An Experimental Investigation of Heat Transfer in Forced Convective Boiling of R134a, R123 and R134a/R123 in a Horizontal Tube

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This paper reports an experimental study on flow boiling of pure refrigerants R134a and R123 and their mixtures in a uniformly heated horizontal tube. The flow pattern was observed through tubular sight glasses with an internal diameter of 10 mm located at the inlet and outlet of the test section. Tests were run at a pressure of 0.6 MPa in the heat flux ranges of 5-50 kW/m^2, vapor quality 0-100 percent and mass velocity of 150-600 kg/m^{2}s. Both in the nucleate boiling-dominant region at low quality and in the two-phase convective evaporation region at higher quality where nucleation is supposed to be fully suppressed, the heat transfer coefficient for the mixture was lower than that for an equivalent pure component with the same physical properties as the mixture. The reduction of the heat transfer coefficient in mixture is explained by such mechanisms as mass transfer resistance and non-linear variation in physical properties etc. In this study, the contribution of convective evaporation, which is obtained for pure refrigerants under the suppression of nucleate boiling, is multiplied by the composition factor by Singal et al. (1984). On the basis of Chen's superposition model, a new correlation is presented for heat transfer coefficients of mixture.

Key Words: Convective Boiling, Flow Pattern, Heat Transfer, Horizontal Tube, Mixture

Nomenclature

a : Thermal diffusivity (m^2/s)
Bo : Boiling number ($=q/Gh_{fg}$)
Co : Convection number ($=(\rho_v/\rho_l)^{0.5}((1-\beta)/\beta)^{0.8}$)
C_p : Specific heat (J/kgk)
D : Tube Diameter (m)
g : Gravitational acceleration (m/s^2)
h : Specific enthalpy (J/kg)
h_{fg} : Latent heat of vaporization (J/kg)
Fr : Froude number
G : Mass velocity (kg/m^{2}s)
k : Thermal conductivity (W/mK)
P : Pressure (Pa)
q : Heat flux (W/m^2)
Re : Reynolds number
T : Temperature (K)
X : Mole fraction in liquid
X_{tt} : Martinelli parameter
Y : Mole fraction in vapor
z : Axial distance (m)

Greek Letters

α : Heat transfer coefficient (W/m^{2}K)
β : Vapor quality
θ : Contact angle ($=35°$)
ΔT_{bp}: Boiling point range (K)

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ΔT_{id} : Ideal wall superheat (K)
ρ : Density (kg/m^3)
σ : Surface tension (N/m)
μ : Viscosity (Pa·s)
ν : Kinematic viscosity (m^2/s)

Subscripts

c : Critical
in : Inlet of the heat transfer section
l : Liquid
nb : Nucleate boiling
s : Saturation
v : Vapor

1. Introduction

In the past decade, a number of experimental studies have been carried out to predict the heat transfer performance on refrigerants that substitute for CFCs that cause the ozone layer to delete. Of such new refrigerants, zeotropic mixtures attract the special attention because of their variable temperature feature during phase change in evaporators or condensers, which leads to a reduction in available energy loss in such heat exchangers. In spite of these advantages, mixtures generally show lower heat transfer performance during their phase change compared with pure fluids of the same physical properties as the mixtures. With zeotropic mixtures, an amount of data for heat transfer coefficient was obtained on horizontal in-tube flow boiling, and also various correlations from these data were developed.

Webb and Gupte (1992) classified heat transfer correlations proposed until now into three groups of models : the superposition, the enhancement and the asymptotic models. The superposition model was originated by Chen (1966), which was followed by Bennett and Chen (1980), Gungor and Winterton (1986), Jung (1989) and others. Typical correlations based on the enhancement model are those of Shah (1976), Gungor and Winterton (1987), and Kandlikar (1990, 1991). The asymptotic model was employed in Kutateladze (1961), Liu and Winterton (1988), and Klimenko (1988) in developing their correlations.

Chen divided the two-phase flow boiling heat transfer into the nucleate boiling contribution and the convective evaporation contribution, which include a suppression factor and an enhancement factor, respectively. The suppression factor is correlated as a function of the two-phase Reynolds number and the enhancement factor as a function of the Martinelli parameter that does not depend on heat flux.

Shah proposed the enhancement factor as a function of the boiling number and the convection number that was used instead of the Martinelli parameter. The nucleate boiling contribution and the convective evaporation contribution are calculated first and the larger one is chosen as the two-phase heat transfer coefficient.

Gungor and Winterton modified the Chen correlation based on a data base containing over 4200 data points for seven fluids. They assumed that the enhancement factor is a function of both the Martinelli parameter and the boiling number, and a functional relationship between them was obtained for vertical tubes. They introduced the boiling number in order to consider the heat transfer improvement as a result of a disturbed liquid film due to the generation of vapor.

Jung et al. developed a correlation for pure fluids (R22, R114) in the same form as the Chen correlation. The correlation of Jung et al. showed a mean deviation of 7.2% and 9.6% for pure fluid and mixture data more than 1200 data points they obtained. In order to consider the effect of heat flux in nucleate boiling region, the suppression factor by Chen is alternated by the factor, which is expressed as a function of the Martinelli parameter and the boiling number.

Kandlikar developed a correlation using the same parameters as Shah, and added an empirical fluid-dependent parameter. This parameter has to be determined experimentally for every fluid. For horizontal tubes with the Froude number more than 0.04, he divided flow boiling heat transfer into two regions : the nucleate boiling dominant and the convective boiling dominant regions.

The objectives of the present study are to obtain the experimental data for pure refrigerants R134a and R123, and their mixture in horizontal flow

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boiling and to estimate the heat transfer characteristics of pure refrigerants and their mixture due to mixing composition, and finally to propose the correlations which are able to predict not only heat transfer coefficients of pure refrigerants but also those reduced by mixtures. Especially, a transition criterion to annular flow is introduced in a new suppression factor, which is expressed in terms of the boiling number and the modified Froude number. In this study, the reason of using zeotropic refrigerant mixtures of R134a and R123 as the test fluid is that these mixture have moderate levels of temperature and pressure in performing experiments, large deviation between the dew and bubble point curves, and due to the availability of property data including both pure components.

2. Experimental Apparatus and Procedure

2.1 Flow loop and test section

The schematic of the experimental apparatus is shown in Fig. 1. The circulation loop of test fluid consists of a reservoir tank, pump, flow meters, mixing chambers, preheaters, the sight glass sections, the heat transfer section, condenser and other accessories.

Test fluid in the tank is pumped through a strainer and the 1st preheater to the mixing chamber where temperature and pressure are measured in a subcooled liquid state. Then the fluid is heated in the 2nd preheater to a prescribed enthalpy and enters the heated test section where the fluid evaporates on the tube wall heated at uniform heat flux. Flow patterns of boiling fluid are observed at the upstream and downstream of the test section through glass tubes of the same diameter as the test tube. Figure 2 shows the test section, a 3 m-long stainless steel tube of 10 mm I.D. and 1.5 mm wall thickness, the central 2 m of which is the heat transfer section and is heated by directly passing stabilized AC that is supplied from a low-voltage and high-current transformer. Electric heating is allowed so as to supply a constant heat flux to the fluid flowing inside a tube along a fixed tube length. Also, we can obtain a desired quality at the inlet of the test section by adjusting heat flux at 2nd preheater. This method, however, can bring upon burn-out in case that the flow is stratified or tube is partially dried out.

Heated tube wall temperature is measured at ten axial locations of the heat transfer section by Cromel-Alumel thermocouples spot-welded on the tube outer surface. The first location is 10 mm downstream of the inlet of heated section, and succeeding nine locations are aligned at an equal interval of 200 mm. At each location, the tube temperature is measured at six peripheral positions at angles of 0°(top), 45°, 90°(side), 135°, 180°(bottom) and 270°(another side) in the clockwise direction. The inside tube wall temperature and heat flux to fluid are calculated from the measured outside wall temperature assuming one-

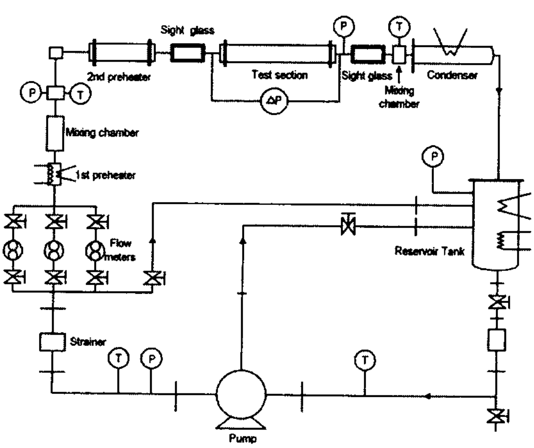

Fig. 1 Experimental apparatus

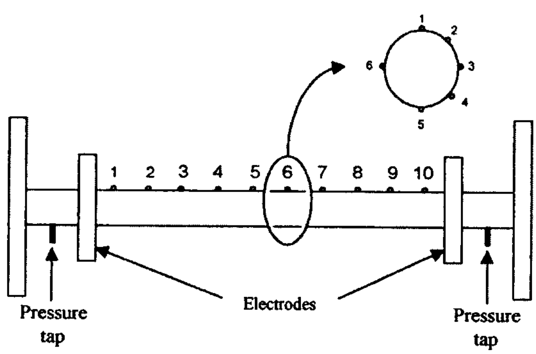

Fig. 2 Test section

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dimensional heat conduction and accounting for uniform heat generation in the wall and heat loss to the surroundings.

Though the heated test section and preheaters are well insulated with glass fiber in order to reduce heat loss to the surroundings, heat loss is inevitable. The heat loss was calibrated as a function of the temperature difference between the tube wall and ambient air, and used in calculating the tube inside temperature and heat flux.

Fluid temperature and pressure are measured in the mixing chambers at the inlet and exit of the test section. Pressure drop across the pressure taps are measured using differential pressure transducer. These data of fluid temperature and pressure are used to determine the local fluid temperature and pressure along the test section, as will be mentioned later.

Major parameters that affect heat transfer coefficient in flow boiling are mass velocity, heat flux and vapor quality. In the present experiment the mass velocity is set at 150, 300 and 600 kg/m^2s, and heat flux is varied at 5, 10, 20 and 50 kW/m^2. Inlet quality of the fluid entering the heated section is varied to realize a wide range of vapor quality in a limited 2 m length of the heated section.

2.2 Physical property of mixture

Table 1 shows several properties of pure refrigerants and their mixture used in the experiment at the saturation temperature and at the bubble point temperature corresponding to 0.6 MPa. Mole fraction of the more volatile component (R134a) in mixture is used in this investigation to express the mixture composition. The mole fractions presented in Table 1 were, prior to the test, drawn at mixing chamber before the test section and checked with gas chromatography. In determining the local thermodynamic conditions of mixture at an arbitrary cross section of the heated tube, the two-phase equilibrium data and thermodynamic properties of mixtures are required. Such data was obtained using the modified Benedict-Webb-Rubin equation of state with fifteen constant (1977). Figure 3 shows the phase diagram obtained in this way for 0.6 MPa.

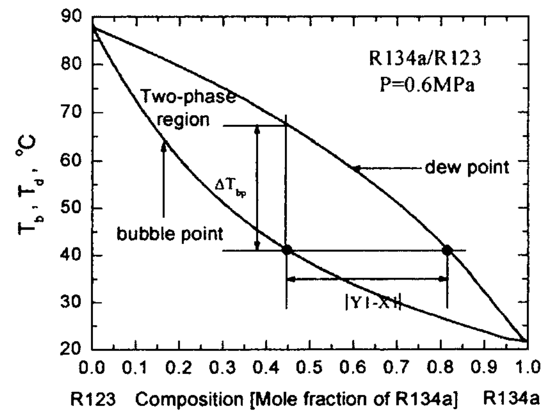

Fig. 3 Phase equilibrium diagram

Table 1 Physical properties for the saturation or bubble-point temperature at 0.6 MPa

Property	Unit	Pure fluid		Mixture (X_1=0.275)	Mixture (X_1=0.49)	Mixture (X_1=0.751)
		R134a	R123	R134a/R123	R134a/R123	R134a/R123
Saturation or bubble-point temperature	℃ [K]	21.55 [294.7]	88.244 [361.394]	53.313 [326.46]	38.9 [312.1]	28.55 [301.7]
Density liquid vapor	kg/m^3	1218.76 29.064	1285.3 35.92	1313.73 29.851	1304.02 29.07	1267.89 28.925
Viscosity liquid vapor	Pa·s	2.102×10^{-4} 1.227×10^{-5}	2.23×10^{-4} 1.29×10^{-5}	2.53×10^{-4} 1.288×10^{-5}	2.506×10^{-4} 1.26×10^{-5}	2.324×10^{-4} 1.196×10^{-5}
Specific heat liquid vapor	kJ/kgK	1.42 1.005	1.107 0.847	1.15 1.033	1.205 1.022	1.293 0.869
Latent heat	kJ/kg	181.07	140.58	163.63	171.12	180.427
Surface tension	N/m	8.378×10^{-3}	8.03×10^{-3}	9.78×10^{-3}	9.87×10^{-3}	9.259×10^{-3}

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2.3 Definition of heat transfer coefficient

Heat transfer coefficient at an axial distance, z, from the inlet of the heat transfer section is defined as

$$\alpha = q/(T_w - T_b) \tag{1}$$

where q is heat flux, T_w tube inside wall temperature, and T_b bulk fluid temperature. The bulk temperature is determined so as to satisfy the following heat balance equation from the inlet to the location of interest in the heat transfer section.

$$h = h_{in} + \frac{4qz}{GD} \tag{2}$$

Here h_{in} is fluid enthalpy at the inlet, G mass velocity and D tube inside diameter. Assuming thermodynamic equilibrium, the specific enthalpy of mixture, h, and the vapor quality, β, are expressed as

$$h = h_l(1-\beta) + h_v\beta,\ \beta = \frac{X^*_{1,in} - X^*_1}{Y^*_1 - X^*_1} \tag{3}$$

Where the superscript * denotes the mass fraction, and the liquid and vapor enthalpies are a function of pressure, temperature and composition, respectively, as

$$h_l = h_l(P,\ T_b,\ X_1),\ h_v = h_v(P,\ T_b,\ Y_1) \tag{4}$$

Fluid pressure, P, is determined from a linear interpolation of measured pressure drop across the heated section. Hence, P, h and $X^*_{1,in}$, are known at each cross section, so that the remaining four unknown variables, T_b, X_1 and Y_1, are uniquely determined from the Benedict-Webb-Rubin equation of state with fifteen constant so as to satisfy the above Eqs. (3) to (4).

Tube wall temperature varies around a tube, so that heat transfer coefficient is defined at each peripheral positions. Average heat transfer coefficient around the tube is defined using an average wall temperature.

$$T_w = [T_o + 2(T_{45} + T_{90} + T_{135}) + T_{18}]/8 \tag{5}$$

In the present test, the quantities measured directly were flow rate, pressure and electrical power. Therefore, the uncertainty in the heat transfer coefficient in eq. (1) is caused by uncertainties in the heat flux and the wall superheat. The uncertainty in heat flux due to the measurements of electrical current and voltage was estimated to be 2~3%. The uncertainty in the wall temperature measurement was estimated to be 0.2℃ judged from a calibration of thermocouple. The bulk fluid temperature T_b was determined by the accuracies of the refrigerant pressure measurement (1 kPa, corresponding to 0.12℃ in saturation temperature) and flow rate measurement (1%, corresponding to 0.1℃ in saturation temperature). From these analyses, the uncertainty in the heat transfer coefficient in eq. (1) was estimated to be within ±10%.

3. Heat Transfer Characteristics

In order to investigate heat transfer characteristics during horizontal flow boiling, heat transfer experiments were accomplished at various conditions of heat flux and mass velocity at a constant pressure of 0.6 MPa.

3.1 Temperature variation of pure and mixed refrigerants

The circumferential variations of the measured wall temperature at the top, side, and bottom of the tube are plotted along the heated section together with axial variations of the bulk fluid temperature. Figure 4(a) and (b) show such a plot for pure and mixed refrigerants. For pure refrigerant, the wall temperature at the tube bottom is higher than that at the top and side because the liquid film at the bottom due to gravity is thicker than that at the others. Accordingly, heat transfer coefficient at the bottom of the tube is lower than that at the top. In annular flow realized at moderate mass velocities, the gravity force becomes more influential to the formation of annular liquid film. Thus liquid film is thinnest at the top of tube and thickest at the bottom, so that the heat transfer coefficient at the top is expected to be higher than that at the bottom. As vapor quality is increased, the decrease of the saturation temperature of pure refrigerant flowing into the test tube is accom-

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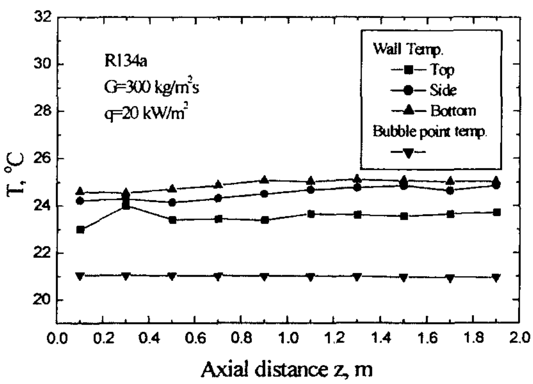
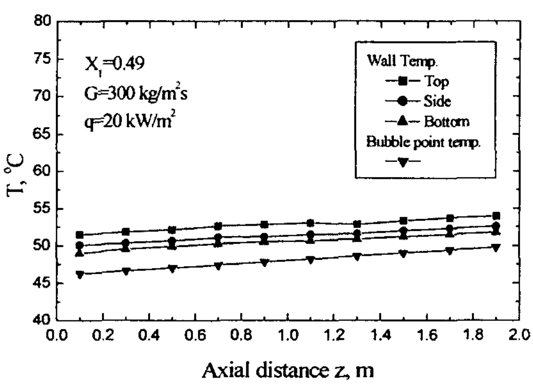
panied with the pressure drop in a tube during flow boiling. However, The refrigerant mixture represents an opposite variation compared with pure refrigerant as shown in Fig. 4(b). It is clear that wall temperature varies around the tube periphery, with higher temperature at the tube top compared to the bottom and side. The opposite phenomenon in mixture, different from pure refrigerant, is explained by the composition variation of the liquid film at the top and bottom of the tube. Jung et al. (1989) measured that local liquid compositions at the top are always smaller than the ones at the bottom in the test for R22/R114 mixture. The saturation temperature for mixture different from pure refrigerant increases as quality is increased as shown in Fig. 4(b). This is explained by phase equilibrium diagram shown in Fig. 3. In two-phase region of phase equilibrium diagram, the increase of quality is accompanied with the increase of the saturation temperature, which is called the gliding temperature effect.

3.2 Effects of heat flux and mass velocity

Figure 5 shows heat transfer coefficient versus

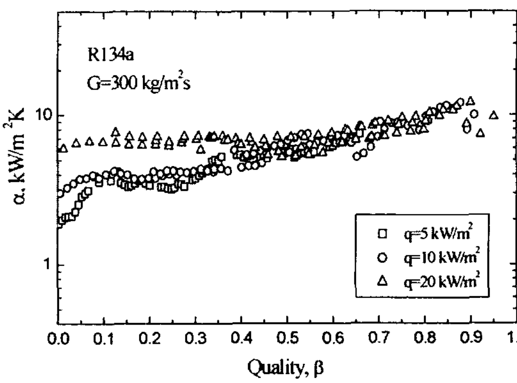


(a) Pure refrigerant (R134a)

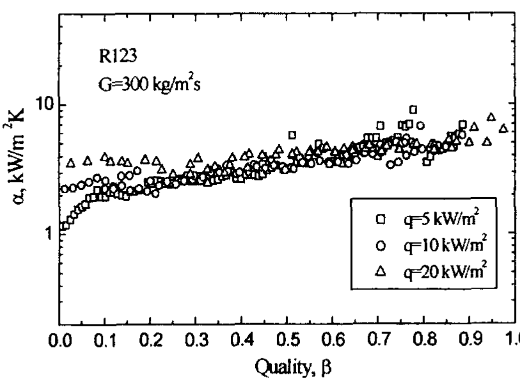


(b) Mixture (X_1=0.49)

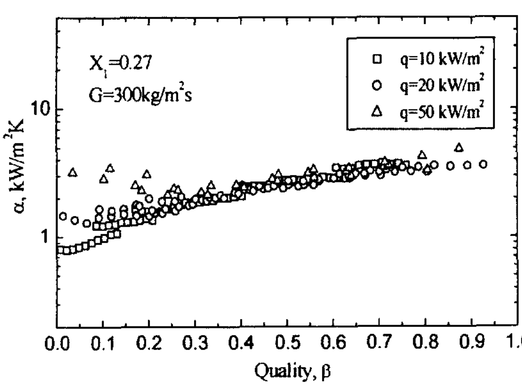
Fig. 4 Typical plots of wall and fluid temperature along the tube axial



(a) R134a



(b) R123



(c) X_1=0.27

Fig. 5 Heat transfer coefficient versus quality (Influence of heat flux)

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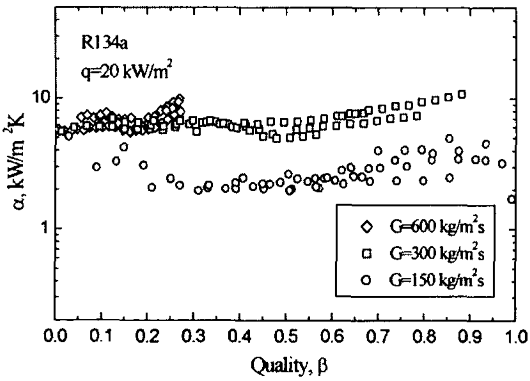
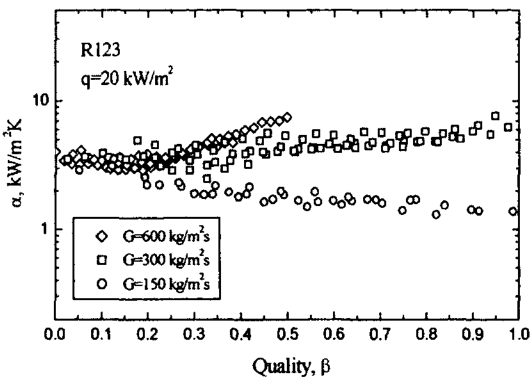
quality for $G=300$ kg/m^{2}s with heat flux as parameter. It is evident in this figure that, heat transfer coefficient is dependent on heat flux in the low quality region, with higher coefficients being at higher heat fluxes. This heat flux dependence of heat transfer coefficient indicates that nucleate boiling dominates heat transfer in this region.

As vapor quality is increased, however, the influence of heat flux becomes less significant and heat transfer coefficients at different heat flux tend to merge into a single curve and are dependent only on vapor quality. The independence of heat transfer coefficient on heat flux and the dependence only on quality mean that convective evaporation dominates heat transfer in the high quality region.

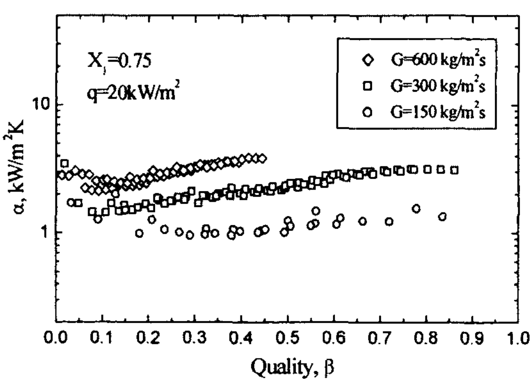
Figure 6 shows the influence of mass velocity on heat transfer coefficient for $q=20$ kW/m^2. Heat transfer coefficient increases in the whole quality region with an increase in mass velocity. Heat transfer characteristics obtained for mixture as mentioned above are qualitatively similar to those for pure fluids.



(a) R134a



(b) R123



(c) $X_1=0.75$

Fig. 6 Heat transfer coefficient versus quality (Influence of mass flux)

4. Correlation Development for Pure Refrigerant

4.1 Transition criterion from stratified to un-stratified flow

Flow patterns in the present experiments are observed through the sight glass tube downstream of the heated section. Flow patterns observed in this study are simply classified into stratified flow (including intermittent, stratified and stratified-wavy flow) and unstratified flow (annular flow). Shah (1982) used the liquid Froude number as the transition criterion between all wet and partially wet wall around the tube circumference during horizontal flow boiling. The liquid Froude number is given as

$$Fr_l=\frac{G^2}{\rho^2 gD} \qquad (6)$$

when $Fr_l<0.04$, Shah considered the tube wall as to be partially wetted. Also, Kandlikar (1990) obtained the same value of 0.04 based on a large experimental data bank. However, Wattelet et al. (1994) recommended a much higher value of 0.25 than Shahs value of 0.04. These different liquid Froude number values were shown to be inadequate for predicting the transition from stratified to unstratified flow. It is necessary to exactly predict the transition criterion from stratified to

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unstratified flow because it is strongly related to the heat transfer and pressure drop as well as flow pattern.

It was found in the previous work (2003b) that the transition from the stratified to the annular flow occurred at lower quality when mass velocity was increased. Thus the transition is not a unique function of the liquid Froude number but it also depends on the vapor quality. Accordingly, the following modified parameter is employed to establish the transition criterion between the stratified and the annular flow.

$$C_{Fr_l}=0.25+\frac{G^2}{\rho^2 gD}(1-\beta) \tag{7}$$

From the analysis of the present flow pattern

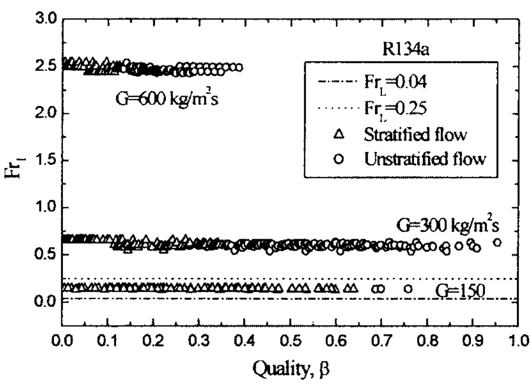

Fig. 7 Comparison between the two liquid froude number criterions and the experimental data

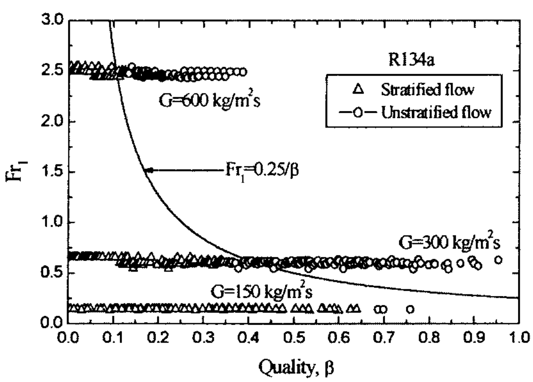

Fig. 8 Comparison of the experimental data against the modified liquid froude number calculated at the saturation temperature

data both for pure fluids and their mixtures, it was found that the stratified flow occurs if $C_{Frl}>Fr_l$, and the unstratified flow prevails if $C_{Frl}<Fr_l$. This criterion gives the condition for existence of annular flow as

$$Fr_l \geq \frac{0.25}{\beta} \tag{8}$$

Figure 7 shows the comparison between the two different liquid Froude number values and the experimental data for R134a with three mass velocities during horizontal flow boiling. As shown in this figure, it is found that the accuracy for predicting the transition from stratified-wavy to annular flow decreases as mass velocity is increased. Figure 8 indicates the observed flow patterns on the liquid Froude number versus quality map, where the transition boundary from the stratified-wavy flow to the annular flow is well predicted by eq. (8).

4.2 Heat transfer coefficient

Heat transfer for flow boiling in a tube can be basically thought of being governed by the nucleate boiling and convective evaporator mechanisms. This form called as superposition model was originated by Chen, and can be written as

$$\alpha=S\alpha_{nb}+F\alpha_l \tag{9}$$

where α_l is the convective heat transfer coefficient that is a function of flow parameters, and α_{nb} the nucleate boiling heat transfer coefficient that is dependent on heat flux. An enhancement factor, F, and a suppression factor, S, are empirical constants, which have been obtained by many authors from a curve fit to their heat transfer data, mostly of pure fluids and partly of mixtures.

At a limiting case that the nucleate boiling is fully suppressed, the influence of heat flux is no longer significant and heat transfer coefficient becomes a function of the vapor quality alone. Under such a situation, Eq. (9) becomes

$$\frac{\alpha}{\alpha_l}=F(1/X_{tt}) \tag{10}$$

Here, the convective heat transfer coefficient is that of the Dittus-Boelter (1930) which is used by

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replacing G with $G(1-\beta)$ for liquid only flow.

$$\alpha_l=0.023\frac{k_l}{D}\left[\frac{G(1-\beta)D}{\mu_l}\right]^{0.8}\left(\frac{C_{pl}\mu_l}{k_l}\right)^{0.4} \quad (11)$$

$$X_{tt}=\left(\frac{1-\beta}{\beta}\right)^{0.9}\left(\frac{\rho_v}{\rho_l}\right)^{0.5}\left(\frac{\mu_l}{\mu_v}\right)^{0.1} \quad (12)$$

As indicated in the previous work (2003a), the resulting form for F factor obtained from a curve fit of the present data under the assumption that nucleate boiling is fully suppressed, is expressed as

$$F=\frac{\alpha}{\alpha_l}=0.7+3.1(1/X_{tt})^{0.77} \quad (13)$$

The results obtained from the measured data of this study will be predicted by taking an additive form of heat transfer coefficient as Eq. (9). As mentioned earlier, the transition criterion from stratified to unstratified flow is used to derive the suppression factor, S, which becomes a function of heat flux, mass velocity, and quality. That is, this factor can be expressed by employing the boiling number and modified Froude number, Bo and C_{Fr}

$$S=f(C_{Fr}, Bo) \quad (14)$$

A regression analysis was carried out to correlate the present for pure component. The resulting form is

$$S=0.0031\left(\frac{1}{C_{Fr}\cdot\beta}\right)+Bo \quad (15)$$

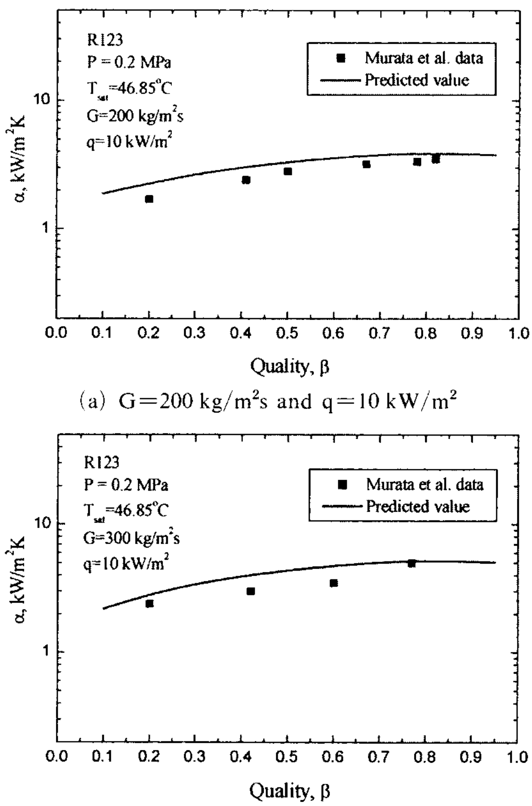
It should be noted that for $\beta<0.01$, S becomes 1. In this study, the nucleate boiling heat transfer coefficient in Eq. (9) was calculated using that of Stephan and Abdelsalam (1980), which can be expressed as

$$\alpha_{nb}=207\left(\frac{qd}{k_lT_s}\right)^{0.745}\left(\frac{\rho_v}{\rho_l}\right)^{0.581}\left(\frac{\nu_l}{a_l}\right)^{0.533}\left(\frac{k_l}{d}\right) \quad (16)$$

where, $d=0.0146\theta\sqrt{2\sigma/g(\rho_l-\rho_v)}$

Table 2 gives comparisons between the predicted heat transfer coefficients and the experimental results for R134a and R123. The present correlation proposed in this study predicts satisfactorily the present data for pure refrigerants within a mean deviation of 18%.

Figure 9(a) and (b) show the comparison between the present correlation and the R123 data obtained by Murata et al.(1993). They obtained the data using the horizontal tube of 10.3 mm I.D. with an electrical heating method. The present correlation is found to satisfactorily predict the data of Murata et al.(1993).



(a) G=200 kg/m²s and q=10 kW/m²

(b) G=300 kg/m²s and q=10 kW/m²

Fig. 9 Comparison between the predicted heat transfer coefficient and the experimental data obtained by Murata et al.

Table 2 Deviations between some correlations and the present data

Refrigerants	Shah (1976) Mean & Ave.		Gungor et al.(1987) Mean & Ave.		Kandlikar (1990) Mean & Ave.		This study Mean & Ave.	
R134a	32.8	4.4	48.6	24.6	38.2	−35.8	18.7	−4.9
R123	31.5	22.6	38.4	32.3	22.1	−19.3	17.4	10.9

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4.3 Correlation development for mixture

Figure 10 shows the typical results comparing heat transfer coefficient of mixture with those of pure fluids in a α/α_l versus $1/X_{tt}$ plot. It is seen that, for mixture, the normalized heat transfer coefficient is less than those for pure fluids, both in the small $1/X_{tt}$ region and in the large $1/X_{tt}$ region. The reduced heat transfer coefficient in the small $1/X_{tt}$ region is perhaps due to the reduction of nucleate boiling heat transfer coefficient in mixture as usual in pool boiling mentioned above. On the contrary, at the large $1/X_{tt}$ region where flow pattern is annular flow and convective evaporation is prevailing, the reduction of heat transfer coefficient is ascribed to the peripheral variation of mixture concentration in annular film. An annular liquid film is thinnest at the tube top so that evaporation is very intensive there. Intensive evaporation depletes the more volatile component, resulting in a rise of the bubble point temperature. In this way the annular film temperature is expected to vary around a tube, with the highest at the top and the lowest at the bottom. This non-uniformity of film temperature may be a main reason of heat transfer reduction in the high quality region.

Mishra et al.(1979) pointed out that the composition factor $(1-|Y_1-X_1|)$ was a parameter affecting the heat transfer behavior during flow boiling of binary mixture as indicated in Singal et al.(1984). Singal et al. examined, based on experiments for R12/R13 mixtures, whether the prediction of the heat transfer coefficient was improved by incorporating the composition factor into the form of Lavin-Youngs type (1965).

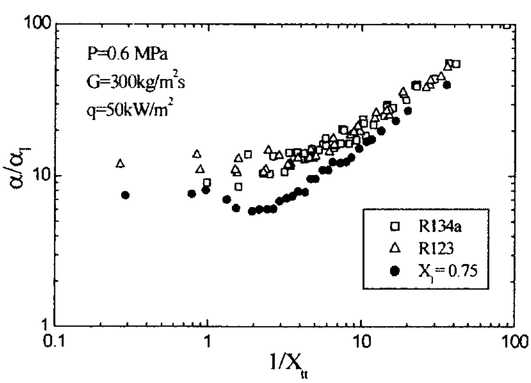

Fig. 10 Normalized heat transfer coefficient for mixture and pure fluid

In this study, the contribution of convective evaporation, which is obtained for pure refrigerant under the suppression of nucleate boiling, is multiplied by the composition factor affecting the heat transfer characteristics of mixture as estimated by Singal et al. (1984). Therefore, Eq. (10), for the mixture, is given as follows:

$$\frac{\alpha}{\alpha_l}=C_F F \tag{17}$$

where

$$C_F=a(1-|Y_1-X_1|)^b \tag{18}$$

A regression analysis was carried out to obtain the constants a and b in Eq. (18), and the resulting C_F factor is

$$C_F=0.407(1-|Y_1-X_1|)^{-1.398} \tag{19}$$

Fujita and Tsutsui (1994), to predict nucleate pool boiling heat transfer coefficients in binary mixtures, modified the Thome (1983) correlation as a function of heat flux, which was empirically determined for five different mixtures for the entire range of heat flux from the onset of nucleate boiling to the peak heat flux. The correlation is given as

$$\frac{\alpha}{\alpha_{id}}=\frac{1}{1+\left[1-0.8\exp\left(\frac{-q}{10^8}\right)\right]\left(\frac{\Delta T_{bp}}{\Delta T_{id}}\right)} \tag{20}$$

where the unit of heat flux is W/m^2. Because this correlation has the dimension form, they later changed the influence of heat flux into a dimensionless term as

$$\frac{\alpha}{\alpha_{id}}=\frac{1}{1+\left[1-\exp\left(-60\left(\frac{q}{\rho_v h_{fg}}\right)\left[\frac{\rho_v^2}{\sigma g(\rho_l-\rho_v)}\right]^{1/4}\right)\right]\left(\frac{\Delta T_{bp}}{\Delta T_{id}}\right)} \tag{21}$$

Equation (21) is used as the nucleate pool boiling heat transfer coefficient for mixture in this study. Based on the heat transfer correlation developed for pure refrigerants, the final form of that for mixtures becomes

$$\alpha=S\alpha_{F-Z}+FC_F\alpha_l \tag{22}$$

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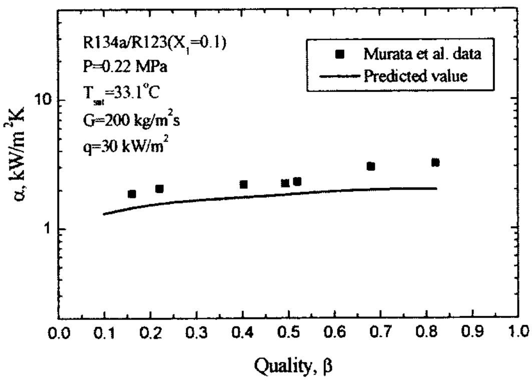
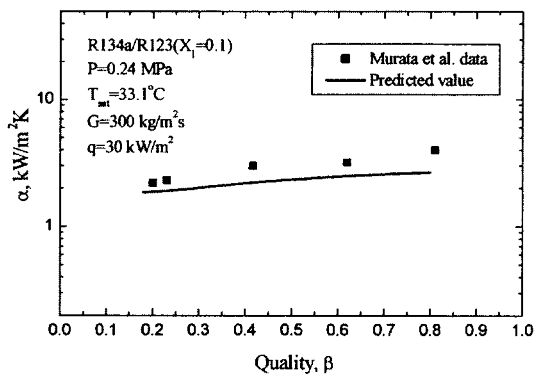
Table 3 Deviations between some correlations and the present data

Refrigerants	Shah (1976) Mean & Ave.		Gungor et al.(1987) Mean & Ave.		Kandlikar (1990) Mean & Ave.		This study Mean & Ave.	
$X_1=0.27$	31.9	17.6	45.7	36.1	46.4	−39.7	20.6	6.2
$X_1=0.49$	58.4	33.3	45.1	15.7	35.1	−34.8	18.2	−4.4
$X_1=0.75$	26.3	34.2	53.8	53.1	43.7	−31.6	16.1	7.8



(a) Mixture ($X_1=0.1$), $G=200\ kg/m^2s$ and $q=30\ kW/m^2$



(b) Mixture ($X_1=0.1$), $G=300\ kg/m^2s$ and $q=30\ kW/m^2$

Fig. 11 Comparison between the predicted heat transfer coefficient and the experimental data obtained by Murata et al.

Table 3 lists the mean deviation and the average deviation obtained with a few correlations. The correlation of Shah, Gungor and Winterton predicted the present data with a mean deviation of 39% and 48%, respectively. However, the correlation of Kandlikar considerably underpredicted most of the data, and showed the mean deviation of 42%. The present correlation proposed in this study predicts satisfactorily the present data for mixtures within a mean deviation of 18.3%.

Figure 11(a) and (b) show the comparison between the correlation proposed in this study and the R134a/R123 data obtained by Murata et al.(1993). It is found from Fig. 11 that the present correlation somewhat underpredicted the data of Murata et al.

5. Conclusions

An experimental study of flow boiling heat transfer for pure refrigerants R134a and R123, and their mixtures was performed in a uniformly heated horizontal tube. Based on the measured data, the following conclusions were reached.

(1) The wall temperature of pure refrigerants was found to be higher at the tube bottom compared with the top and side because the liquid film at the bottom is thicker than that at the top due to gravity. Their mixture, however, represented an opposite variation to pure refrigerants. This is explained by the composition variation of the liquid film at the top and bottom of the tube.

(2) Heat transfer coefficients for both pure refrigerants, at low quality, are dependent on heat flux, mass velocity and vapor quality. Influence of heat flux on heat transfer coefficient is significant at low quality, with higher coefficient at higher heat flux. The suppression of the nucleate boiling heat transfer coefficient in this study was expressed by modifying the liquid Froude number, and also this modified liquid Froude number was used as the transition quality from stratified to unstratified flow.

(3) Compared with heat transfer coefficient of pure fluids comprising mixtures, heat transfer

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coefficients of mixtures are always low. This reduction is large in the low quality region. This is because nucleate boiling heat transfer is highly reduced for mixtures.

(4) The proposed correlation for pure refrigerants, based on the superposition model of Chen, is a function of the modified liquid Froude and boiling number. It predicted the present data with a mean deviation of 18%.

(5) For mixture, the contribution of convective evaporation, which is obtained for pure refrigerant under the suppression of nucleate boiling, is multiplied by the composition factor by Singal et al. This correlated the data with a mean deviation of 18.3%.

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