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CONDENSATION OF BINARY ZEOTROPIC WORKING FLUID IN A PLATE HEAT EXCHANGER

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ABSTRACT This report deals with the condensation of binary zeotropic working fluids HCFC22/ HCFC123 in a plate heat exchanger. The correlations for vapor single-phase heat transfer, condensate heat transfer and vapor mass transfer are determined based on the experimental data of condensation of 100mol%HCFC123 and 20mol%HCFC22/80mol%HCFC123. By using these correlations, the prediction calculation for condensation of HCFC22/ HCFC123 mixtures is carried out in wide ranges of mole fraction and flow rate. Then, the characteristics of averaged heat and mass transfer for binary zeotropic working fluids are examined, and a simple prediction method of the averaged condensation characteristics is proposed.

Keywords : Plate Heat Exchanger, Condensation, Binary Zeotropic Working Fluid, Prediction Method

1. INTRODUCTION

The establishment of utilization technology of natural energy sources such as wind power, hydraulic power and geothermal power is one of important and urgent research subjects in order to reduce the discharge of carbon dioxide and restrain the global warming. Especially the development of new geothermal power generation systems using binary zeotropic working fluid is expected in Japan. To design these systems well, it is required to establish the prediction method of heat and mass transfer of binary zeotropic working fluid in heat exchanger. From this point of view, we have been investigating the condensation of binary zeotropic working fluid in a plate heat exchanger.

In the present paper, the prediction calculation of the condensation characteristics of binary zeotropic working fluids HCFC22/HCFC123 in a plate heat exchanger is carried out using the same prediction method that was developed for a plate-fin heat exchanger [1]. Then, a simple method to predict the averaged condensation characteristics of binary zeotropic working fluid in the plate heat exchanger is proposed.

2. PREDICTION METHOD OF LOCAL HEAT AND MASS TRANSFER CHARACTERISTICS

2.1 Basic Equations

Figure 1 shows the physical model to predict the condensation characteristics of binary zeotropic working fluid in a plate heat exchanger. In this figure, P is the pressure, T is the temperature, y is the mass fraction of more volatile component (HCFC22), x is the vapor quality, h is the specific enthalpy, q is the heat flux, W is the mass flow rate, α is the heat transfer coefficient, β is the mass transfer coefficient, λ is the thermal conductivity, δ is the wall thickness, $\dot{m}_A$ is the condensation mass flux of more volatile component (HCFC22), and $\dot{m}_B$ is the condensation mass flux of less volatile component (HCFC123). Subscripts denote as follows: V the vapor of working fluid, L the liquid film of working fluid, R the working fluid, W the cooling water, i the vapor-liquid interface, b the bulk of working fluid, *wall* the heat transfer surface. By employing the same assumptions as in the prediction method for a plate-fin heat exchanger [1], the following basic equations are derived for heat and mass transfer characteristics in two-phase region in the plate heat exchanger.

1) Heat balance of mixture working fluid

$$q_{wall} = -\frac{W_R}{l_R}\frac{d}{dz}\{x\, h_{Vb} + (1-x)h_{Lb}\} = \alpha_L (T_i - T_{wall\,R}) \quad (1)$$

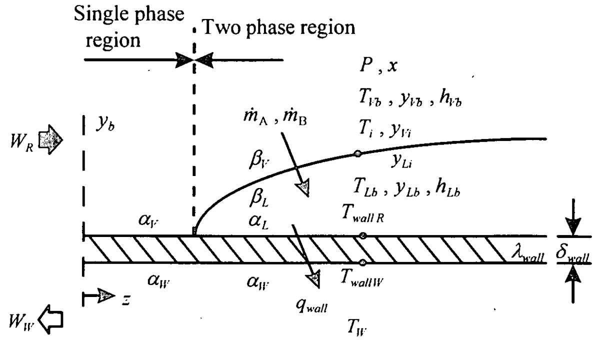


Fig. 1 Physical model

where l_R is the peripheral length of working fluid path.

2) Mass balance of more volatile component in vapor phase

$$\dot{m}_A = -\frac{W_R}{l_R}\frac{d}{dz}\left(x\, y_{Vb}\right) = -\frac{W_R\, y_{Vi}}{l_R}\frac{dx}{dz} - \beta_V\left(y_{Vi} - y_{Vb}\right) \quad (2)$$

3) Mass balance of more volatile component in liquid film

$$y_{Lb} = y_{Li} \quad (3)$$

4) Relation between vapor quality and mass fractions

$$x = \frac{y_b - y_{Lb}}{y_{Vb} - y_{Lb}} \quad (4)$$

where y_b is the bulk mass fraction at the working fluid inlet of the heat exchanger.

5) Heat conduction in the heat transfer wall

$$q_{wall} = \frac{\lambda_{wall}}{\delta_{wall}}\left(T_{wall\,R} - T_{wall\,W}\right) \quad (5)$$

6) Heat balance of cooling water

$$q_{wall} = -\frac{W_W\, c_{pW}}{l_W}\frac{dT_W}{dz} = \alpha_W\left(T_{wall\,W} - T_W\right) \quad (6)$$

where l_W is the peripheral length of cooling water path.

In the vapor single-phase region the heat transfer characteristics are determined by the following equation along with Eq. (5) and Eq. (6).

$$q_{wall} = -\frac{W_R}{l_R}\frac{dh_{Vb}}{dz} = \alpha_V\left(T_{Vb} - T_{wall\,R}\right) \quad (7)$$

2.2 Correlations of Local Heat and Mass Transfer

To solve the above mentioned equations, correlations of α_L, β_V, α_W and α_V should be given. Therefore, these correlations are determined using the following assumptions.

1) The condensate heat transfer characteristic is expressed by a correlation similar to the semi-empirical equation for free convection condensation of pure working fluid in a vertical smooth tube [2].
2) The vapor mass transfer characteristic is expressed by a functional form similar to the Colburn equation.
3) The cooling water heat transfer characteristic is expressed by a functional form similar to the Colburn equation.
4) The vapor single-phase heat transfer characteristic is calculated by a correlation, which is derived from the correlation of vapor mass transfer based on the Chilton-Colburn analogy.

In the concrete, first, the correlation of cooling water heat transfer is determined by the preliminary experiments on single-phase heat transfer. Next, the correlation of the condensate heat transfer characteristic is determined by trial prediction calculation using experimental data of 100mol% HCFC123. Then, the correlation of the vapor mass transfer is determined by trial prediction calculation using experimental data of the mixture 20mol%HCFC22/80mol% HCFC123. The correlation of vapor single-phase heat transfer is also determined. The correlations used in the present study are summarized in Table 1. For reference, the dimensions of the test heat exchanger are shown in Table 2.

Table 1 Correlations of heat and mass transfer

Vapor single phase heat transfer

$$\alpha_V = C_V\, 0.023 Re_V^{0.8} Pr_V^{1/3} \lambda_V / d_{hR}$$

where

$$C_V = 1.2\,,\quad Re_{Vd} = \frac{G_R\, x\, d_{hR}}{\mu_V}\,,\quad Pr_V = \frac{c_{pV}\, \mu_V}{\lambda_V}$$

Condensate heat transfer

$$\alpha_L = C_L\, Nu^*\, \lambda_L / d_{hR}$$

where

$$C_L = 1.055$$

$$Nu^* = \begin{cases} \left(\dfrac{4}{3}\dfrac{Ga_{Ld}}{Re_{Ld}}\right)^{1/3}\left\{1 + a\left(Re_{Ld} - 108\right)^m\left[1 - exp\left(\dfrac{-b}{Re_{Ld} - 108}\right)\right]^n\right\} & \left(Re_{Ld} > 108\right) \\ \left(\dfrac{4}{3}\dfrac{Ga_{Ld}}{Re_{Ld}}\right)^{1/3} & \left(Re_{Ld} \le 108\right) \end{cases}$$

$$a = 5.96\times10^{-5} Pr_L^{1.21},\; b = 428 Pr_L^{-0.045},$$

$$m = 1.74 - 0.3 Pr_L^{0.29},\; n = 1.25 - 0.349 Pr_L^{0.29}$$

$$Re_{Ld} = \frac{G_R(1-x)d_{hR}}{\mu_L}\,,\quad Ga_{Ld} = \frac{g\,\rho_L^2\, d_{hR}^3}{\mu_L^2}\,,\quad Pr_L = \frac{c_{pL}\,\mu_L}{\lambda_L}$$

Mass transfer of vapor core in two phase region

$$\beta_V = C_V\, 0.023 Re_V^{0.8} Sc_V^{1/3} \rho_V D_V / d_{hR}$$

where

$$Sc_V = \frac{\mu_V}{\rho_V\, D_V}$$

Heat transfer of cooling water

$$\alpha_W = 0.0345 Re_W^{0.8} Pr_W^{1/3} \lambda_W / d_{hW}$$

where

$$Re_{Wd} = \frac{G_W\, d_{hW}}{\mu_W}\,,\quad Pr_W = \frac{c_{pW}\,\mu_W}{\lambda_W}$$

Table 2 Dimensions of test heat exchanger

	Working fluid path	Cooling water path
Number of layer	2	1
Wall thickness [m]	6.0×10^{-4}	6.0×10^{-4}
Hydraulic diameter [m]	6.73×10^{-3}	8.24×10^{-3}
Plate width [m]	0.1	0.1
Effective heat transfer length [m]	0.97	0.97
Cross-section area [m^2]	7.39×10^{-4}	4.77×10^{-4}
Total heat transfer area [m^2]	0.2134	0.2134
Peripheral length [m]	0.22	0.22

2.3 Prediction Calculation Conditions

As the experiment of binary zeotropic working fluids was only carried out for about 20mol%HCFC22 mixture, the effects of the mole fraction, pressure and flow rate of working fluid on the heat transfer characteristics are examined by the prediction calculation. The calculation conditions are as follows,

1) cooling water temperature at outlet of heat exchanger,

$T_{Wout} = 30\ [^\circ C]$

2) flow rate of cooling water, $W_W = 398$ [kg/h]
3) working fluid super-heat at inlet of heat exchanger, $T_{bin} - T_{Rdew} = 20$ [K]
4) temperature difference between dew point of working fluid and cooling water at inlet of working fluid, $\Delta T_{WD} = T_{Rdew} - T_{Wout} = 5$, 10 and 20 [K]
5) flow rate of working fluid, $W_R = 35$, 55 and 75 [kg/h]
6) mole fraction of more volatile component (HCFC22), $y_b = 0$, 5, 10, 20, 30 and 50 [mol%]

By giving the above conditions as known parameters, the basic equations from Eq. (1) to Eq. (7) are solved numerically from the inlet of working fluid toward the outlet. Each calculation is terminated when one of the following conditions is satisfied,
1) the value of vapor quality at the outlet of the heat exchanger x_{out} is equal to 0.2
2) the value of temperature difference between working fluid bulk and cooling water in the heat exchanger $T_b - T_{Win}$ is equal to or less than 0.5 [K].

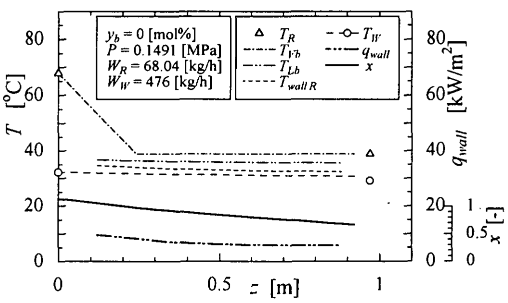
In the above prediction calculations and trial prediction calculations to determine correlations of local heat and mass transfer, thermodynamic and thermophysical properties of HCFC22/ HCFC123 mixtures are calculated using PROPATH [3], and the properties of cooling water are calculated using reference [4].

3. RESULTS AND DISCUSSION

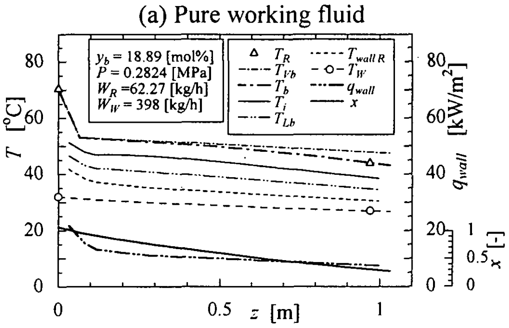
3.1 Distribution of Temperature, Heat Flux and Vapor Quality in Heat Exchanger

Figures 2 shows comparison between experimental and predicted results, where the prediction calculations in figures (a) and (b) were carried out to determine the correction factors of the condensate heat transfer and the vapor heat and mass transfer, C_L and C_V, in correlations shown in Table 1. In these figures symbols of open triangle and circle denote the measured temperature of working fluid, T_R, and cooling water, T_W, at inlet and outlet of the heat exchanger, while predicted results are represented as follows. The chain-dotted line and chain-double-dotted lines denote vapor bulk, T_{Vb}, and liquid bulk, T_{Lb}, for working fluid, respectively. The dotted and broken lines denote wall of working fluid side, $T_{wall\,R}$, and cooling water, T_W, respectively. The chain-double-dotted (thick) and solid (thick) lines denote heat flux, q_{wall}, and vapor quality, x, respectively. The chain-dotted (thick) and solid (thick) lines denote bulk, T_b, and vapor-liquid interface, T_i, of working fluid, respectively. The predicted values of T_R and T_W agree well with experimental ones and heat transfer length almost equal to 0.97 [m], when using correlations in Table 1.

Figure 3 shows the temperature, heat flux and vapor quality profiles in the heat exchanger on conditions of W_R =75[kg/h] and ΔT_{WD} =20[K]. Figures (a), (b) and (c) are the results for 0 (pure HCFC123), 20 and 50 mol%HCFC22, respectively. In these figures the lines are same as in Figure

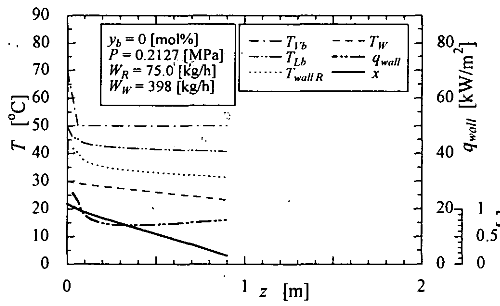


(a) Pure working fluid

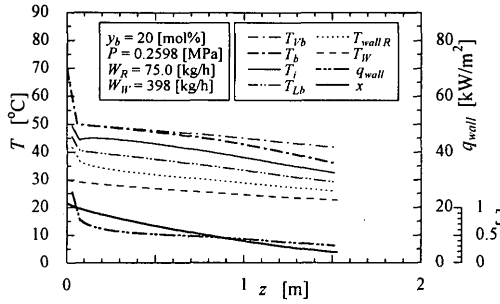


(b) Mixed working fluid

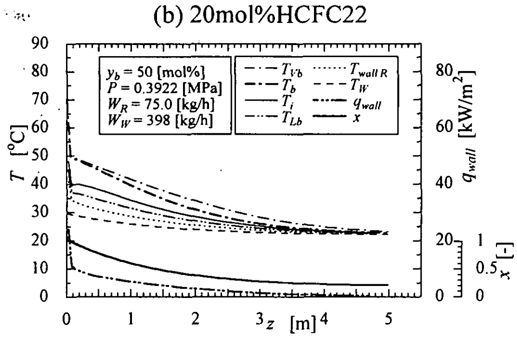
Fig.2 Results of prediction calculation



(a) 0mol%HCFC22



(b) 20mol%HCFC22



(c) 50mol%HCFC22

Fig.3 Temperature, heat flux and quality profiles in heat exchanger

2. The heat transfer length z, where the condition of x_{out} = 0.2 or $T_b - T_{Win} \approx 0.5$[K] is satisfied, increases with increase of y_b. This is caused by the following reasons. 1) The temperature glide of mixture in condensation process increases with increase of y_b on the condition that ΔT_{WD} is kept at a constant value. 2) The mass transfer resistance in vapor core increases with increase of y_b.

3.2 Comparison between Pure and Mixed Working Fluid on Averaged condensation Heat Transfer Characteristics

First, the overall heat transfer coefficient K_{m} defined by Eq. (8) is calculated from the results of the prediction calculations,

$$K_{\mathrm{m}} = q_{wall\,\mathrm{m}} / \Delta T_{ln} \tag{8}$$

where the averaged wall heat flux $q_{wall\,\mathrm{m}}$ and the logarithmic mean temperature difference ΔT_{ln} are defined as,

$$q_{wall\,\mathrm{m}} = W_R \left(h_{bin} - h_{bout} \right) / A_T \tag{9}$$

$$\Delta T_{ln} = \frac{\left(T_{Rdew} - T_{Wout}\right) - \left(T_{bout} - T_{Win}\right)}{ln\left(\dfrac{T_{Rdew} - T_{Wout}}{T_{bout} - T_{Win}}\right)} \tag{10}$$

Next, the averaged condensation heat transfer coefficient $\alpha_{R\mathrm{m}}$ is evaluated by

$$\frac{1}{\alpha_{R\mathrm{m}}} = \frac{1}{K_{\mathrm{m}}} - \frac{\delta_{wall}}{\lambda_{wall}} - \frac{1}{\alpha_W} \tag{11}$$

where α_W is estimated by the correlation shown in Table 1. Then, the averaged Nusselt number $Nu_{R\mathrm{m}}$ is obtained by

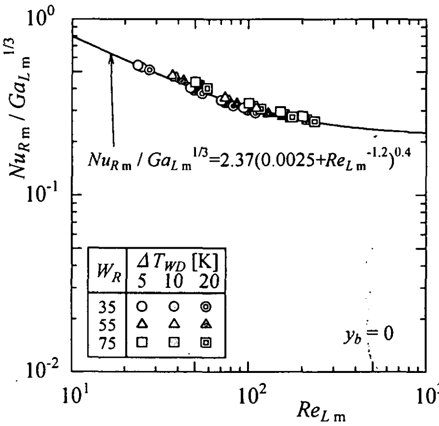
$$Nu_{R\mathrm{m}} = \frac{\alpha_{R\mathrm{m}}\, d_{hR}}{\lambda_L} \tag{12}$$

where thermophysical properties of liquid are calculated at the representative thermodynamic state of P, $y_{Lb\mathrm{m}}$ and $T_{b\mathrm{m}} = \left(T_{Rdew} + T_{bout}\right)/2$. The Reynolds number $Re_{L\mathrm{m}}$ and the Galileo number $Ga_{L\mathrm{m}}$ are also calculated as,

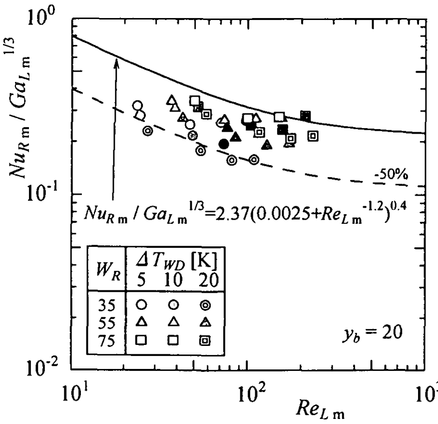
$$Re_{L\mathrm{m}} = G_R \left(\frac{\left(1 - x_{in}\right) + \left(1 - x_{out}\right)}{2} \right) d_{hR} \Big/ \mu_L \tag{13}$$

$$Ga_{L\mathrm{m}} = g\, \rho_L^2\, d_{hR}^3 / \mu_L^2 \tag{14}$$

Figure 4 (a) and (b) show the relation between $Nu_{R\mathrm{m}}/Ga_{L\mathrm{m}}^{1/3}$ and $Re_{L\mathrm{m}}$ in the cases of y_b= 0 and 20 mol%HCFC22, respectively. In these figures the symbols of circle, triangle and square represent the results of W_R = 35, 55 and 75 [kg/h], respectively, where the open, closed and double symbols represent the results of ΔT_{WD} = 5, 10 and 20 [K], respectively. Each symbol plotted in figure corresponding to one of the results of the different outlet vapor quality such as x_{out} = 0.8, 0.6, 0.4 and 0.2. In figure (a) for pure working fluid, the value of $Nu_{R\mathrm{m}}/Ga_{L\mathrm{m}}^{1/3}$ is not affected by ΔT_{WD} and W_R variation, but affected by $Re_{L\mathrm{m}}$. As a result, the averaged Nusselt number of pure working fluid is correlated well by



(a) 0mol%HCFC22 (HCFC123)



(b) 20mol%HCFC22

Fig. 4 Relation between $Nu_{R\mathrm{m}}/Ga_{L\mathrm{m}}^{1/3}$ and $Re_{L\mathrm{m}}$

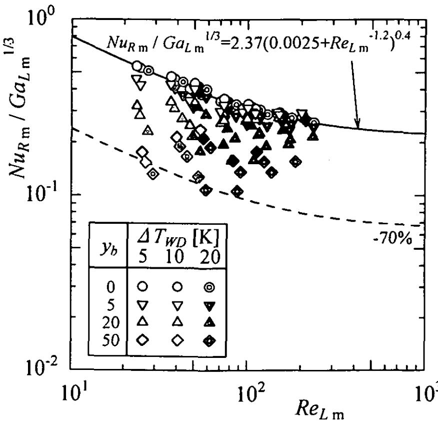


Fig.5 Effect of mole fraction on $Nu_{R\mathrm{m}}/Ga_{L\mathrm{m}}^{1/3}$

$$Nu_{R\mathrm{m}} = 2.37\, Ga_{L\mathrm{m}}^{1/3} \left(0.0025 + Re_{L\mathrm{m}}^{-1.2}\right)^{0.4} \tag{15}$$

In figure (b) for mixed working fluids, on the other hand, the value of $Nu_{R\mathrm{m}}/Ga_{L\mathrm{m}}^{1/3}$ is affected not only by $Re_{L\mathrm{m}}$, but also by W_R and ΔT_{WD}, and smaller than the predicted value with Eq. (15), which is represented by a solid line. It is also found that the value of $Nu_{R\mathrm{m}}/Ga_{L\mathrm{m}}^{1/3}$ approaches the solid line with increase of W_R and decrease of ΔT_{WD}; it is inferred that this trend corresponds to the decrease of the mass transfer resistance in vapor core.

Figure 5 shows the relation between $Nu_{R\mathrm{m}}/Ga_{L\mathrm{m}}^{1/3}$

and $Re_{L\mathrm{m}}$ to examine effects of y_b and ΔT_{WD} on the averaged condensation heat transfer characteristics. In the figure the symbols of circle, inverted triangle, triangle and diamond represent the results of y_b= 0, 5, 20 and 50 mol%HCFC22, respectively, and the solid line represents Eq. (15). The value of $Nu_{R\mathrm{m}}/Ga_{L\mathrm{m}}^{1/3}$ for mixture working fluids decreases with increase of y_b, and approaches the solid line with decrease of ΔT_{WD}.

3.3 Simple Prediction Method of Averaged Heat and Mass Transfer Characteristics for Mixture

In order to predict the performance of power generation systems using binary zeotropic working fluid accurately, modeling of momentum, heat and mass transfer characteristics in all components should be established. In this point of view, the present prediction method of local heat and mass transfer characteristics of binary mixture condensing in plate heat exchanger is effective. However, to reduce the computing time, it is necessary to develop a simple prediction method of the heat exchanger performance. Therefore, a simple method to predict the averaged condensation characteristics of binary zeotropic working fluid is examined based on the prediction results of the local condensation characteristics. In the prediction, the following assumptions are employed:

1) The condensate heat transfer characteristics for mixture working fluids are similar to pure working fluid. That is, the averaged Nusselt number is calculated by the following equation,

$$Nu_{L\mathrm{m}} \equiv \frac{q_{wall\,\mathrm{m}}\, d_{hR}}{\left(T_{i\mathrm{m}} - T_{wallR\,\mathrm{m}}\right)\lambda_L} = 2.37\, Ga_{L\mathrm{m}}^{1/3}\left(0.0025 + Re_{L\mathrm{m}}^{-1.2}\right)^{0.4} \qquad (16)$$

where $T_{i\mathrm{m}}$ is the representative temperature at vapor-liquid interface.

2) The averaged Sherwood number in vapor core is expressed as,

$$Sh_{V\mathrm{m}} = C \times 0.023\, Re_{V\mathrm{m}}^{0.8}\, Sc_{V\mathrm{m}}^{1/3} = \frac{\beta_{V\mathrm{m}}\, d_{hR}}{\rho_V\, D_V} \qquad (17)$$

where $Re_{V\mathrm{m}} = \dfrac{G_R\left(\dfrac{x_{in} + x_{out}}{2}\right) d_{hR}}{\mu_V}$, $Sc_{V\mathrm{m}} = \dfrac{\mu_V}{\rho_V\, D_V}$

and the optimum value of correction factor C is determined by the results of this prediction calculation.

The prediction procedure is as follows:

1) The following parameters are given as known quantities: total heat transfer area A_T, pressure of working fluid P, flow rate of working fluid W_R, mass fraction of more volatile component in working fluid y_b, bulk working fluid temperatures at inlet and outlet of heat exchanger T_{bin} & T_{bout}.
2) Calculate thermophysical properties of working fluid at inlet and outlet of heat exchanger.
3) Calculate averaged heat flux and averaged condensation mass fluxes as,

$$q_{wall\,\mathrm{m}} = W_R\left(h_{bin} - h_{bout}\right)/A_T \qquad (18)$$

$$\dot{m}_{\mathrm{m}} = W_R\left(x_{in} - x_{out}\right)/A_T \qquad (19)$$

$$\dot{m}_{A\mathrm{m}} = W_R\left(x_{in}\, y_{Vbin} - x_{out}\, y_{Vbout}\right)/A_T \qquad (20)$$

4) Calculate the value of $Sh_{V\mathrm{m}}$ from Eq. (17), then obtain the averaged mass transfer coefficient $\beta_{V\mathrm{m}}$ as,

$$\beta_{V\mathrm{m}} = Sh_{V\mathrm{m}}\, \rho_V\, D_V / d_{hR} \qquad (21)$$

5) Assume the representative temperature at vapor-liquid interface $T_{i\mathrm{m}}$
6) Calculate the averaged mass transfer coefficient $\left(\beta_{V\mathrm{m}}\right)_i$ obtained from total mass balance of more volatile component as,

$$\left(\beta_{V\mathrm{m}}\right)_i = \frac{\dot{m}_{\mathrm{m}}\, y_{Vi\mathrm{m}} - \dot{m}_{A\mathrm{m}}}{y_{Vi\mathrm{m}} - y_{Vb\mathrm{m}}} = \dot{m}_{\mathrm{m}}\left(\frac{y_{Vi\mathrm{m}} - \xi_{A\mathrm{m}}}{y_{Vi\mathrm{m}} - y_{Vb\mathrm{m}}}\right) \qquad (22)$$

where $\xi_{A\mathrm{m}} = \dot{m}_{A\mathrm{m}}/\dot{m}_{\mathrm{m}}$

7) Repeat the calculation from procedure 5) to procedure 6), by modifying $T_{i\mathrm{m}}$ until the following convergence criterion is satisfied within a convergence radius.

$$\left(\beta_{V\mathrm{m}}\right)_i = \beta_{V\mathrm{m}}$$

8) Calculate the value of $Nu_{L\mathrm{m}}$ from Eq. (16), then calculate the representative wall temperature $T_{wallR\,\mathrm{m}}$ as,

$$T_{wallR\,\mathrm{m}} = T_{i\mathrm{m}} - \frac{q_{wall\,\mathrm{m}}\, d_{hR}}{Nu_{L\mathrm{m}}\, \lambda_L} \qquad (23)$$

9) Calculate the averaged heat transfer coefficient $\left(\alpha_{R\mathrm{m}}\right)_{cor}$ and the averaged Nusselt number $\left(Nu_{R\mathrm{m}}\right)_{cor}$ defined as,

$$\left(\alpha_{R\mathrm{m}}\right)_{cor} = \frac{q_{wallR\,\mathrm{m}}}{T_{b\mathrm{m}} - T_{wallR\,\mathrm{m}}} \qquad (24)$$

$$\left(Nu_{R\mathrm{m}}\right)_{cor} = \frac{\left(\alpha_{R\mathrm{m}}\right)_{cor}\, d_{hR}}{\lambda_L} \qquad (25)$$

where it should be noted that $\left(\alpha_{R\mathrm{m}}\right)_{cor}$ and $\left(Nu_{R\mathrm{m}}\right)_{cor}$ comprise both effects of the condensate heat transfer and the vapor mass transfer.

Figure 6 shows a comparison between $\left(Nu_{R\mathrm{m}}\right)_{exp}$ and $\left(Nu_{R\mathrm{m}}\right)_{cor}$. The values of $\left(Nu_{R\mathrm{m}}\right)_{cor}$ defined by Eq. (25) and the values of $\left(Nu_{R\mathrm{m}}\right)_{exp}$ defined by Eq. (12) are calculated based on the experimental results. In this figure, symbols of square, diamond, circle and triangle denote the value of C =1.5, 1.8, 2.0 and 3.0, respectively. The value of $\left(Nu_{R\mathrm{m}}\right)_{cor}$ increases with increase of the value of C. As a result, it is found that an optimum value exists around 2.0.

Figure 7 shows a comparison between $\left(Nu_{R\mathrm{m}}\right)_{sim}$ and $\left(Nu_{R\mathrm{m}}\right)_{cor}$ in the case of C = 1.8. The values of $\left(Nu_{R\mathrm{m}}\right)_{cor}$ defined by Eq. (25) and the values of $\left(Nu_{R\mathrm{m}}\right)_{sim}$ defined by Eq. (12) are calculated based on the prediction results of local condensation characteristics. In this figure, symbols of inverted triangle, square, triangle, pentagon and diamond denote y_b = 5, 10, 20, 30 and 50 mol%HCFC22, respectively, where the open, closed and double symbols are the same as in Figure 5. The difference between the values of $\left(Nu_{R\mathrm{m}}\right)_{sim}$ and $\left(Nu_{R\mathrm{m}}\right)_{cor}$ are within $\pm$30%. This also proves that the optimum value of C is about 1.8.

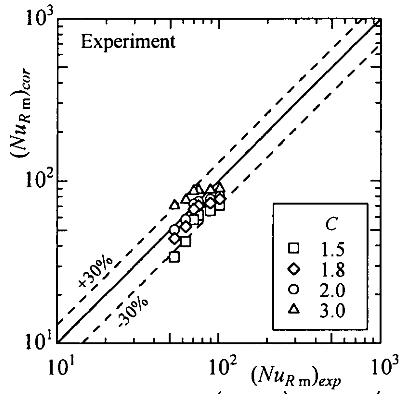


Fig.6 Comparison between $(Nu_{R\mathrm{m}})_{exp}$ and $(Nu_{R\mathrm{m}})_{cor}$

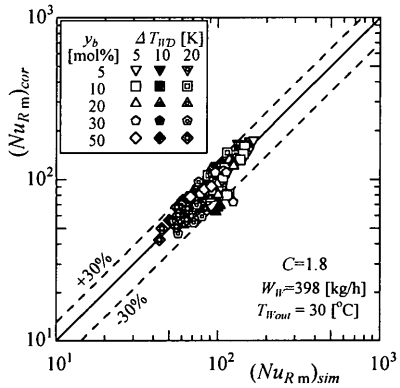


Fig.7 Comparison between $(Nu_{R\mathrm{m}})_{sim}$ and $(Nu_{R\mathrm{m}})_{cor}$

4. CONCLUSION

Prediction calculation of the condensation of binary zeotropic working fluid HCFC22/HCFC123 in a plate heat exchanger is carried out on various conditions, and the following results are obtained:

1) Correlations of local heat and mass transfer characteristics are determined based on the experimental results of pure and mixed working fluids.
2) The correlation of the averaged condensate heat transfer characteristics is proposed.
3) The simple method to predict the averaged heat and mass transfer characteristics is proposed.

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NOMENCLATURE

A_T : total heat transfer area [m^2]
C : correction factor [-]
c_p : isobaric specific heat [J / kg K]
D : diffusion coefficient [m^2 / s]
d_h : hydraulic diameter [m]
G : mass velocity [kg / m^2 s]
Ga : Galileo number [-]
g : gravitational acceleration [m / s^2]
h : specific enthalpy [J / kg]
K : overall heat transfer coefficient [W / m^2 K]
l : peripheral length [m]
$\dot{m}$: condensation mass flux [kg / m^2 s]
$Nu_{L\mathrm{m}}$: Nusselt number defined by T_i [-]
$Nu_{R\mathrm{m}}$: Nusselt number defined by T_b [-]
P : pressure [Pa] or [MPa]
Pr : Prandtl number [-]
q : heat flux [W / m^2]
Re : Reynolds number [-]
Sc : Schmidt number [-]
Sh : Sherwood number [-]
T : temperature [K]
W : flow rate [kg / s] or [kg / h]
x : vapor quality [-]
y : mole or mass fraction of more volatile component [mol%] or [wt%]
z : distance from inlet of working fluid [m]
α : heat transfer coefficient [W / m^2 K]
β : mass transfer coefficient [kg / m^2 K]
ΔT_{ln} : logarithmic mean temperature difference [K]
ΔT_{WD} : temperature difference between dew point and cooling water at inlet of working fluid [K]
δ : wall thickness [m]
λ : thermal conductivity [W / m K]
μ : viscosity [Pa s]
ρ : density [kg / m^3]
ξ_{A} : the ratio of more volatile component to total condensation mass fluxes [-]

Subscripts

A : more volatile component
B : less volatile component
b : bulk
cor : simple prediction method
dew : dew point
exp : experiment
i : vapor liquid interface
in : inlet
L : liquid
m : mean
out : outlet
R : working fluid
sim : local prediction method
V : vapor
W : cooling water
wall : heat transfer surface