Enhancement Effect of a Microfin Tube in Condensation of R407C

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ENHANCEMENT EFFECT OF A MICROFIN TUBE IN CONDENSATION OF R407C

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ABSTRACT

In the present paper the heat exchange performance of a counter flow double-tube condenser for a ternary refrigerant mixture R407C, composed of HFC32/HFC125/HFC134a, is calculated using a prediction method developed by the present authors. The axial distributions of vapor and liquid bulk mass fraction, temperature, heat flux, mass flux, etc., are obtained for both a smooth tube and a microfin tube. The calculation results for both tubes are compared to examine the condensation enhancement effect of the microfin tube.

NOMENCLATURE

c_p	isobaric specific heat
D_{kj}	coefficient of diffusion
D_{kj}^M	diffusivity of pair (k-j) in multi-component mixture
D_{kj}^*	coefficient of diffusion
d_{wi}	inside diameter of the inner tube
d_{wo}	outside diameter of the inner tube
D	inside diameter of the outer tube
G	mass velocity
Ga	Galileo number
h	enthalpy
j_k	diffusion mass flux of component k
M_k	molecular weight of component k
$\dot{m}$	total condensation mass flux
$\dot{m}_k$	condensation mass flux of component k
Nu	Nusselt number
P	pressure
$\frac{dP_F}{dz}$	frictional pressure gradient
Ph	phase change number
Pr	Prandtl number
q	heat flux
Re	Reynolds number
Sc_k	Schmidt number of component k
Sh_k	Sherwood number of component k
T	temperature
W	mass flow rate
x	vapor quality
y_k	mass fraction of component k
z	refrigerant flow direction

Greeks

α	heat transfer coefficient
β_k	mass transfer coefficient of component k
η_A	enlargement ratio of heat transfer area
λ	thermal conductivity
μ	dynamic viscosity
ρ	density
ξ_k	ratio of $\dot{m}_k$ to $\dot{m}$
Φ_V	two-phase multiplier
X_{tt}	Lockhart-Martinelli parameter
ψ	void fraction

Subscripts

b	bulk
C	cooling water
i	vapor-liquid interface

in	refrigerant inlet (cooling water outlet)	r	refrigerant
k	component k (k=1, 2, 3)	V	vapor
L	liquid	w	wall

INTRODUCTION

In recent years we began to use the zeotropic refrigerant mixture R407C as the substitute for R22 widely used as working fluid in air-conditioning and refrigeration industries. However, the use of such zeotropic refrigerant mixture results in the reduction of heat exchanger performance due to mass transfer resistance. Consequently, the enhancement of the heat and mass transfer is required more strongly when we use a zeotropic refrigerant mixture as working fluid. From this point of view, special interests have been taken in the improvement of heat transfer performance of microfin tubes.

The present paper deals with heat transfer performance of a counter flow double-tube condenser for ternary zeotropic refrigerant mixture R407C. The condensation characteristics of R407C in a horizontal smooth tube and a horizontal microfin tube are calculated using a prediction method developed by the present authors [1]. Then, the calculation results of the microfin tube are compared with the calculation ones of the smooth tube, to examine the condensation enhancement effect of the microfin tube.

PREDICTION MODEL

Figure 1 shows the physical model of a counter flow double-tube condenser, where the inside and outside diameters of an inner tube are d_{wi} and d_{wo}, respectively, and the inside diameter of an outer tube is D. A smooth tube and a microfin tube are used as the inner tube of the condenser. The vapor of ternary zeotropic refrigerant mixture R407C flows into the inner tube with a mass flow rate W_{in} (mass velocity G_r), while the cooling water flows counter-currently in an annulus with mass flow rate W_C (mass velocity G_C). The refrigerant vapor starts to condense at an axial position $z = 0$. At an arbitrary position z in the two-phase region (vapor quality x), the bulk vapor is represented by the thermodynamic state $(P, T_{Vb}, y_{1Vb}, y_{2Vb}, y_{3Vb})$, the vapor-liquid interface is of state $(P, T_i, y_{1Vi}, y_{2Vi}, y_{3Vi}, y_{1Li}, y_{2Li}, y_{3Li})$ and the bulk liquid is of state $(P, T_{Lb}, h_{Lb}, y_{1Lb}, y_{2Lb}, y_{3Lb})$. The temperature of the cooling water is represented by T_C. Symbols T_{wi} and T_{wo} denote the inside and outside wall temperature of the inner tube, and q_{wi} is the wall heat flux based on the inside surface area of the inner tube. Symbol $\dot{m}$ denotes the total condensation mass flux. Symbols α_L and α_V are the heat transfer coefficients of the liquid film and the vapor, respectively, and symbols β_{kL} and β_{kV} are the liquid and vapor mass transfer coefficients of component k $(k = 1, 2, 3)$, respectively. Symbol α_C is the heat transfer coefficient of the cooling water.

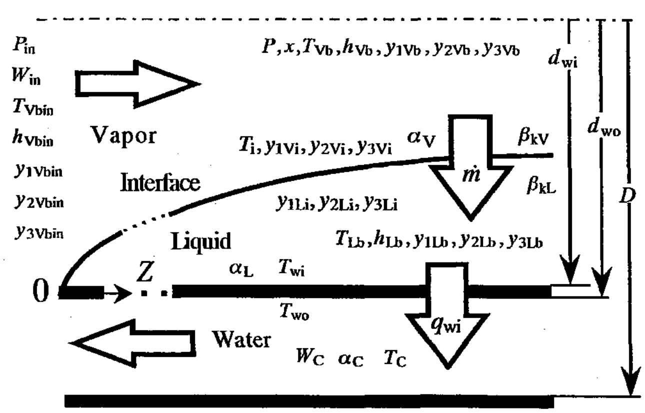


Fig.1 Physical model of double.tube condenser

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 subcooled.

(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 microfin tube [2, 3].

(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 [2] and a horizontal microfin tube [2, 4].

(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 k is calculated by a correlation shown in Table 1. This correlation was derived from the correlation of the frictional pressure drop [5], based on the Chilton-Colburn analogy [6]. In this calculation, the diffusion coefficient of ternary vapor mixture is evaluated using the Stefan-Maxwell equation [7], where the interaction effect of mass transfer between component k and j is neglected.

The governing equations to predict the heat exchange performance of a counter flow double-tube condenser are as follows:

(1) Momentum balance of refrigerant

$$\frac{dP}{dz} = -\left(\frac{4W_{\mathrm{in}}}{\pi\, d_{\mathrm{wi}}^{2}}\right)^{2} \frac{d}{dz}\left[\frac{x^2}{\psi \rho_{\mathrm{V}}} + \frac{(1-x)^2}{(1-\psi)\rho_{\mathrm{L}}}\right] + \frac{dP_{\mathrm{F}}}{dz} \tag{1}$$

where dP/dz is the static pressure gradient, ψ is the void fraction estimated from the Smith equation for two phase flow in a smooth tube [8], and dP_{F}/dz is the frictional pressure gradient calculated using correlations in Table 1.

(2) Heat balance of refrigerant

$$q_{\mathrm{wi}} = -\frac{W_{\mathrm{in}}}{\eta_{\mathrm{A}} \pi\, d_{\mathrm{wi}}} \frac{d}{dz}\left\{x\, h_{\mathrm{Vb}} + (1-x) h_{\mathrm{Lb}}\right\} = \alpha_{\mathrm{L}}\left(T_{\mathrm{i}} - T_{\mathrm{w}}\right) \tag{2}$$

where η_{A} is the enlargement ratio of heat transfer area of microfin tube. In the case of smooth tube, η_{A} is set to be equal to 1. The liquid film heat transfer coefficient α_{L} is calculated using correlations in Table 1.

(3) Mass balance of component k in vapor core

$$\dot{m}_{\mathrm{k}} = -\frac{W_{\mathrm{in}}}{\pi\, d_{\mathrm{wi}}} \frac{d}{dz}\left(x\, y_{\mathrm{kVb}}\right) = -\frac{W_{\mathrm{in}}\, y_{\mathrm{kVi}}}{\pi\, d_{\mathrm{wi}}} \frac{dx}{dz} - \beta_{\mathrm{kV}}\left(y_{\mathrm{kVi}} - y_{\mathrm{kVb}}\right) \tag{3}$$

where $\dot{m}_{\mathrm{k}}$ is the condensation mass flux of component k. The vapor mass transfer coefficient β_{kV} is calculated using the correlation in Table 1.

(4) Mass balance of component k in liquid film

$$y_{\mathrm{kLb}} = y_{\mathrm{kLi}} \tag{4}$$

(5) Relation between vapor quality and mass fraction

$$x = \frac{y_{\mathrm{kVbin}} - y_{\mathrm{kLb}}}{y_{\mathrm{kVb}} - y_{\mathrm{kLb}}} \tag{5}$$

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

(6) Radial wall heat conduction in inner tube

$$q_{\mathrm{wi}} = \frac{2\lambda_{\mathrm{w}}\left(T_{\mathrm{wi}} - T_{\mathrm{wo}}\right)}{\eta_{\mathrm{A}}\, d_{\mathrm{wi}}\, ln\left(d_{\mathrm{wo}}/d_{\mathrm{wi}}\right)} \tag{6}$$

where λ_{w} is the thermal conductivity of the inner tube.

(7) Heat balance of cooling water

$$q_{\mathrm{wi}} = -\frac{W_{\mathrm{C}}\, c_{\mathrm{pC}}}{\eta_{\mathrm{A}}\, \pi\, d_{\mathrm{wi}}} \frac{dT_{\mathrm{C}}}{dz} = \frac{d_{\mathrm{wo}}}{\eta_{\mathrm{A}}\, d_{\mathrm{wi}}} \alpha_{\mathrm{C}}\left(T_{\mathrm{wo}} - T_{\mathrm{C}}\right) \tag{7}$$

where the heat transfer coefficient of cooling water α_{C} is calculated using the Dittus-Boelter equation.

Table 1 Correlations for frictional pressure drop, heat and mass transfer characteristics

<table>
<tr><th>Smooth tube</th><th>Microfin tube</th></tr>
<tr><td>Correlation for frictional pressure drop
$\Phi_V = 1 + 0.5\left[\frac{G_r}{\sqrt{g d_{wi} \rho_V (\rho_L - \rho_V)}}\right]^{0.75} X_{tt}^{0.35}$
where:
$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_V}{\rho_L}\right)^{0.5} \left(\frac{\mu_L}{\mu_V}\right)^{0.1}$</td><td>Correlation for frictional pressure drop
$\Phi_V = 1.1 + 1.3\left[\frac{G_r X_{tt}}{\sqrt{g d_w \rho_V (\rho_L - \rho_V)}}\right]^{0.35}$
where:
$X_{tt} = \left(\frac{1-x}{x}\right)^{0.9} \left(\frac{\rho_V}{\rho_L}\right)^{0.5} \left(\frac{\mu_L}{\mu_V}\right)^{0.1}$</td></tr>
<tr><td>Correlation of liquid film heat transfer
$Nu \equiv \frac{\alpha_L d_{wi}}{\lambda_L} = \left(Nu_F{}^2 + Nu_B{}^2\right)^{1/2}$
$Nu_F = 0.0152\,(1 + 0.6 Pr_L^{0.8})\,(\Phi_V / X_{tt})\, Re_L^{0.77}$
$Nu_B = 0.725 H(\psi)\left(\frac{Ga Pr_L}{Ph}\right)^{\frac{1}{4}}$
$H(\psi) = \psi + \{10[(1-\psi)^{0.1} - 1] + 1.7 \times 10^{-4} Re\}$
$\sqrt{\psi}(1 - \sqrt{\psi})$</td><td>Correlation of liquid film heat transfer
$Nu \equiv \frac{\alpha_L d_{wi}}{\lambda_L} = \left(Nu_F{}^2 + Nu_B{}^2\right)^{1/2}$
$Nu_F = 0.0152(3 + Pr_L^{1.1})(\Phi_V / X_{tt}) Re_L^{0.68}$
$Nu_B = \frac{0.725}{\eta_A^{1/4}} H(\psi)\left(\frac{Ga Pr_L}{Ph}\right)^{\frac{1}{4}}$
$H(\psi) = \psi + \{10(1-\psi)^{0.1} - 8.0\}\sqrt{\psi}(1 - \sqrt{\psi})$</td></tr>
<tr><td colspan="2">Correlation of vapor mass transfer
$Sh_{kV} \equiv \frac{\beta_{kV} d_{wi}}{\rho_V D_{kk}^*} = 0.023 \cdot \sqrt{\psi}\, \Phi_V{}^2 Re_V^{0.8} Sc_{kV}^{1/3}$
where:
$Sc_{kV} = \frac{\mu_V}{\rho_V D_{kk}^*}$, $D_{kj}^* = D_{k3} - D_{kj}$, $D_{kj} = M_k D_{kj}^M \left(\sum_{m=1}^{3} \frac{y_{mVb}}{M_m}\right) - \frac{M_k}{M_j}\left(\sum_{m=1}^{3} D_{km}^M y_{mVb}\right)$</td></tr>
<tr><td colspan="2">Correlation of Dittus-Boelter
$Nu_C = \frac{\alpha_C (D - d_{wo})}{\lambda_C} = 0.023 Re_C^{0.8} Pr_C^{0.4}$</td></tr>
</table>

Table 2 Calculation conditions

(a) Condition of refrigerant and cooling water

Refrigerant		R407C
T_{Vbin}	(K)	323
P_{in}	(MPa)	1.991
G_r	(kg/m^2 s)	100, 300
Cooling water		water
T_{Cin}	(K)	316
G_C	(kg/m^2 s)	200

(b) Dimensions of condenser

Inner tube		Smooth	Microfin
d_{wi}	(m)	0.0064	0.0064
η_A	(-)	1	1.62
d_{wo}	(m)	0.0075	0.0075
λ_w	(W/m K)	385	385
Outer tube		Smooth	Smooth
D	(m)	0.016	0.016

Table 2 shows the calculation conditions. The prediction calculation is done for R407C composed of HFC32 (23wt%), HFC125 (25wt%) and HFC134a (52wt%), which are denoted by components 1, 2 and 3, respectively. In the calculation, 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. Thermodynamic and transport properties of refrigerants are calculated using the program package REFPROP Ver. 6.0 [9].

RESULTS AND DISCUSSION

Figure 2 shows an example of the prediction results of microfin tube. In Fig. 2(a), it is shown that the values of all temperatures T_{Vb}, T_{i}, T_{Lb}, T_{wi} and T_{C} decrease in the refrigerant flow direction. This result represents the typical characteristics of zeotropic refrigerant mixture in a counter-flow condenser. It is also shown that the temperature difference $(T_{\mathrm{Vb}} - T_{\mathrm{i}})$ is not so large. This corresponds to the small mass transfer resistance in the vapor core. In Fig. 2(b), it is shown that the values of $\dot{m}_1$ and $\dot{m}_2$ increase toward the downstream region, while the value of $\dot{m}_3$ decreases. In Figs. 2(c) and (d), it is shown that the values of $y_{1\mathrm{Vb}}$, $y_{1\mathrm{Vi}}$ and $y_{1\mathrm{Lb}} (= y_{1\mathrm{Li}})$ increase toward the downstream region, whereas those of $y_{3\mathrm{Vb}}$, $y_{3\mathrm{Vi}}$ and $y_{3\mathrm{Lb}} (= y_{3\mathrm{Li}})$ decrease. This means that component 1 (HFC32) is concentrated toward the downstream, while component 3 (HFC134a) is diluted. It is also shown that the ratio of $\dot{m}_{\mathrm{k}}$ to $\dot{m}$, ξ_{k}, coincides with y_{kLb} at the beginning point of condensation and approaches to y_{kVb} in downstream region. This reveals that the diffusion in the vapor side is dominant at the beginning point of condensation and it weakens gradually in the refrigerant flow direction.

Figure 3 shows the predicted values of $j_{\mathrm{kVi}}/\dot{m}$, $j_{\mathrm{kLi}}/\dot{m}$ and $(T_{\mathrm{i}} - T_{\mathrm{wi}})/(T_{\mathrm{Vb}} - T_{\mathrm{wi}})$ in smooth and microfin tubes, where ratios of diffusion to total condensation mass flux, $j_{\mathrm{kVi}}/\dot{m}$ and $j_{\mathrm{kLi}}/\dot{m}$, are defined as

$$j_{\mathrm{kVi}}/\dot{m} = y_{\mathrm{kVi}} - \xi_{\mathrm{k}} \tag{8}$$

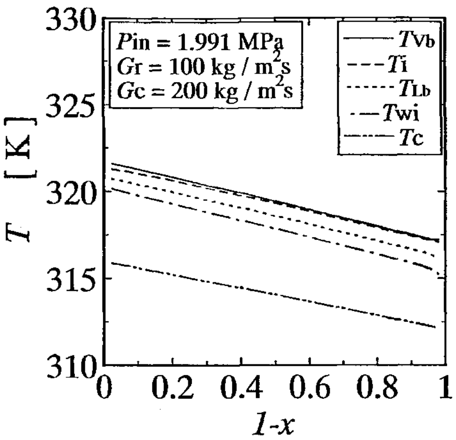
$$j_{\mathrm{kLi}}/\dot{m} = \xi_{\mathrm{k}} - y_{\mathrm{kLi}} \tag{9}$$

The vapor side diffusion dominates the total mass transfer characteristics in the upstream region and it diminishes gradually toward the downstream region. On the other hand, the liquid side diffusion increases toward the downstream region. The trend of $(T_{\mathrm{i}} - T_{\mathrm{wi}})/(T_{\mathrm{Vb}} - T_{\mathrm{wi}})$ reveals that the total heat transfer characteristics are controlled by both the liquid heat transfer and the vapor side mass transfer in the upstream region, while the liquid heat transfer dominates the total heat transfer characteristics in the downstream region. In this prediction calculation, the maximum deterioration of heat transfer due to mass transfer is about 20 %.

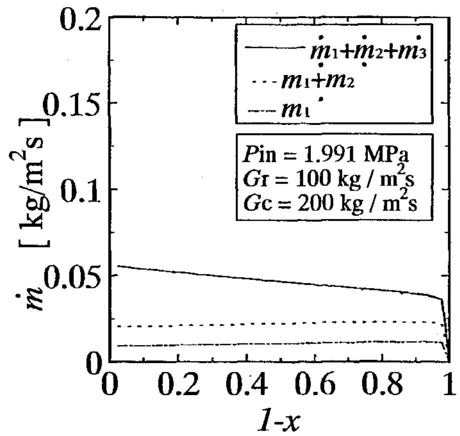
Figure 4 shows the vapor side mass transfer coefficient of component 1 in smooth and microfin tubes. The value of $\beta_{1\mathrm{V}}$ increases as the condensation proceeds. Then, after it reaches a maximum, the value of $\beta_{1\mathrm{V}}$ decreases.

Figure 5 shows the local heat transfer coefficient of liquid film in smooth and microfin tubes. The value of α_{L} decreases as the condensation proceeds for all cases.

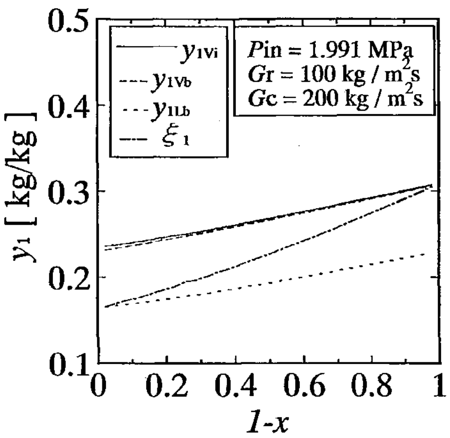
Table 3 shows the prediction results of the heat exchange performance using smooth and microfin tubes. Making a comparison between cases (1) and (2) on the same condition of refrigerant and cooling



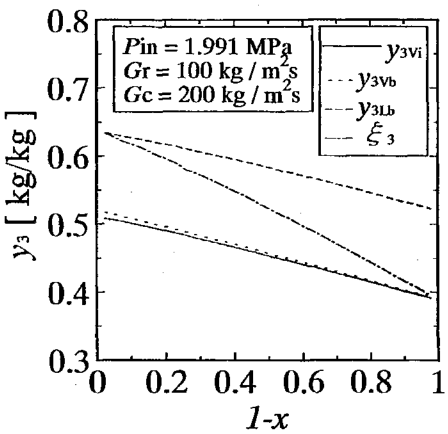
(a) Temperature distribution



(b) Condensation mass flux distribution

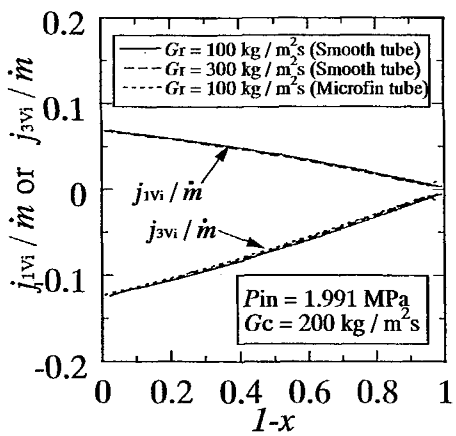


(c) Mass fraction of component 1

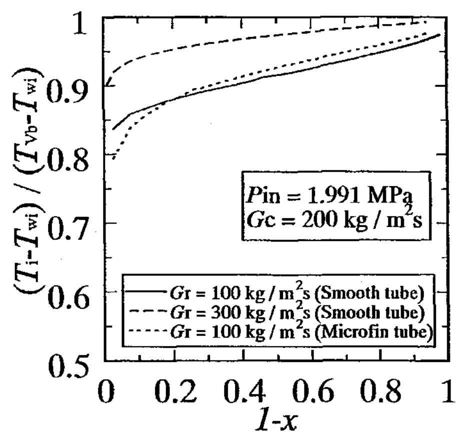


(d) Mass fraction of component 3

Fig. 2 Example of prediction results for microfin tube



(a) Diffusion mass flux distribution



(b) Interface temperature distribution

Fig. 3 Heat and mass transfer characteristics

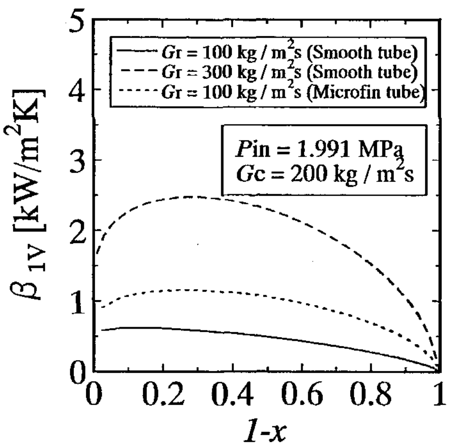


Fig. 4 Vapor mass transfer coefficient of component 1

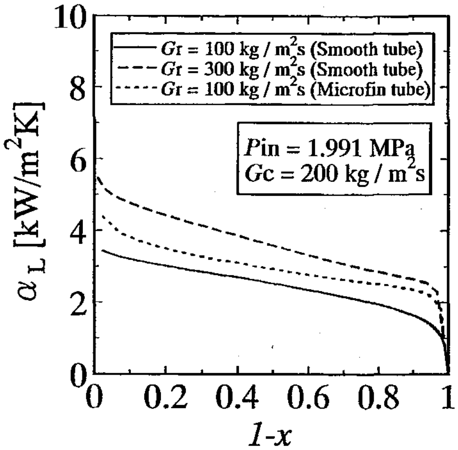


Fig. 5 Liquid heat transfer coefficient

Table 3 Comparison of condensation characteristics of R407C

Case	Inner tube type	T_{Vbin} (K)	P_{in} (MPa)	ΔP_T (kPa)	Q_T (W)	K_m (W/m^{2}K)	α_{rm} (kW/m^{2}K)	α_{Cm} (kW/m^{2}K)	l (m)
1	Smooth	323	1.991	0.27	516	1236	3.352	1.671	3.657
2	Microfin	323	1.991	0.48	509	1489	6.202	1.671	2.718
3	Microfin	322	1.915	0.67	516	1504	6.489	1.671	3.494

(Note) $G_r = 100 \text{kg/m}^2\text{s}$, $G_C = 200 \text{kg/m}^2\text{s}$, $T_{Cin} = 316\text{K}$

water, it is clarified that the total pressure drop and the overall heat transfer coefficient of microfin tube are higher than those of smooth tube. It is also shown from this comparison that the total heat transfer rate and the condensation tube length of microfin tube are lower. When making a comparison between cases (1) and (3) on almost the same condition of the condensation tube length and the total heat transfer rate, the total pressure drop and the overall heat transfer coefficient in microfin tube are higher than those in smooth tube.

CONCLUSIONS

The prediction calculation for the condensation characteristics of R407C in a counter flow double-tube condenser was carried out, and the following results are obtained.

(1) The total mass transfer is controlled by the vapor side at the beginning point of condensation and the effect of the liquid side appears gradually as the condensation proceeds.
(2) The total heat transfer characteristics are controlled by both the liquid heat transfer and the vapor side mass transfer in the upstream region, while the liquid heat transfer dominates the total heat transfer characteristics in the downstream region.
(3) Degree of the heat transfer deterioration due to the mass transfer resistance is maximum 20% at the beginning point of condensation in the present prediction.
(4) When making a comparison between smooth and microfin tubes on the same condition of refrigerant

and cooling water, the total pressure drop and the overall heat transfer coefficient of microfin tube are higher than those of smooth tube, while the total heat transfer rate and the condensation tube length of microfin tube are lower.

(5) When making a comparison between smooth and microfin tubes on the same condition of the condensation tube length and the total heat transfer rate, the total pressure drop and the over all heat transfer coefficient in microfin tube are higher than those in smooth tube.

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