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CONDENSATION OF NATURAL REFRIGERANT MIXTURES IN HORIZONTAL SMOOTH TUBE - PREDICTION MODEL AND CALCULATION RESULTS -

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ABSTRACT

This paper deals with the heat exchange performance prediction of a counter flow double-tube condenser for binary natural refrigerant mixtures (propane/n-butane & propane/i-butane). The local characteristics of the heat transfer and pressure drop are calculated using a prediction method developed by authors, and the total pressure drop, the overall heat transfer coefficient and the average heat transfer coefficient are evaluated on various heat transfer conditions. It is confirmed that heat transfer performance of the natural refrigerant mixtures is higher than that of HCFC22 in most cases.

INTRODUCTION

According to the decision in COP3 (Kyoto, Japan, 1997), the hydrofluorocarbons will be limited in use for their global warming effect, although they have no ozone-depletion potential and have been already used as the alternatives in heat pump and refrigeration system. This requirement forces us to find some new alternatives that have not any ozone-depletion potential and relatively low global warming effect. In the present stage, the hydrocarbons, such as propane, n-butane, i-butane, have been considered as compatible materials [1].

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 counter flow double-tube condenser are predicted using a non-equilibrium model proposed by authors [2]. The prediction results of the natural refrigerant mixtures are compared with those of HCFC22.

PREDICTION MODEL

Figure 1 shows the physical model of a counter flow double-tube condenser. The refrigerant vapor mixture flowing into an inner horizontal smooth tube with a mass flow rate W_{v0} (mass velocity G_r) starts to condense at the axial position $z=0$. At an arbitrary position z in two phase region (vapor quality x), the refrigerant bulk vapor is represented by thermodynamic state $(P, T_{vb}, h_{vb}, y_{vb})$, the vapor-liquid interface is of state (P, T_i, y_{vi}, y_{Li}), and the refrigerant bulk liquid is of state $(P, T_{Lb}, h_{Lb}, y_{Lb})$, where P, T, h and y denote the pressure, the temperature, the enthalpy and the mass fraction of more volatile component, respectively. The cooling water flowing counter-currently in an outer annulus with mass flow rate W_c (mass velocity G_c) is of temperature T_c. Symbols T_{wi} and T_{wo} denote the inside and outside wall

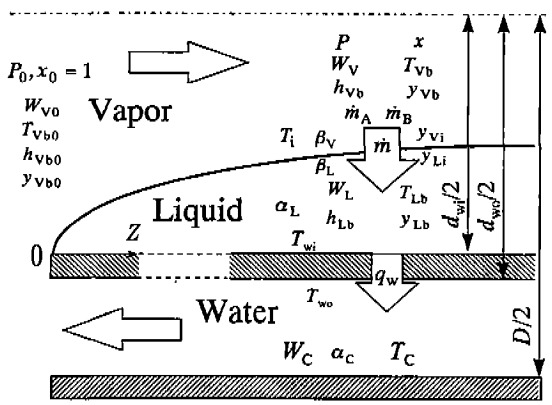


Figure 1 Physical model

temperature of the inner tube, respectively, and q_w is the wall heat flux based on the inside surface area of the inner tube. Symbols $\dot{m}$, $\dot{m}_A$ and $\dot{m}_B$ denote the total condensation mass flux and the condensation mass fluxes of more volatile and less volatile components, respectively. Symbols α_L and α_C are the heat transfer coefficients of refrigerant liquid and cooling water, respectively, and symbols β_V and β_L are the mass transfer coefficients of refrigerant in vapor and liquid phases, respectively.

The condensation characteristics of refrigerant mixture are calculated using a non-equilibrium model proposed by authors [2]. In this model 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 subcooled.

(2) The frictional pressure change is estimated by the correlation equation shown in Table 1. This equation was developed for the condensation of pure refrigerant in a horizontal smooth tube (Haraguchi et al. [3]). In the prediction of pressure drop the void fraction ψ is estimated by the Smith equation [4].

(3) The heat transfer coefficient of the liquid film is estimated from the correlation equation shown in Table 1. This equation was developed for the condensation of pure refrigerant in a horizontal smooth tube (Haraguchi et al. [5]).

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

(5) The mass transfer coefficient of vapor core is calculated by the correlation equation shown in Table 1. This equation is derived from the correlation equation of the frictional pressure drop, based on the Chilton-Colburn analogy.

The basic equations to predict the heat transfer performance of a double-tube counterflow condenser are summarized as:

(a) Momentum balance of refrigerant

$$\frac{dP}{dz} = -\frac{4W_{v0}}{\pi d_{wi}^2}\frac{d}{dz}\left[\frac{x^2}{\psi^2\rho_V} + \frac{(1-x)^2}{(1-\psi)^2\rho_L}\right] + \frac{dP_F}{dz} \qquad (1)$$

where the void fraction ψ and the frictional pressure change dP_F/dz are calculated using correlation equations in Table 1.

(b) Heat balance of refrigerant

$$q_w = -\frac{W_{v0}}{\pi d_{wi}}\frac{d}{dz}\{x h_{Vb} + (1-x)h_{Lb}\} = \alpha_L(T_i - T_{wi}) \qquad (2)$$

where the liquid film heat transfer coefficient α_L is calculated using the correlation equation in Table 1.

(c) Mass balance of more volatile component in vapor core

$$\dot{m}_A = -\frac{W_{v0}}{\pi d_{wi}}\frac{d}{dz}(x\, y_{Vb}) = -\frac{W_{v0}\, y_{Vi}}{\pi d_{wi}}\frac{dx}{dz} - \beta_V(y_{Vi} - y_{Vb}) \qquad (3)$$

where the vapor mass transfer coefficient β_V is calculated using the correlation equation in Table 1.

(d) Mass balance of more volatile component in liquid film

Table 1 Correlation equations used in prediction calculation

Correlation equation for frictional pressure drop [3]

$$\Phi_V \equiv \sqrt{\frac{dP_F/dz}{dP_V/dz}} = 1 + 0.5\left[\frac{G}{\sqrt{g d_{wi}\rho_V(\rho_L - \rho_V)}}\right]^{0.75} X_{tt}^{0.35}$$

where:

$$\frac{dP_V}{dz} = -\frac{0.092 G x^2}{d_{wi}\,\rho_V (G x d_{wi}/\mu_V)^{0.2}}$$

$$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_V}{\rho_L}\right)^{0.5}\left(\frac{\mu_L}{\mu_V}\right)^{0.1}$$

Correlation equation of void fraction [4]

$$\psi = \left[1 + \frac{\rho_V}{\rho_L}\left(\frac{1-x}{x}\right)\left(0.4 + 0.6\sqrt{\frac{\frac{\rho_L}{\rho_V} + 0.4\frac{1-x}{x}}{1 + 0.4\frac{1-x}{x}}}\right)\right]^{-1}$$

Correlation equation of liquid film heat transfer [5]

$$Nu \equiv \frac{\alpha_L d_{wi}}{\lambda_L} = \left(Nu_F^2 + Nu_B^2\right)^{1/2}$$

where:

$$Nu_F = 0.0152(1 + 0.6 Pr_L^{0.8})(\Phi_V / X_{tt}) Re_L^{0.77}$$

$$Nu_B = 0.725 H(\psi)\left(\frac{GaPr_L}{Ph}\right)^{\frac{1}{4}}$$

$$H(\psi) = \psi + \{10[(1-\psi)^{0.1} - 1] + 1.7\times10^{-4} Re\}\sqrt{\psi}(1 - \sqrt{\psi})$$

$$Re_L = \frac{G(1-x)d_{wi}}{\mu_L}, \quad Re = \frac{G d_{wi}}{\mu_L}$$

Correlation equation of vapor mass transfer

$$Sh_V \equiv \frac{\beta_V d_{wi}}{\rho_V D} = 0.023\sqrt{\psi}\,\Phi_V^2\, Re_V^{0.8}\, Sc_V^{1/3}$$

$$y_{Lb} = y_{Li} \tag{4}$$

(e) Relation between vapor quality and mass fraction

$$x = (y_{Vb0} - y_{Lb})/(y_{Vb} - y_{Lb}) \tag{5}$$

where y_{Vb0} is the bulk mass fraction of more volatile component at the refrigerant inlet.

(f) Radial wall heat conduction in the inner tube

$$q_w = \frac{2\lambda_w (T_{wi} - T_{wo})}{d_{wi}\, ln(d_{wo}/d_{wi})} \tag{6}$$

where λ_w is the thermal conductivity of the inner tube.

(g) Heat balance of cooling water

$$q_w = -\frac{W_C\, c_{pC}}{\pi\, d_{wi}} \frac{dT_C}{dz} = \frac{d_{wo}}{d_{wi}} \alpha_C (T_{wo} - T_C) \tag{7}$$

where the heat transfer coefficient of cooling water α_C is calculated using the Dittus-Boelter equation.

In the prediction calculation the local values of vapor quality, the 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 (1) to (7) when the conditions of refrigerant and cooling water at the inlet of the condenser are specified together with dimensions of the condenser. The dimension of the double-tube condenser evaluated in the present study is shown in Table 2. The inner tube is smooth copper one with 7.9 mm I. D. and 10.0 mm O. D., and the outer tube is 16.0 mm I. D. The total condensation tube length is 5.0 m. 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 L, the total heat transfer rate Q_T, the mass velocity of cooling water G_C and outlet temperature of cooling water T_{C0} 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. The mass velocity of HCFC22 is shown in Table 3 for reference. Thermodynamic and transport properties of the propane/ butane mixtures are calculated using the program package REFPROP Ver. 5.0 (Gallagher et al., [6]).

Table 2 Dimensions of condenser

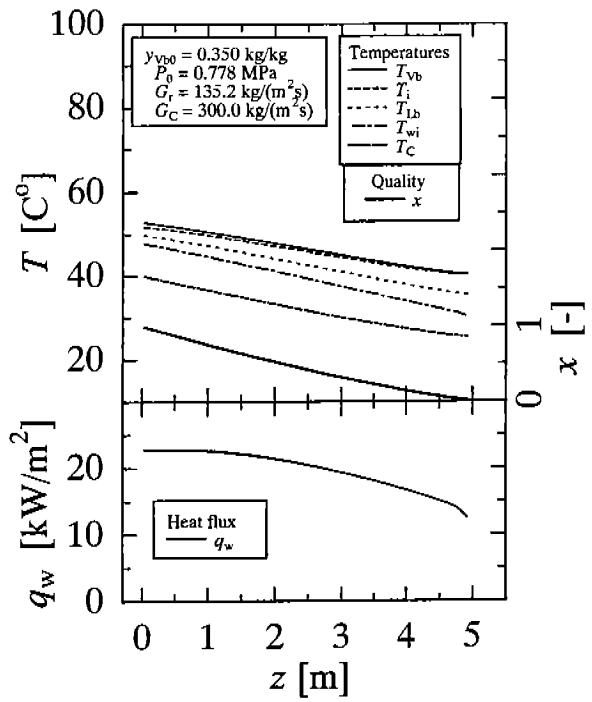
Inside diameter of inner tube	d_{wi} [m]	0.0079
Outside diameter of inner tube	d_{wo} [m]	0.010
Inside diameter of outer tube	D [m]	0.016
Total condensation tube length	L [m]	5.0

Table 3 Calculation condition for Propane/n-Butane and Propane/i-Butane

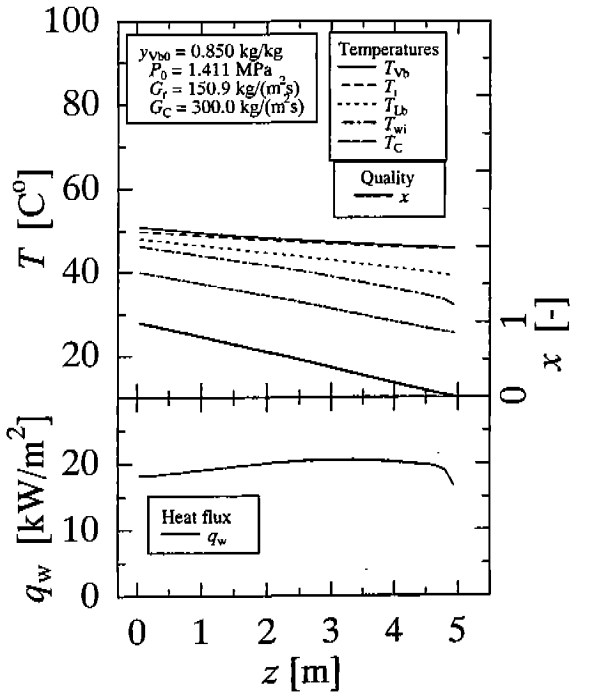
Case	Refrigerant	y_{Vb0} [kg/kg]	G_C [kg/(m²s)]	T_{C0} [℃]	Q_T [kW]	$(G_r)_{HCFC22}$ [kg/(m²s)]
(a-1)	Propane /n-Butane	0.0～1.0	300	40.0	2.431	300
(a-2)	Propane /n-Butane	0.0～1.0	300	40.0	2.912	360
(a-3)	Propane /n-Butane	0.0～1.0	200	40.0	2.441	300
(a-4)	Propane /n-Butane	0.0～1.0	400	40.0	2.422	300
(b-1)	Propane /i-Butane	0.0～1.0	300	40.0	2.431	300

RESULTS AND DISCUSSION

Figures 2 (a) and (b) show the prediction results of 35wt%propane/65wt%n-butane and 85wt%propane /15wt%n-butane, respectively, for Case (a-1). In both figures the values of T_{Vb}, T_i and T_{Lb} decrease in z-direction due to zeotropic characteristics of the mixtures. The temperature difference $(T_{Vb} - T_i)$ also decreases in z-direction. This reason is that the vapor diffusion mass flux of more volatile component at the vapor-liquid interface decreases in z-direction.



(a) y_{Vb0} =0.35 [kg/kg]



(b) y_{Vb0} =0.85 [kg/kg]

Figure 2 Prediction results of Propane/n-Butane mixtures (Case (a-1))

Figure 3 shows the relation between the mass velocity of refrigerant G_r and the mass fraction of more volatile component at the refrigerant inlet y_{Vb0}, where symbols ○, △, □, ▽ and ◇ represent the results for Case (a-1), (a-2), (a-3), (a-4) and (b-1), 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 and have the minimum near y_{Vb0} =0.25[kg/kg].

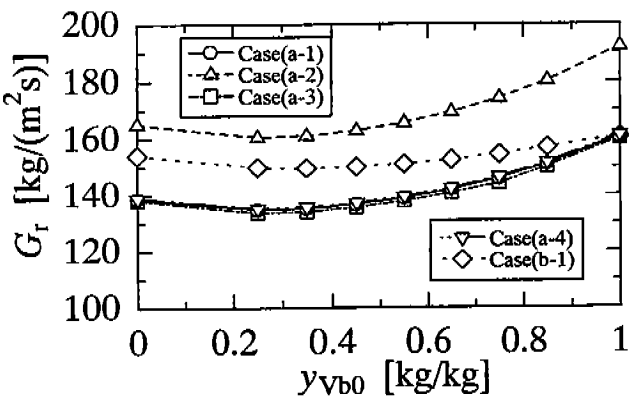


Figure 3 Relation between G_r and y_{Vb0}

Figure 4 shows the relation between the total pressure drop of refrigerant ΔP and the mass fraction of more volatile component at the refrigerant inlet y_{Vb0}, where symbols ● and △ represent the results for Case (a-1) and (b-1), respectively. In the figure the result of HCFC22 is also plotted using symbol ○ for reference. Both ΔP values of propane/n-butane and propane/i-butane decrease with increase of y_{Vb0}, and are almost the same when the value of y_{Vb0} is lager than about 0.25[kg/kg]. It is also found that the ΔP values of propane/n-butane and propane/i-butane are lower than that of HCFC22 when y_{Vb0} is larger than about 0.35[kg/kg].

Figure 5 shows the overall heat transfer coefficient of propane/n-butane and HCFC22. The overall heat transfer coefficient K_m is defined by

$$K_m = Q_T / (A_T \, \Delta T_{ln}) \tag{8}$$

where A_T is the total heat transfer area ($=\pi d_{wi} L$) and ΔT_{ln} is the logarithmic mean temperature. In all cases

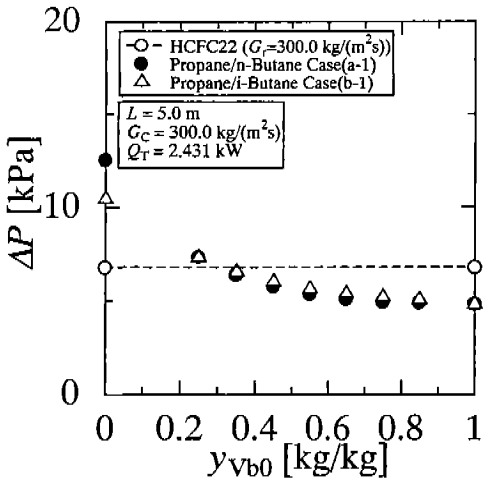


Figure 4 Relation between ΔP and y_{Vb0}

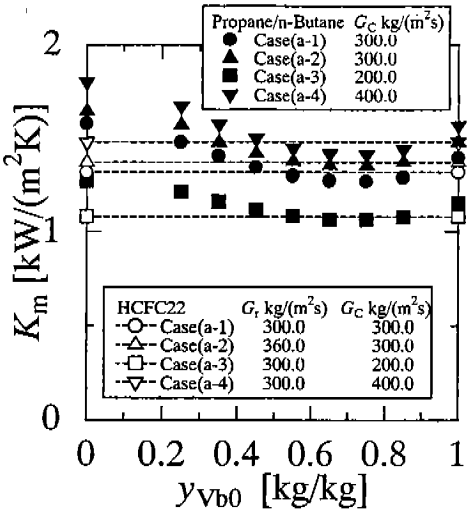


Figure 5 Relation between K_m and y_{Vb0}

the K_m value of propane/n-butane once decreases with increase of y_{Vb0} and reaches a minimum near y_{Vb0} =0.75 [kg/kg]. Then it increases again with increase of y_{Vb0}. The K_m value of propane/n-butane in the region of $y_{Vb0} \le 0.35$ [kg/kg] or $y_{Vb0} \ge 0.85$ [kg/kg] is higher than that of HCFC22. It is also confirmed that the K_m value of propane/i-butane in almost all region of y_{Vb0} is higher than that of HCFC22.

The average heat transfer coefficient of refrigerant in total condensation region α_{rm} is evaluated from the following equation.

$$\frac{1}{\alpha_{rm} d_{wi}} = \frac{1}{K_m d_{wi}} - \frac{1}{2\lambda_w} ln\left(\frac{d_{wo}}{d_{wi}}\right) - \frac{1}{\alpha_C d_{wo}} \qquad (9)$$

The calculation results of Case (a-1) and (b-1) are shown in Figure 5. In this figure the result of HCFC22 is also plotted by a dashed line with symbol ○. In both cases of propane/n-butane and propane/i-butane, the relation between α_{rm} and y_{Vb0} has almost the same trend as that of K_m. In the case of propane/n-butane the value of α_{rm} in the region of $y_{Vb0} \le 0.35$ [kg/kg] or $y_{Vb0} \ge 0.85$ [kg/kg] is higher than that of HCFC22, while in the case of propane/i-butane the value of α_{rm} in almost all region of y_{Vb0} is higher than that of HCFC22.

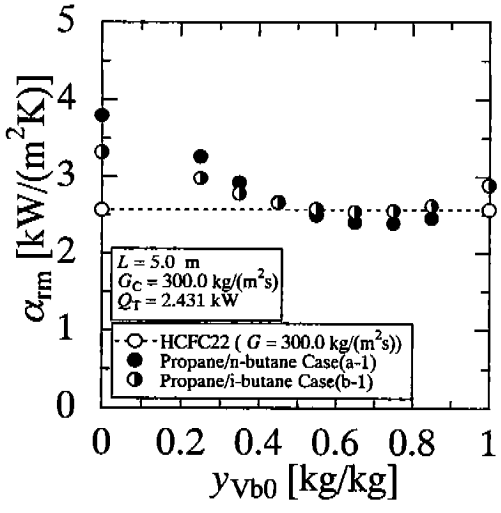


Figure 6 Relation between α_{rm} and y_{Vb0}

CONCLUSIONS

Using a non-equilibrium model proposed by authors, the heat transfer and pressure drop characteristics of natural refrigerant mixtures (propane/n-butane and propane/i-butane) condensing in a double-tube counterflow heat exchanger is predicted on various conditions such as inlet mass fraction of refrigerant, mass velocity of refrigerant and cooling water. The prediction results are compared with those of HCFC22.

(1) The values of mass velocity of propane/n-butane and propane/i-butane are lower than those of HCFC22 on

the same heat transfer condition. This reason is that the latent heat of propane/n-butane and propane/i-butane is higher than that of HCFC22.

(2) The values of total pressure drop of propane/n-butane and propane/i-butane are lower than that of HCFC22 when the inlet mass fraction of more volatile component is larger than about 0.35[kg/kg].

(3) The overall heat transfer coefficient of propane/n-butane in the region of $y_{vb0} \leq 0.35$ [kg/kg] or $y_{vb0} \geq 0.85$ [kg/kg] is higher than that of HCFC22, while that of propane/i-butane in almost all region of y_{vb0} is higher than that of HCFC22.

(4) The average heat transfer coefficient of propane/n-butane in the region of $y_{vb0} \leq 0.35$ [kg/kg] or $y_{vb0} \geq 0.85$ [kg/kg] is higher than that of HCFC22, while that of propane/i-butane in almost all region of y_{vb0} is higher than that of HCFC22.

The natural binary refrigerant mixtures (propane/n-butane & propane/i-butane) seem to be appropriate candidates for the alternative of HCFC22 from the viewpoint of thermal design.

REFERENCES

[1] Jung, D., Kim, C. B., Lim, B. H. and Lee, H. W., Testing of a Hydrocarbon Mixture in Domestic Refrigerators, *ASHRAE Trans.*, **102**-1, (1996), 1077-1084.

[2] Koyama, S., Yu, J. and Ishibashi, A., Condensation of Binary Refrigerant Mixtures in a Horizontal Smooth Tube, *Thermal Science and Engineering*, **6**-1 (1998), 123-129.

[3] Haraguchi, H., Koyama, S. and Fujii, T., Condensation of Refrigerants HCFC22, HFC134a and HCFC123 in a Horizontal Smooth Tube (1st Report, Proposal of Empirical Expressions for the Local Frictional Pressure Drop), *Trans. JSME*, (B), **60**-574 (1994), 239-244 (in Japanese).

[4] Smith, S. L., Void Fractions in Two-phase Flow: A Correlation based upon an Equal Velocity Head Model, *Heat and Fluid Flow*, **1**-1 (1971), 22-39.

[5] Haraguchi, H., Koyama, S. and Fujii, T., Condensation of Refrigerants HCFC22, HFC134a and HCFC123 in a Horizontal Smooth Tube (2nd Report, Proposal of Empirical Expressions for the Local Heat Transfer Coefficient), *Trans. JSME*, (B), **60**-574 (1994), 245-252 (in Japanese).

[6] Gallagher,J.S., McLinden,M., Morrison,G., and Huber, M., 1996, *NIST thermodynamic properties of refrigerants and Refrigerant mixtures database (REFPROP)*, Version 5.0. NIST Standard Reference Database 23.