shin et al. (1997), this can be figured out in Fig. 9.

4. Concluding Remarks

Forced convective boiling heat transfer coefficients have been predicted for an annular flow of pure refrigerants and nonazeotropic binary refrigerant mixtures based on the turbulent temperature profile in liquid film and vapor core. Composition difference between vapor and liquid phases, and nonlinear mixing rules in the calculation of mixture properties are also taken into account.

For pure refrigerants, average root-mean-square deviation of the predicted heat transfer coefficients from the available experimental data is ±14% for quality range over 0.2. For binary refrigerant mixtures without consideration of mass transfer effect, average deviation is calculated as ±18%.

It is shown that the heat transfer coefficients of binary refrigerant mixtures are lower than those obtained from a linear interpolation based on the data of both pure refrigerants. This degradation is analyzed by several factors such as the deviation of liquid composition from overall composition, conductivity ratio and viscosity ratio of two pure

refrigerants. Heat transfer degradation becomes greater with larger conductivity ratio and smaller viscosity ratio of more and less volatile components. Larger difference between y_L and y_o in any binary refrigerant mixtures will give lower heat transfer coefficients.

It is also found that the temperature change due to the concentration gradient is an important factor for the degradation. At low mass and heat flux conditions, the heat transfer coefficients are well predicted, but the heat transfer coefficients are overpredicted as the mass and heat flux increase.

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