Condensation Heat Transfer of Pure Refrigerants in Microfin Tubes

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ABSTRACT

In the present study the local condensation heat transfer characteristics in a horizontal microfin tube are found to be about 2 times higher than those of a smooth tube with the same inner diameter. This enhancement effect on heat transfer coefficient seems mainly caused by the enlargement ratio of heat transfer area. From this point of view, a correlation, which is modified from the correlation of Haraguchi et al. for smooth tubes, is developed for the condensation heat transfer in microfin tubes with pure refrigerants. With this correlation, the condensation heat transfer characteristics in some kinds of microfin tubes can be predicted very well.

INTRODUCTION

The microfin tube is widely used in heat pump and refrigeration systems for its high heat transfer performance and relatively low flow resistance. Many kinds of microfin tubes have been developed by different companies, and studied by many researchers.

Honda et al. [1] directly observed the flow pattern, correlated the pressure drop characteristic and analyzed heat transfer mechanics of the liquid film in a microfin tube. Haraguchi et al. [2,3] made experiments on condensation of pure refrigerants (HFC134a, HCFC123, HCFC22) in a microfin tube and proposed a correlation for pressure drop characteristic. Shikazono et al. [4] made analysis on condensation heat transfer characteristics in microfin tube in consideration of the surface tension effect on liquid film and compared their analytical results with experiments. On the other hand, Koyama et al. [7], Miyara et al. [8] and Hayashi [9] made a lot of experiments on condensation in different kind of Microfin tube and obtained a series experimental data.

In the present study, we try to develop a model to correlate the local heat transfer coefficient of pure refrigerants condensed in horizontal microfin tubes.

DEVELOPMENT OF IN-TUBE CONDENSATION MODEL

Haraguchi et al. [5] developed a model for the condensation in smooth tube and obtained a good agreement with many experimental results. In the model of Haraguchi et al. [5], the heat transfer coefficient is defined based on the inner surface area in the smooth tube as follows,

$$\alpha \equiv \frac{Q}{\pi d_i \, \Delta l \left(T_{sat} - T_{wi}\right)} \tag{1}$$

$$Nu \equiv \alpha \, d_i / \lambda_L \tag{2}$$

where Q is the heat transfer rate in a subsection which length is Δl. d_i is the mean inner diameter of tube, T_{sat} is the saturated refrigerant temperature, T_{wi} is the wall temperature. λ_L is the saturated liquid thermal conductivity. This model is suitable for the annular and separate flow pattern and covers the range of vapor quality from 0.9 to 0.1. With above definition, the correlation of Haraguchi et al. agrees well with the experimental data of smooth tube, but is only about half of those of microfin tube, as shown in Fig.1.

In the present study, we will modify this model to the condensation in microfin tube and define the heat

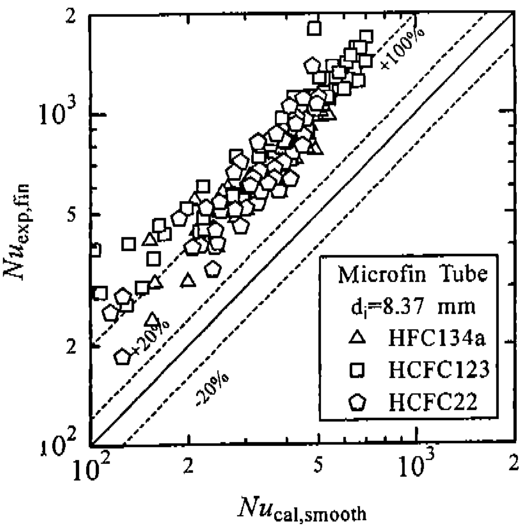

Fig.1 Comparison between the correlation for smooth tubes and the experimental data for a microfin tube

transfer coefficient as follow.

$$\alpha \equiv \frac{Q}{\pi d_{\mathrm{i}}\, \eta_{\mathrm{A}} \Delta l \left(T_{\mathrm{sat}} - T_{\mathrm{wi}}\right)} \tag{3}$$

where η_{A} is the enlargement ratio of heat transfer area. So that the total heat transfer area, other than normal heat transfer area $\pi d_{\mathrm{i}} \Delta l$, is used in the above definition. As the same with the correlation for the smooth tube, the Nu number in microfin tube is also supposed as follow.

$$Nu \equiv \alpha\, d_{\mathrm{i}} / \lambda_{\mathrm{L}} = (Nu_{\mathrm{F}}^{\mathrm{m}} + Nu_{\mathrm{B}}^{\mathrm{m}})^{1/\mathrm{m}} \tag{4}$$

where Nu_{F} is the forced convective condensation component, and Nu_{B} is the natural convective condensation component. The exponent m is determined by the experimental data.

From the turbulent liquid film theory, the forced convective condensation component Nu_{F} can be expressed as follow

$$Nu_{\mathrm{F}} = \mathrm{Re}_{\mathrm{L}}^{*}\, \mathrm{Pr}_{\mathrm{L}} / T_{\mathrm{i}}^{+} \tag{5}$$

where $\mathrm{Re}_{\mathrm{L}}^{*}$ is the liquid Reynolds number. Pr_{L} is the liquid Prandtl number. T_{i}^{+} is the dimensionless temperature difference between the vapor-liquid interface and the tube wall. Their definitions are shown as

$$\mathrm{Re}_{\mathrm{L}}^{*} = \frac{\rho_{\mathrm{L}} \sqrt{\tau_{\mathrm{w}}/\rho_{\mathrm{L}}}\, d_{\mathrm{i}}}{\mu_{\mathrm{L}}} \tag{6}$$

$$T_{\mathrm{i}}^{+} = \frac{\rho_{\mathrm{L}} Cp_{\mathrm{L}} \sqrt{\tau_{\mathrm{w}}/\rho_{\mathrm{L}}} \left(T_{\mathrm{sat}} - T_{\mathrm{wi}}\right)}{Q/\left(\eta_{\mathrm{A}} \pi\, d_{\mathrm{i}}\, \Delta l\right)} \tag{7}$$

where ρ_{L} is the liquid density, μ_{L} is the liquid viscosity, Cp_{L} is the liquid heat capacity. The shear stress at wall τ_{w} can be expressed as

$$\tau_{\mathrm{w}} = \tau_{\mathrm{wV}} \Phi_{\mathrm{V}}^{2} \tag{8}$$

where τ_{wV} is the shear stress of vapor-only flow at wall and is estimated by the following correlation.

$$\tau_{\mathrm{wV}} = \frac{0.023 G^{2} x^{2} / \rho_{\mathrm{V}}}{\left(G\, x\, d_{\mathrm{i}} / \mu_{\mathrm{V}}\right)^{0.2}} \tag{9}$$

where G is the refrigerant mass flow rate, x is the vapor quality, ρ_{V} is the vapor density, μ_{V} is the vapor viscosity. For the two-phase multiplier Φ_{V}, the experimental correlation of Haraguchi et al. [2] can be used as follow.

$$\Phi_{\mathrm{V}} = 1.1 + 1.3 \left[\frac{G \cdot \chi_{\mathrm{tt}}}{\sqrt{g\, d_{\mathrm{i}} \rho_{\mathrm{V}} \left(\rho_{\mathrm{L}} - \rho_{\mathrm{V}}\right)}} \right]^{0.35} \tag{10}$$

where

$$\chi_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_V}{\rho_L}\right)^{0.5}\left(\frac{\mu_L}{\mu_V}\right)^{0.1} \tag{11}$$

Substituting Eqs.6, 8 and 9 into Eq.5, we can obtain the following equation (Eq.12) for the forced convective condensation component, in which the liquid Reynolds number $\mathrm{Re_L}$ is defined by Eq.13.

$$Nu_F = 0.152(\Phi_V/\chi_{tt})\mathrm{Re_L}^{0.9}(\mathrm{Pr_L}/T_i^+) \tag{12}$$

$$\mathrm{Re_L} = G(1-x)d_i/\mu_L \tag{13}$$

On the other hand, the natural convective condensation component Nu_B can be expressed by the following equation.

$$Nu_B = \frac{0.725}{\eta_A^{1/4}}\cdot H(\xi)\left(\frac{Ga\,\mathrm{Pr_L}}{Ph_L}\right)^{1/4} \tag{14}$$

where Ga is the Galileo number ($= g\rho_L^2 d^3/\mu_L^2$) and Ph_L is the phase change number ($= Cp_L(T_{sat}-T_{wi})/\Delta h_{vl}$). In the above equation, the function $H(\xi)$ is the modification for the difference of the condensed liquid film between the inner surface of the tube and the plate wall on which the Nusselt theory is suitable. In our model, this function only depends on the void fraction.

$$H(\xi) = \xi + A\sqrt{\xi}(1-\sqrt{\xi}) \tag{15}$$

where A is the function of void fraction ξ which is estimated by Smith's correlation [6] as follow. Although this correlation was developed for the smooth tube, it is also considered to apply for microfin tube without taking large error.

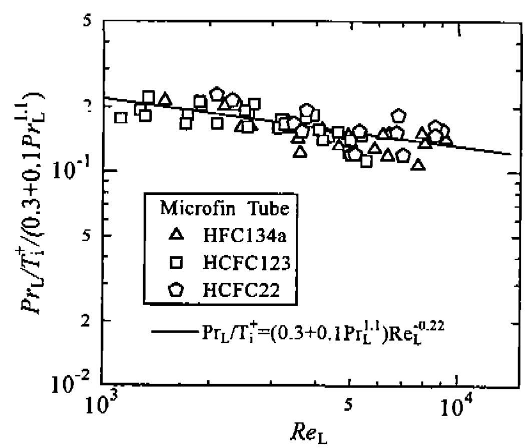

Fig. 2 Forced convective condensation component

$$\xi = \left[1+\frac{\rho_V}{\rho_L}\left(\frac{1-x}{x}\right)\left(0.4+0.6\sqrt{\frac{\frac{\rho_L}{\rho_V}+0.4\frac{1-x}{x}}{1+0.4\frac{1-x}{x}}}\right)\right]^{-1} \tag{16}$$

In the above equations, the two parameters Pr/T_i^+ and A are unknown and should be determined by the experimental data with the same method to the smooth tube [5]. From the experimental data of Haraguchi [2], we obtained the optimum values for the above two parameters by trial calculation, and show them in Figs. 2 and 3. The exponent m is set to 2. In Fig. 2, the experimental data in high flow rate and high quality region are used to reply the effect of forced convective condensation. On the other

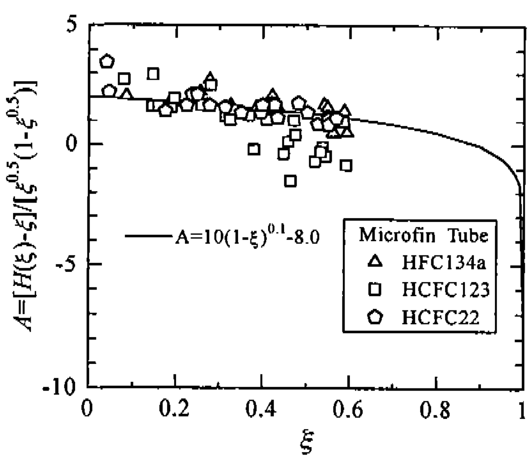

Fig.3 Natural convective condensation component

hand, the experimental data in low flow rate and low quality region are used to reply the effect of natural convective condensation in Fig. 3. The final correlation for local condensation heat transfer characteristics in the microfin tube is developed as follows:

$$Nu = (Nu_F^{\,2} + Nu_B^{\,2})^{1/2} \tag{17}$$

$$Nu_F = 0.152[0.3 + 0.1\mathrm{Pr_L}^{1.1}](\Phi_V/\chi_{tt})\mathrm{Re_L}^{0.68} \tag{18}$$

$$\Phi_V = 1.1 + 1.3\left[\frac{G\cdot\chi_{tt}}{\sqrt{g\,d_i\rho_V(\rho_L-\rho_V)}}\right]^{0.35} \tag{19}$$

$$Nu_B = \frac{0.725}{\eta_A^{1/4}}\cdot H(\xi)\left(\frac{Ga\,\mathrm{Pr_L}}{Ph_L}\right)^{1/4} \tag{20}$$

$$H(\xi) = \xi + A\sqrt{\xi}(1-\sqrt{\xi}) \tag{21}$$

$$A = 10(1-\xi)^{0.1} - 8.0 \tag{22}$$

COMPARISON BETWEEN CORRELATION AND EXPERIEMNTAL DATA

Figure 4 shows the comparison between the correlation equation (17) and the experimental data of Haraguchi et al. [2]. From this figure, it is found that the prediction values from the correlation agree very well with the experimental data for three kinds of refrigerants HFC134a, HCFC 123 and HCFC22. Figures 5(a) and (b) illustrate the local heat transfer characteristics along the flow direction under the conditions of G=300, 100 kg/(m^{2}s). In Fig. 5(a), the prediction values of the present correlation are matched well with experimental results in high flow rate condition, while in low flow rate condition the experimental data are higher than those of prediction in high quality region, as shown in Fig.5(b).

Because this correlation is developed from the model for the smooth only using the experimental data of Haraguchi et al.[2], we have to compare it with the experimental data from other sources to make sure its generality. Figures 6(a), (b) and (c) show the comparison between the correlation equation (17) and the experimental data of Koyama et al. [7], Miyara et al. [8] and Hayashi [9]. All of the experimental data show a good agreement with the correlation equation (20). The dimensions of

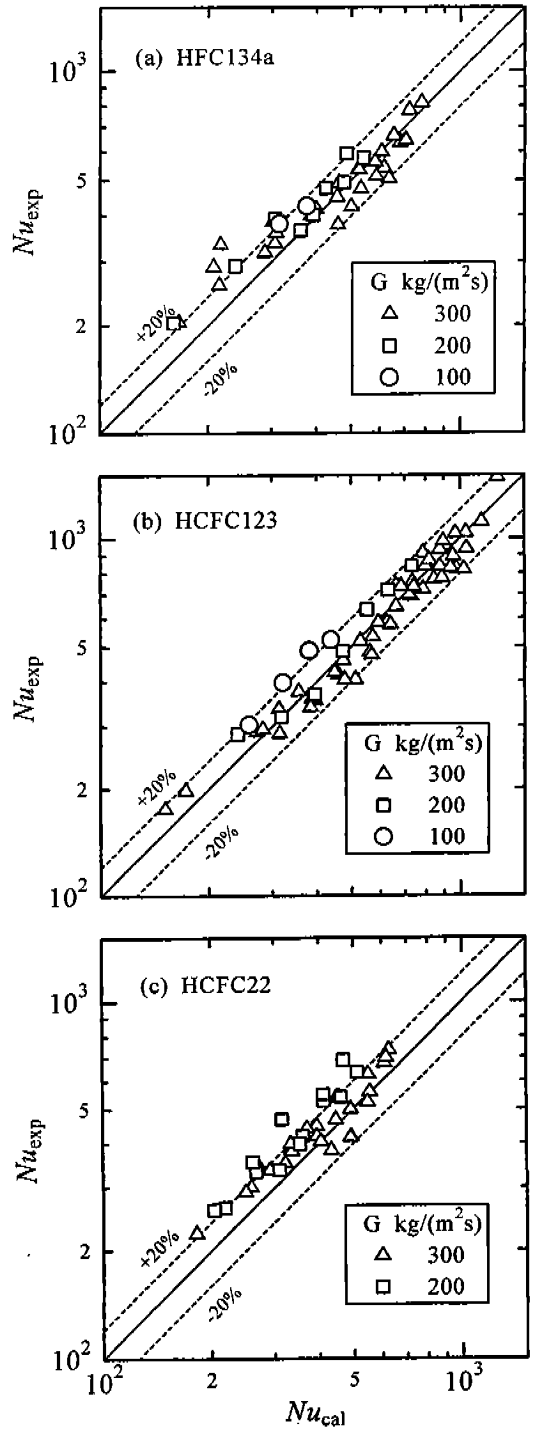

Fig. 4 Comparison between experimental data of Haraguchi et al. and equation (20)

these microfin tubes are listed in Table 1 for reference.

CONCLUSIONS

In the case of pure refrigerant condensed in horizontal microfin tubes, the most important parameter, heat transfer area enlargement ratio η_A is considered in the definition of heat transfer coefficient. With this definition, a correlation equation for local heat transfer characteristics is developed from the smooth tube model. The present correlation can predict well with the experimental data from different sources in relatively high flow rate region.

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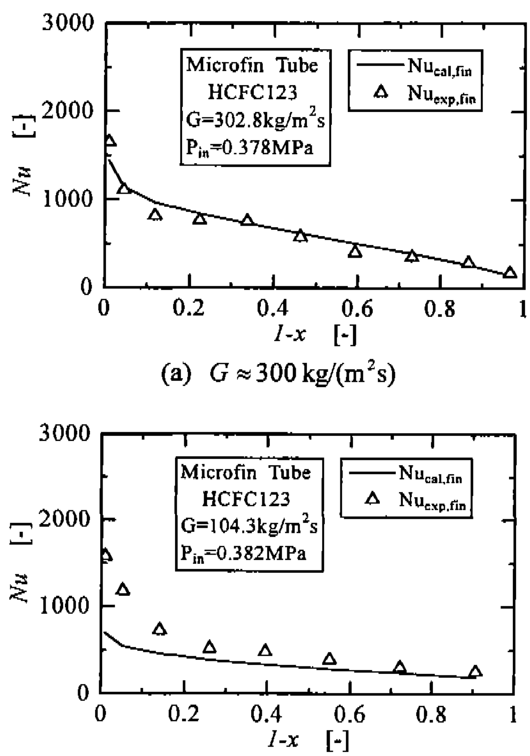

(a) $G \approx 300\,\mathrm{kg/(m^2s)}$

(b) $G \approx 100\,\mathrm{kg/(m^2s)}$

Fig.5 Relation between Nu and 1-x

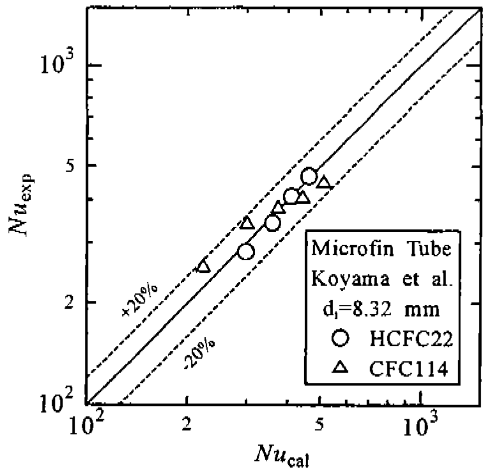

Fig.6 Comparison between present correlation and experimental data of Koyama et al.

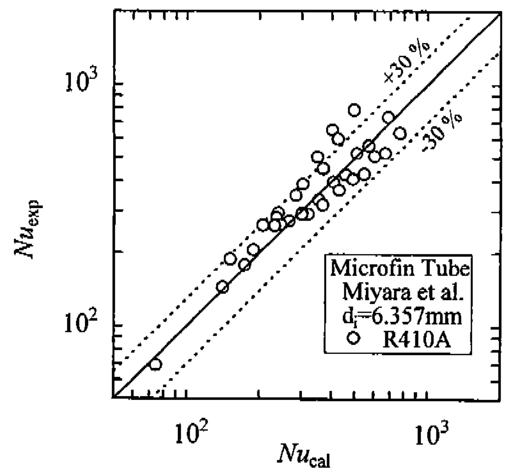

Fig.7 Comparison between present correlation and experimental data of Miyara et al.

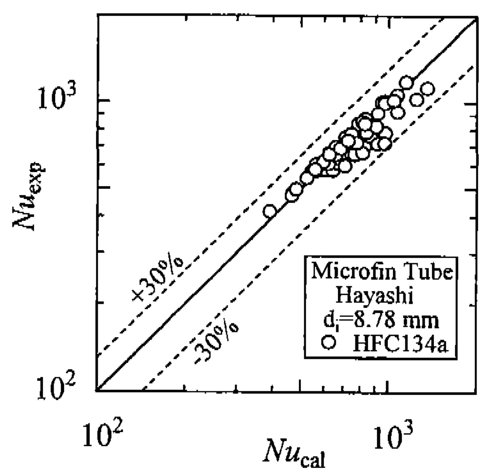

Fig.8. Comparison between present correlation and experimental data of Hayashi

REFERENCES

1. H. Honda, S. Nozu, H. Katayama and H. Nakata, Condensation of Refrigerant R11 in Horizontal Microfin Tubes, Proc. 22nd National Heat Transfer Symp. of Japan, Tokyo, pp.335-337, 1985(in Japanese).
2. H. Haraguchi, S. Koyama, J. Esaki and T. Fujii, Condensation Heat Transfer of Refrigerants HFC134a, HCFC123 and HCFC22 in a Horizontal Smooth Tube and a Horizontal Microfin Tube, Proc. 30th National Heat Transfer Symp. of Japan, Yokohama, pp.343-345, 1993(in Japanese).
3. H. Haraguchi, H. Kogawa, S. Koyama and T. Fujii, An Experiment on Condensation of a HFC134a/ HCFC123 Mixture inside Horizontal Tubes, Proc. 31st National Heat Transfer Symp. of Japan, Sapporo, pp.742-744, 1994(in Japanese).
4. N. Shikazono, M. Itoh, M.Uchida, T. Fukushima and T. Hatada, Predictive Equation Proposal for Condensation Heat Transfer Coefficient of Pure Refrigerants in Horizontal Micro-Fin Tubes, Trans. JSME, Vol.64, No.617, pp.196-203, 1998(in Japanese).
5. H. Haraguchi, S. Koyama and T. Fujii, Condensation of Refrigerants HCFC22, HFC134a and HCFC123 in a Horizontal Smooth Tube (2nd Report, Proposals of Empirical Expressions for Local Heat Transfer Coefficient), Trans. JSME, Vol.60, No.574, pp245-252, 1994(in Japanese).
6. S.L. Smith, Void Fraction in Two-phase Flow: A Correlation Based on Equal Velocity Head Model, Heat and Fluid Flow, Vol.1, No.1, pp.22-39, 1970.
7. S. Koyama, A. Miyara, H. Takamatsu, K. Yonemoto and T. Fujii, Condensation and Evaporation of Non-azeotropic Refrigerant Mixtures of R22 and R114 inside a Spirally Grooved Horizontal Tube, The Reports of Institute of Advanced Material Study, Kyushu University, Vol.1, No.1, pp.57-75, 1988(in Japanese).
8. A. Miyara, K. Nonaka, K. Yoshida, T. Nakashima, H. Uehara and M. Taniguchi, Condensation Heat Transfer and Flow Pattern of a W-shaped Microfin Tube, Proc. 1998 JAR Annual Conference, Tokyo, 1998(in Japanese).
9. T. Hayashi, Enhanced Condensation Heat Transfer of Refrigerant HFC134a in Horizontal Tubes, Master theses, Kyushu University, 1998.3(in Japanese).

Table 1. Dimensions of Microfin Tubes

Dimension		Haraguchi et al. [2]	Koyama et al. [7]	Miyara et al. [8]	Hayashi [9]
O.D.	mm	10.00	9.52	7.00	9.52
I.D.	mm	8.37	8.32	6.357	8.78
Fin heigt	mm	0.17	0.15	0.210	0.20
Fin pitch	mm	0.44	0.44	0.408	0.47
Fin number	-	60	60	50	60
Helix angle	deg	18	30	18	18.7
Area increasing ratio	-	1.52	1.23	1.834	1.51