Performance Evaluation of Double-Tube Condenser using Smooth and Micro-Fin Tubes for Natural Mixture Refrigerant (Propane/Butane)

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Key words: Natural mixture refrigerant, Propane/Butane, HCFC22, Condensation, Smooth tube, Micro-fin tube

ABSTRACT: The investigation has been made into the prediction of heat exchange performance of a counter flow type double-tube condenser for natural refrigerant mixtures composed of Propane/n-Butane or Propane/i-Butane in a smooth tube and micro-fin tube. Under various heat transfer conditions, mass flux, pressure drop and heat transfer coefficient of the mixed refrigerants were calculated using a prediction method, when the length of condensing tube, total heat transfer rate, mass flux and outlet temperature of coolant were maintained constant. Also, the predicted results were compared with those of HCFC22. The results showed that the mixed refrigerants of Propane/n-Butane or Propane/i-Butane could be substituted for HCFC22, while the pressure drop and overall heat transfer coefficient of the refrigerants were evaluated together.

Nomenclature

C_P	: isobaric specific heat [J/kgK]
D	: inside diameter of the outer tube [m]
d	: diameter of the inner tube [m]
D_{12}	: coefficient of diffusion [m^2/s]
dP/dz	: static pressure [kPa/m]
G	: mass flux [kg/m^{2}s]
Ga	: Galileo number, $g\rho d/\mu$
h	: enthalpy [J/kg]
j	: diffusion mass flux of component [kg/m^{2}s]
K	: overall heat transfer coefficient [kW/m^{2}K]
$\dot{m}$	: total condensation mass flux [kg/m^{2}s]
$\dot{m}_1$	: condensation mass flux of more volatile component [kg/m^{2}s]
Nu	: Nusselt number, $\alpha d/\lambda$
P	: pressure [kPa]
Ph	: phase change number, $C_p(T_{sat} - T_w)/h$
Pr	: Prandtl number, $C_p\lambda/\mu$
q	: heat flux [W/m^2]
Q_T	: total heat transfer rate [kW]
Re	: Reynolds number, Gd/μ
Sc	: Schmidt number, $\mu/\rho D_{12}$
Sh	: Sherwood number, $\beta d/\rho D_{12}$
T	: temperature [K]
W	: mass flow rate [kg/s]
x	: vapor quality [-]
y_1	: mass fraction of more volatile component [kg/kg]
z	: refrigerant flow direction [m]

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Greek Symbols

α : heat transfer coefficient [W/m^{2}K]
β : mass transfer coefficient [kg/m^{2}s]
η_A : enlargement ratio of heat transfer area
λ : thermal conductivity [W/mK]
μ : dynamic viscosity [Pa·s]
ξ : ratio of $\dot{m}_1/\dot{m}$
ρ : density [kg/m^3]
Φ_V : two-phase multiplier
X_{tt} : Lockhart-Martinelli parameter
Ψ : void fraction

Subscripts

b : bulk
C : cooling water
F : forced convection
i : vapor-liquid interface
in : inlet
k : component k (k=1, 2)
L : liquid
out : outlet
r : refrigerant
V : vapor
w : wall of inner tube
wi : outside of the inner tube
wo : inside of the inner tube

1. Introduction

Recently, HFC's were developed as alternative refrigerants for CFC's and HCFC's in order to protect the ozone layer in the stratosphere. However, at the 1997 Kyoto Conference, it was determined that HFC's must be reduced due to their global warming potential. Today, natural working fluids such as HCs, Water, CO_2, NH_3, are attracting a great deal of attention all over the world. Natural refrigerants, such as propane, n-butane, i-butane, have been considered as alternatives for CFC12 and HCFC22 in vapor compression heat pump and refrigeration systems,[1-2] however, the concern is their flammability. Moreover, the research on the system in comparison with the characteristic of CFC12 or HCFC22 is also advancing recently using these results.[3-4]

In the present paper, two kinds of natural refrigerant mixtures of Propane/n-Butane and Propane/i-Butane are selected as candidates for alternatives, and their condensation characteristics in a counterflow double-tube condenser are predicted using a model proposed by authors.[5] The results obtained for the natural refrigerant mixtures are compared those of HCFC22 in a smooth tube and a micro-fin tube.

2. Prediction model

Figure 1 shows the physical model of a counter flow double-tube condenser. A smooth tube and a micro-fin tube are used as the inner tube of the condenser. The vapor of natural refrigerant mixtures composed of Propane/n-Butane or Propane/i-Butane flow into the inner tube, while the cooling water flows counter-currently in an annulus. The refrigerant vapor starts to condense at an axial position z=0. At an arbitrary position z in the two-phase region, the bulk vapor is represented by the thermodynamic state (P, T_{Vb}, h_{Vb}, y_{1Vb}, y_{2Vb}), the vapor-liquid interface is of state (P, T_i, y_{1Vi}, y_{2Vi}, y_{1Li}, y_{2Li}) and the bulk liquid is of state (P, T_{Lb}, h_{Lb}, y_{1Lb}, y_{2Lb}).

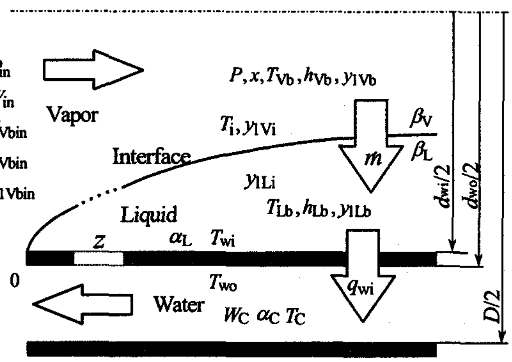

Fig. 1 Physical model.

Table 1 Correlation equation used in the prediction method

Smooth tube	Micro-fin tube
Correlation for frictional pressure drop $\Phi_V = 1 + 0.5\left[\dfrac{G_r}{\sqrt{g d_{wi} \rho_V (\rho_L - \rho_V)}}\right]^{0.75} X_{tt}^{0.35}$ where: $X_{tt} = \left(\dfrac{1-x}{x}\right)^{0.9}\left(\dfrac{\rho_V}{\rho_L}\right)^{0.5}\left(\dfrac{\mu_L}{\mu_V}\right)^{0.1}$	Correlation for frictional pressure drop $\Phi_V = 1.1 + 1.3\left[\dfrac{G_r X_{tt}}{\sqrt{g d_{wi} \rho_V (\rho_L - \rho_V)}}\right]^{0.35}$ where: $X_{tt} = \left(\dfrac{1-x}{x}\right)^{0.9}\left(\dfrac{\rho_V}{\rho_L}\right)^{0.5}\left(\dfrac{\mu_L}{\mu_V}\right)^{0.1}$
Correlation of liquid film heat transfer $\mathrm{Nu} \equiv \dfrac{\alpha_L d_{wi}}{\lambda_L} = \left(\mathrm{Nu}_F^2 + \mathrm{Nu}_B^2\right)^{1/2}$ where: $\mathrm{Nu}_F = 0.0152(1 + 0.6\mathrm{Pr}_L^{0.8})(\Phi_V / X_{tt})\mathrm{Re}_L^{0.77}$ $\mathrm{Nu}_B = 0.725 H(\Psi)\left(\dfrac{\mathrm{Ga}\mathrm{Pr}_L}{Ph}\right)^{1/4}$ $H(\Psi) = \Psi + [10\{(1-\Psi)^{0.1} - 1\} + 1.7\times 10^{-4}\mathrm{Re}]\sqrt{\Psi}(1 - \sqrt{\Psi})$	Correlation of liquid film heat transfer $\mathrm{Nu} \equiv \dfrac{\alpha_L d_{wi}}{\lambda_L} = \left(\mathrm{Nu}_F^2 + \mathrm{Nu}_B^2\right)^{1/2}$ where: $\mathrm{Nu}_F = 0.0152(3 + \mathrm{Pr}_L^{1.1})(\Phi_V / X_{tt})\mathrm{Re}_L^{0.68}$ $\mathrm{Nu}_B = \dfrac{0.725}{\eta_A^{1/4}} H(\Psi)\left(\dfrac{\mathrm{Ga}\mathrm{Pr}_L}{Ph}\right)^{1/4}$ $H(\Psi) = \Psi + \{10(1-\Psi)^{0.1} - 8.0\}\sqrt{\Psi}(1 - \sqrt{\Psi})$
Correlation of vapor mass transfer $\mathrm{Sh}_V \equiv \dfrac{\beta_V d_{wi}}{\rho_V D_{12}} = 0.023\sqrt{\Psi}\,\Phi_V^2 \mathrm{Re}_V^{0.8}\mathrm{Sc}_V^{1/3}$	
Correlation of Dittus-Boelter $\mathrm{Nu}_C \equiv \dfrac{\alpha_C (D - d_{wo})}{\lambda_C} = 0.023\mathrm{Re}_C^{0.8}\mathrm{Pr}_C^{0.4}$	

In the present prediction calculation, the following assumptions are employed:

(1) The phase equilibrium is only established at the vapor-liquid interface. The bulk vapor is in saturation, while the bulk liquid is sub-cooled.

(2) The frictional pressure drop is calculated using correlations shown in Table 1. These correlations were developed for the condensation of pure refrigerant in a horizontal smooth tube and a horizontal micro-fin tube.[6]

(3) The heat transfer coefficient of the liquid film is evaluated using correlations shown in Table 1. These correlations were developed for the condensation of pure refrigerant in a horizontal smooth tube and a horizontal micro-fin tube.[7-8]

(4) In the liquid film, the radial distribution of mass fraction is uniform, and the mass transfer coefficient is infinite.

(5) The vapor mass transfer coefficient of component is calculated by a correlation shown in Table 1. This correlation was derived from the correlation of the frictional pressure drop,[9] based on the Chilton-Colburn analogy.[10]

The governing equations to predict the heat exchange performance are as follows:

<u>Momentum balance of refrigerant</u>

$$\frac{dP}{dz} = -\left(\frac{4W_{in}}{\pi d_{wi}^2}\right)^2 \frac{d}{dz}\left[\frac{x^2}{\Psi\rho_V} + \frac{(1-x)^2}{(1-\Psi)\rho_L}\right] + \frac{dP_F}{dz} \tag{1}$$

where the void fraction Ψ is estimated from the Smith equation for two phase flow in a smooth tube,[11] and dP_F/dz is the frictional pressure gradient calculated using correlations in Table 1.

Heat balance of refrigerant

$$q_{wi} = -\frac{W_{in}}{\eta_A \pi d_{wi}} \frac{d}{dz}\{x h_{Vb} + (1-x) h_{Lb}\} = \alpha_L (T_i - T_{wi}) \quad (2)$$

where η_A is set to be equal to 1 when the case of smooth tube. The α_L is calculated using correlations in Table 1.

Mass balance of more volatile component in vapor core

$$\dot{m}_1 = -\frac{W_{in}}{\pi d_{wi}} \frac{d}{dz}(x y_{1Vb}) = -\frac{W_{in} y_{1Vi}}{\pi d_{wi}} \frac{dx}{dz} - \beta_V (y_{1Vi} - y_{1Vb}) \quad (3)$$

where the β_V is calculated using the correlation in Table 1.

Mass balance of more volatile component in liquid film

$$y_{1Lb} = y_{1Li} \quad (4)$$

Relation between vapor quality and mass fraction

$$x = \frac{y_{1Vbin} - y_{1Lb}}{y_{1Vb} - y_{1Lb}} \quad (5)$$

where y_{1Vbin} is the bulk mass fraction at the refrigerant inlet.

Radial wall heat conduction in inner tube

$$q_{wi} = \frac{2\lambda_w (T_{wi} - T_{wo})}{\eta_A d_{wi} \ln(d_{wo}/d_{wi})} \quad (6)$$

Heat balance of cooling water

$$q_{wi} = -\frac{W_C C_{PC}}{\eta_A \pi d_{wi}} \frac{dT_C}{dz} = \frac{\alpha_C d_{wo}}{\eta_A d_{wi}}(T_{wo} - T_C) \quad (7)$$

where α_C is calculated using the Dittus-Boelter equation.[12]

The prediction calculation is done for the natural refrigerant composed of Propane/n-Butane and Propane/i-Butane. In the calculation, the conditions of refrigerant and cooling water at the inlet of the double-tube condenser are specified as known parameters together with the dimensions of the condenser. Then, the local values of vapor quality, thermodynamic states of refrigerant bulk vapor, vapor-liquid interface and refrigerant bulk liquid, wall temperature, wall heat flux and cooling water temperature are obtained by solving equations of (1) to (7) simultaneously.

The dimension of the double tube condenser

Table 2 Dimension of condenser

		Smooth	Micro-fin
Inner tube	d_{wi} [m]	0.00637	0.0065
	η_A [-]	1	2.12
	d_{wo} [m]	0.007	0.007
	λ_w [W/mK]	385	385
Outer tube	D [m]	0.012	0.012
	Tube length [m]	3.0	3.0

Table 3 Condition of refrigerant and cooling water for smooth and micro-fin tube

Case	Refrigerant	y_{1Vbin}	G_C	T_{Cout}	Q_T	$(G_r)_{HCFC22}$ Smooth	$(G_r)_{HCFC22}$ Micro-fin
a	Propane/n-Butane	0-1	200	42.85	1.5	318	305
b	Propane/n-Butane	0-1	300	42.85	1.5	315	302
c	Propane/n-Butane	0-1	400	42.85	1.5	314	301
d	Propane/i-Butane	0-1	300	42.85	1.5	315	302

evaluated in the present study is shown in Table 2. The calculation condition for Propane/n-Butane and Propane/i-Butane is shown in Table 3. In each case the refrigerant vapor at the inlet is saturated, and the total condensation tube length, the total heat transfer rate, the mass flux of cooling water and outlet temperature of cooling water are given as constant values. The prediction calculation of HCFC22 is also done on the same condition as that for each case of Propane/Butane mixture.

Thermodynamic and transport properties of refrigerants are calculated using the program package REFPROP Ver. 6.0.[13]

3. Results and discussion

To verify the validity of prediction model of a counter flow double-tube condenser, simulation results were compared with experiment ones.[5] The simulation results predict the experimental data within an error of 25%. Thus it was confirmed the possibility to predict heat exchange performance of a counter flow type double-tube condenser for natural refrigerant mixtures using this prediction model.

Figures 2 and 3 show the relation between the mass flux of refrigerant G_r and the mass fraction of more volatile component at the refrigerant inlet y_{1Vbin} in smooth and micro-fin tubes, where symbols (○, △, □, ▽) represent the results for cases (a), (b), (c) and (d) in Table 3, respectively. In all cases, the G_r values of Propane/n-Butane and Propane/i-Butane are lower than that of HCFC22 shown in Table 3. This is mainly due to the difference in the latent heat of condensation. The G_r values in the micro-fin tube are a little lower than those of the smooth tube. The overall heat transfer coefficient of micro-fin tube is about two times higher than that of smooth tube. This results in lowering the refrigerant pressure and increasing the latent heat in the micro-fin tube. As a result, the G_r values in the micro-fin tube decrease. In all cases, the G_r values of Propane/n-Butan and Porpane/i-Butane are minimum in the range of y_{1Vbin} value, 0.2~0.4. It is affected by the change of the latent heat of condensation with the change of mass fraction of y_{1Vbin} in saturation pressure of each refrigerant.

Figures 4 and 5 show the relation between the total pressure drop of refrigerant ΔP_T over the 3 m length and the mass fraction of more volatile component at the refrigerant inlet y_{1Vbin} in the smooth and the micro-fin tubes, where symbol ▲ represent the result of HCFC22 for reference. The ΔP_T values of Propane/n-Butane

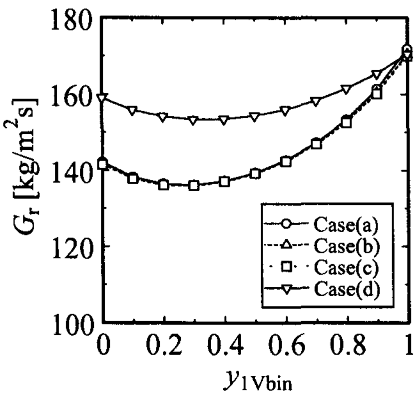

Fig. 2 Relation between G_r and y_{1Vbin} (smooth tube).

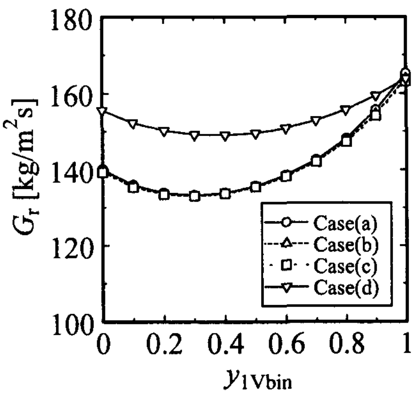

Fig. 3 Relation between G_r and y_{1Vbin} (micro-fin tube).

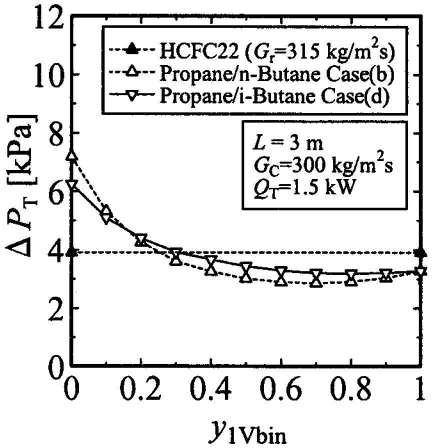

Fig. 4 Relation between ΔP_T and $y_{1\,Vbin}$ (smooth tube).

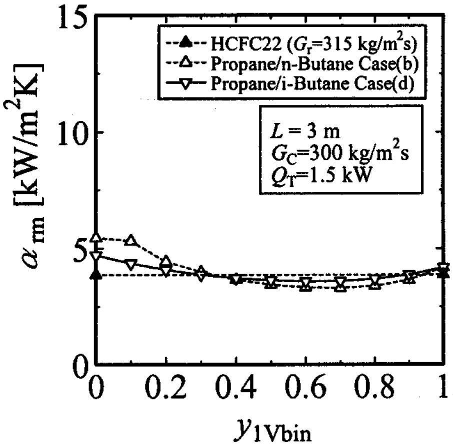

Fig. 6 Relation between α_{rm} and $y_{1\,Vbin}$ (smooth tube).

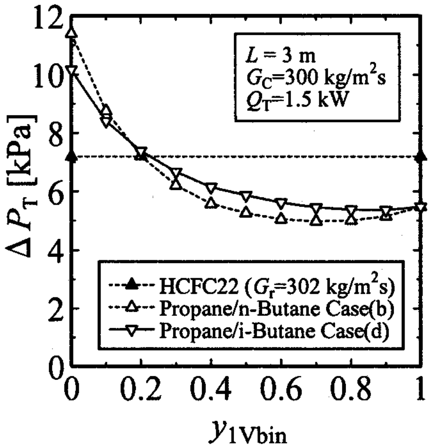

Fig. 5 Relation between ΔP_T and $y_{1\,Vbin}$ (micro-fin tube).

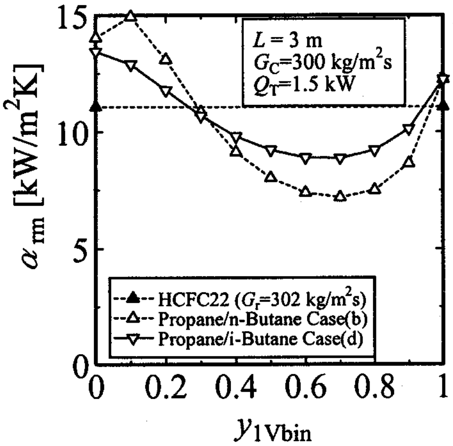

Fig. 7 Relation between α_{rm} and $y_{1\,Vbin}$ (micro-fin tube).

and Propane/i-Butane decrease with the increase of $y_{1\,Vbin}$. It is found that the pressure drop is decreased as liquid dynamic viscosity and liquid density is decreased with the increase of mass fraction. In addition, the ΔP_T values of Propane/n-Butane and Propane/i-Butane are changed at the $y_{1\,Vbin}$=0.2. The reason for this is that the value of liquid dynamic viscosity and liquid density of Propane/n-Butane is higher than that of Propane/i-Butane, while the value of mass flux of Propane/n-Butane is smaller than that of Propane/i-Butane. It is also found that the ΔP_T values of Propane/n-Butane and Propane/i-Butane are lower than that of HCFC22 when $y_{1\,Vbin}$ is larger than about 0.3 in the case of smooth tube, about 0.2 in the case of micro-fin tube.

Figures 6 and 7 show the average heat transfer coefficient α_{rm} of Propane/n-Butane, Propane/i-Butane and HCFC22 in the smooth and the micro-fin tubes, respectively. In the smooth tube, the α_{rm} values of Propane/n-Butane and Propane/i-Butane in the region of $y_{1\,Vbin} \leq 0.3$ or $y_{1\,Vbin} \geq 0.9$ is high than that of HCFC22, while in the micro-fin tube, the α_{rm} values of Propane/n-Butane and Propane/i-Butane in the region of $y_{1\,Vbin} \leq 0.3$ is high than that of HCFC22. In the case of the micro-fin tube, the average heat transfer coefficient of Propane/n-Butane reaches a maximum near $y_{1\,Vbin}=0.1$. The α_{rm} values of Propane/n-Butane and Propane/i-Butane are changed at the value of $y_{1\,Vbin}=0.3$. This reason for this is that the value of the latent heat of condensation of Propane/n-Butane is higher than that of Propane/i-Butane, while the value of mass diffusion resistance of Propane/n-Butane is smaller than that of Propane/i-Butane.

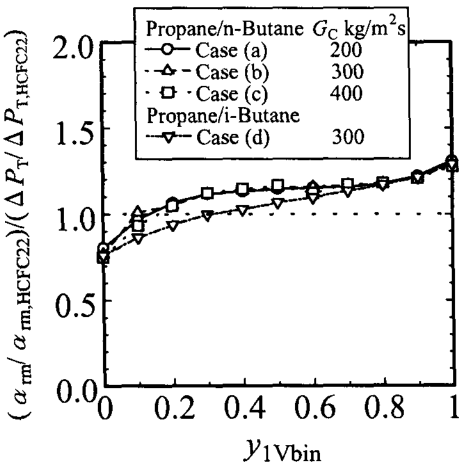

Fig. 8 Relation between $(\alpha_{rm}/\alpha_{rm,HCFC22})/(\Delta P_T/\Delta P_{T,HCFC22})$ and $y_{1\,Vbin}$ (smooth tube).

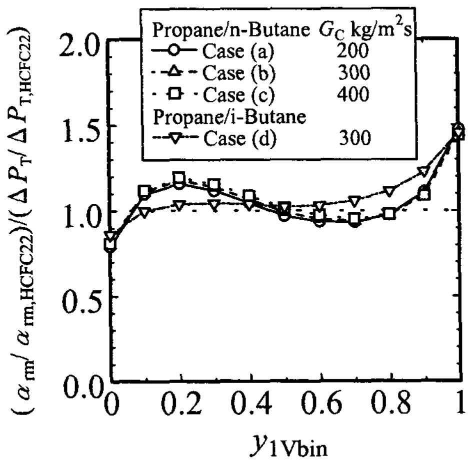

Fig. 9 Relation between $(\alpha_{rm}/\alpha_{rm,HCFC22})/(\Delta P_T/\Delta P_{T,HCFC22})$ and $y_{1\,Vbin}$ (micro-fin tube).

Figures 8 and 9 show the overall comparison of Propane/n-Butane, Propane/i-Butane compared to HCFC22 in the smooth and micro-fin tubes. In the smooth tube, the $(\alpha_{rm}/\alpha_{rm,HCFC22})/(\Delta P_T/\Delta P_{T,HCFC22})$ values of Propane/n-Butane, Propane/i-Butane increase with increase of $y_{1\,Vbin}$. While in the micro-fin tube, the $(\alpha_{rm}/\alpha_{rm,HCFC22})/(\Delta P_T/\Delta P_{T,HCFC22})$ values of Propane/n-Butane, Propane/i-Butane increase with increase of $y_{1\,Vbin}$ and reaches a maximum near $y_{1\,Vbin}=0.2$. Then it decreases again with increase of $y_{1\,Vbin}$. In the smooth tube, the $(\alpha_{rm}/\alpha_{rm,HCFC22})/(\Delta P_T/\Delta P_{T,HCFC22})$ values of Propane/n-Butane and Propane/i-Butane in the region of $y_{1\,Vbin} \geq 0.2$ is higher than that of HCFC22, while in the micro-fin tube, the $(\alpha_{rm}/\alpha_{rm,HCFC22})/(\Delta P_T/\Delta P_{T,HCFC22})$ values of Propane/n-Butane and Propane/i-Butane in the region of $0.1 \leq y_{1\,Vbin} \leq 0.4$ or $y_{1\,Vbin} \geq 0.9$ is higher than that of HCFC22.

Figures 10 and 11 show the dimensionless interface temperature $(T_i - T_{wi})/(T_{Vb} - T_{wi})$ using the conditions shown in Table 3 (b) and (d) for Propane/n-Butane and Propane/i-Butane, respectively. Dimensionless interface temperature between vapor and liquid is lowest at the beginning of condensation and increases during condensation. Because the mass diffusion resistance is the highest at the beginning of condensation and decreases with refrigerant flow. Dimensionless interface temperature approaches to unity in the finishing point of condensation. Because the thermal resistance of liquid film dominates in the finishing point of condensation. As the results shown in Figs. 6~9, Propane/n-Butane is larger than that of Propane/i-Butane for the latent heat of condensation, as shown in Figs. 2 and 3, while Propane/i-Butane is smaller than that of Propane/

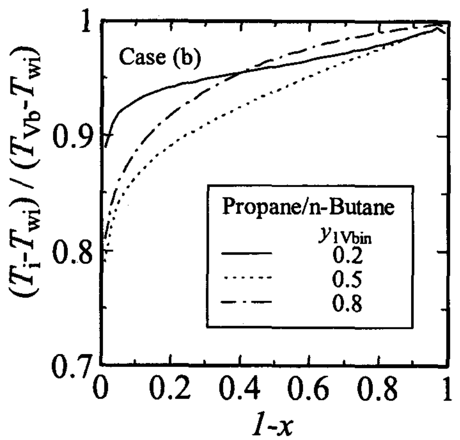

Fig. 10 Distribution of dimensionless interface temperature (Propane/n-Butane).

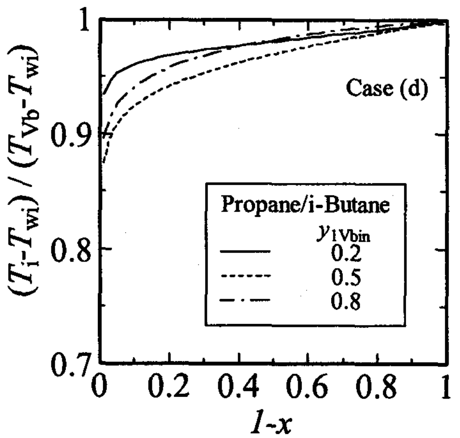

Fig. 11 Distribution of dimensionless interface temperature (Propane/i-Butane).

n-Butane for the heat transfer decrease which is due to mass diffusion resistance, as shown in Figs. 10 and 11.

4. Conclusions

The heat transfer and pressure drop characteristics of natural refrigerant mixtures condensing in a double-tube counter-flow heat exchanger are predicted on various conditions such as inlet mass fraction of refrigerant, mass flux of refrigerant and cooling water. The prediction results are compared with those of HCFC22. The results obtained are as follows:

(1) The mass flux values of Propane/n-Butane and Propane/i-Butane are 46% lower than those of HCFC22. And the mass flux values of micro-fin are 4% lower than those of smooth tube.

(2) The pressure drop values of Propane/n-Butane and Propane/i-Butane are lower than that of HCFC22 when mass fraction of more volatile component is larger than about 0.3 in the case of smooth tube, about 0.2 in the case of micro-fin tube.

(3) In the smooth tube, the average heat transfer coefficient values of Propane/n-Butane and Propane/i-Butane in the region of $y_{1Vbin} \leq 0.3$ or $y_{1Vbin} \geq 0.9$ is higher than that of HCFC22. But in the micro-fin tube, the average heat transfer coefficient values of Propane/n-Butane and Propane/i-Butane in the region of $y_{1Vbin} \leq 0.3$ is higher than that of HCFC22.

(4) In conclusion, natural refrigerant mixtures composed of Propane/n-Butane or Propane/i-Butane are appropriate candidates for alternative refrigerant from the viewpoint of heat transfer and pressure drop characteristics.

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