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Experimental Study on Condensation of Pure Refrigerants in Horizontal Micro-Fin Tube -Proposal of Correlations for Heat Transfer Coefficient and Frictional Pressure drop-

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ABSTRACT

This paper deals with the condensation heat transfer and pressure drop of pure refrigerant in micro-fin tubes. The correlations for heat transfer and frictional pressure drop are proposed using experimental data for 11 micro-fin tubes with different fin dimensions, where test refrigerants were pure refrigerants R22, R123 and R134a. The proposed correlations are developed based on the correlation for void fraction micro-fin tube and the correlation for pressure drop of single phase flow in micro-fin tube. The predicted results show good agreement with experimental results within the deviation of about 30 % for both condensation heat transfer and pressure drop. Experimental results were also compared with previous correlations proposed for micro-fin tube.

1. INTRODUCTION

Micro-fin tubes are widely used in heat pump and refrigeration systems, and the improvement in heat transfer performance of micro-fin tubes are still required in order to make these systems highly efficient and compact. In a viewpoint of optimizing heat exchangers in these systems, proposing correlations of heat transfer and pressure drop of refrigerant in micro-fin tube are effective. Therefore, many researchers have investigated the condensation and flow boiling of refrigerants inside many kinds of micro-fin tubes. For the condensation, Cavallini *et al*. (1995), Kedzieriski-Goncalves (1997), Yu-Koyama (1998), Shikazono *et al*. (1998) and Goto *et al*. (2003) proposed correlations for the condensation heat transfer and/or frictional pressure drop of refrigerant in micro-fin tube. However, there are still unsolved problems as,

(1) The void fraction in micro-fin tube was estimated using the void fraction correlation proposed for smooth tube in many cases.
(2) Many correlations for frictional pressure drop in micro-fin tube were developed based on the correlations proposed for smooth tube such as the Colburn equation, the Blasius equation.
(3) Many correlations for heat transfer in micro-fin tube were developed modifying the correlations proposed for smooth tube.

In the present study, the heat-transfer correlation proposed by Yu-Koyama is modified using correlations for void fraction and friction coefficient, which are developed for micro-fin tube. A correlation for frictional pressure drop is also proposed. To propose these correlations, experimental data obtained by the present authors, Miyara (2003) and Haraguchi (1994) are used.

2. TESTED MICRO-FIN TUBES

2.1 Specification of Tested Micro-fin Tubes

Tables 1 (a), (b) and (c) show the specification of micro-fin tubes tested by the present authors, Miyara and Haraguchi, respectively. In the table, d_o is the outer diameter, d_i is the mean diameter which is the diameter of smooth tube having the same inner cross-section area as that of micro-fin tube, P is the fin pitch, h is the fin height, γ is the vertex angle of fin, β is the helix angle of fins, n is the number of fins, and η_A is the enlargement ratio of heat transfer surface. Experimental data for 11 micro-fin tubes listed in Table 1 are used to develop the correlations for condensation heat transfer coefficient and frictional pressure drop of refrigerant in micro-fin tube.

2.2 Experimental Apparatus

The experimental apparatus used by the present authors is a forced circulation loop by an oil-free liquid pump. The test section to measure the heat transfer and pressure drop characteristics in micro-fin tube is installed in the loop. The superheated vapor is supplied to the test section and condensed in it. The test section is a 4.54 long double-tube heat exchanger, where the refrigerant flows inside an inner tube, while cooling water flows counter-currently in the annulus. The annulus is divided into 7 subsections to measure the local heat transfer rate, refrigerant temperature and refrigerant pressure. The effective heat transfer length of the first to sixth subsections is 0.5 m long, while that of the seventh subsection is 0.624 m long.

The physical quantities measured in the present study are as follows: (1) flow rates of the refrigerant and cooling water, (2) refrigerant temperature and pressure at the both ends of each subsection, (3) wall temperature of the inner tube at the central position of each subsection, and (4) cooling water temperature at the both ends of each subsection.

Miyara and Haraguchi used the experimental apparatuses similar to the present one, and measured the local heat transfer and pressure drop characteristics of refrigerants in micro-fin tubes. Table 2 shows their experimental conditions along with present experimental condition. Tested refrigerants are R134a, R22 and R123.

3. CORRELATION OF FRICTIONAL PRESSURE DROP

Table 1 Dimensions of Micro-fin Tube.

(a) Present data

d_o	d_i	p	h	γ	β	n_g	η_A
7.52	6.51	0.35	0.24	30	13	55	2.08

(b) Reference: Miyara's data

d_o	d_i	p	h	γ	β	n_g	η_A
7.0	6.32	0.36	0.16	58	18	60	1.7
7.0	6.36	0.36	0.21	30	9	60	2.06
7.0	6.35	0.36	0.19	28	18	60	2.00
7.0	6.35	0.35	0.22	30	30	60	2.27
7.0	6.41	0.71	0.19	27	18	30	1.5
7.0	6.30	0.30	0.22	35	18	50	1.9
7.0	6.25	0.30	0.22	30	10	70	2.32
7.0	6.26	0.25	0.22	30	17	70	2.34
7.0	6.27	0.41	0.20	27	7	85	2.46

(c) Reference: Haraguchi's data

d_o	d_i	p	h	γ	β	n_g	η_A
10.0	8.37	0.43	0.17	45	18	60	1.52

Table 2 Experimental conditions.

(a) Present data

Tube Type	Helical micro-fin tube	
Refrigerant	R134a	
P_{in} (MPa)	1.2	1.5
G (kg/(m^{2}s))	200-500	200-450
T_{in} (^{o}C)	46	55

(b) Miyara's data

Tube Type	Helical micro-fin tube
Refrigerant	R22
P_{in} (MPa)	1.2-1.94
G (kg/(m^{2}s))	200-400
T_{in} (^{o}C)	30-50

(c) Haraguchi's data

Tube Type	Helical micro-fin tube		
Refrigerant	R22	R134a	R123
P_{in} (MPa)	1.85	1.25	0.38
G (kg/(m^{2}s))	112-300	120-400	100-350
T_{in} (^{o}C)	48	48	70

3.1 Data Reduction for Frictional Pressure Drop

To obtain the fractional pressure drop through each subsection, the pressure drop due to momentum change, ΔP_M , is estimated from the following equation

$$\Delta P_M = -\Delta\left\{\frac{G^2x^2}{\xi\rho_V}+\frac{G^2(1-x)^2}{(1-\xi)\rho_L}\right\} \tag{1}$$

where G is the mass velocity, x is the vapor quality, ρ is the density, ξ is the void fraction, and subscripts V and L denote vapor and liquid, respectively. The void fraction ξ in micro-fin tube is estimated by the following correlation of Koyama *et al*. (2001), which was proposed based on the experimental data on the void fraction in micro-fin tube.

$$\xi = 0.81\ \xi_{Smith} + 0.19\, x^{100(\rho_v/\rho_l)^{0.8}}\ \xi_{Homo} \tag{2}$$

In the above equation, ξ_{Smith} is the Smith correlation (1971), and ξ_{Homo} is the void fraction of homogeneous two phase flow.

Frictional pressure drop through each subsection, ΔP_F , is calculated from the measured static pressure drop, ΔP_T , and the pressure drop due to momentum change, ΔP_M , as

$$\Delta P_F = \Delta P_T - \Delta P_M \tag{3}$$

3.2 Correlation for Frictional Pressure Drop

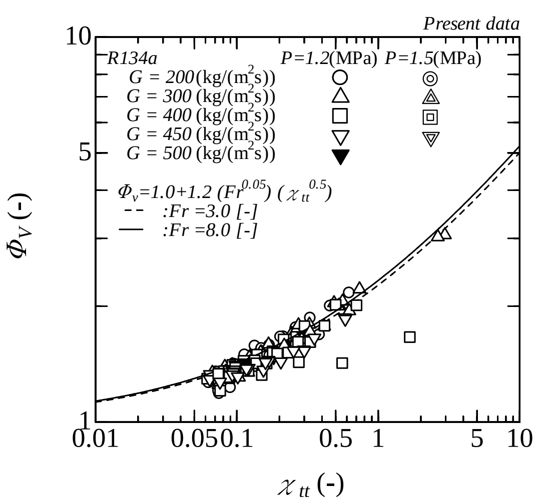
In the present study the frictional pressure drop is attempted to be correlated by the Lockhart-Martinelli parameters as

$$\Phi_V = \sqrt{\left(\frac{\Delta P_F}{\Delta z}\right)\Big/\left(\frac{\Delta P_V}{\Delta z}\right)} \quad , \quad X_{tt} = \left(\frac{1-x}{x}\right)^{0.9}\left(\frac{\rho_V}{\rho_L}\right)^{0.5}\left(\frac{\mu_L}{\mu_V}\right)^{0.1} \tag{4), (5}$$

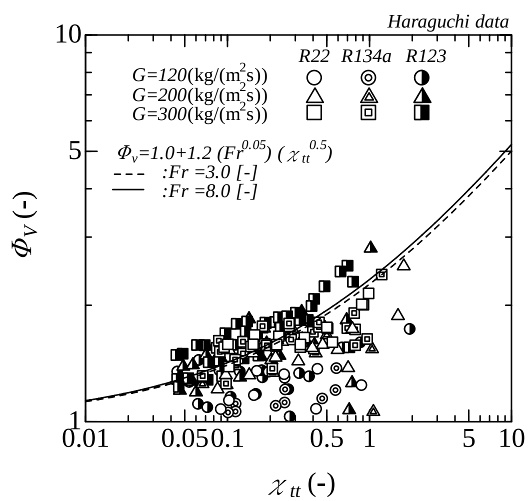
where Δz is the axial length between the neighboring pressure ports, $(\Delta P_V/\Delta z)$ is the pressure drop when only the vapor component flows in tube, and μ is the viscosity. The value of $(\Delta P_V/\Delta z)$ in equation (4) is defined as

$$\left(\frac{\Delta P_V}{\Delta z}\right) = -2\, f_V\, \frac{G^2x^2}{d_i\rho_V} \tag{6}$$

where the friction coefficient, f_V , is estimated by the Carnavos correlation proposed for single phase flow inside



(a) Present data



(b) Haraguchi's data

Fig. 1 Relation between Φ_V and χ_{tt}

low-fin and micro-fin tubes (1980).

The following correlation is obtained based on the present experimental data.

$$\Phi_V = 1 + 1.2\ Fr^{0.05}\ X_{tt}^{0.5} \tag{7}$$

where Fr is the Froude number defined as

$$Fr = G \Big/ \sqrt{g\, d_i\, \rho_V \left(\rho_L - \rho_V\right)} \tag{8}$$

where g is the gravity acceleration.

Fig. 1 shows the comparison between the experimental data and the correlation expressed by equation (7), where Figs. (a) and (b) illustrate the present results and Haraguchi's results, respectively. Equation (7) correlates the present data well, while it does not predicts Haraguchi's data well. Haraguchi's data are scattered over wide area on the graph. This suggests that errors of his measurement in pressure drop are relatively large. Therefore, his data were not referred when equation (7) was obtained.

4. CORRELATION OF HEAT TRANSFER COEFFICIENT

4.1 Data Reduction for Heat Transfer Coefficient

The heat transfer rate in each subsection, Q, is calculated as

$$Q = W_c C_{Pc} \Delta T_c \tag{9}$$

where W_c and C_{Pc} are the mass flow rate and the specific isobaric heat of cooling water, and ΔT_c is the temperature change of cooling water in each subsection. The heat transfer coefficient, α, and the Nusselt number, Nu, are defined as

$$\alpha = \frac{Q}{\pi\, d_i\, \eta_A\, \Delta z_H \left(T_{sat} - T_{wi}\right)}\ , \qquad Nu = \frac{\alpha\, d_i}{\lambda_L} \tag{10), (11}$$

where η_A is the area enlargement ratio, Δz_H is the effective heat transfer length, T_{sat} is the saturation temperature, and T_{wi} is the inner wall temperature. T_{sat} is obtained from the measured static pressure using the equation of thermodynamic state, and T_{wi} is estimated from the measured outer wall temperature using the radial heat conduction equation in the tube wall. The vapor quality, x, is also calculated from the energy conservation equation in each subsection.

4.2 Correlation for Heat Transfer Coefficient

Yu-Koyama (1998) proposed the correlation for heat transfer coefficient in micro-fin tube, which is functionally expressed by the combination of the forced and natural convection condensation terms as

$$Nu = \left(Nu_F^{\,m} + Nu_N^{\,m}\right)^{1/m} \tag{12}$$

where Nu_F is the forced convection condensation term, Nu_N is the natural convection condensation term, and m is the exponent. In their correlation, the void fraction in micro-fin tube is estimated using the void fraction correlation proposed for smooth tube, and the correlation for frictional pressure drop used in calculation of Nu_F is based on the Colburn equation for smooth tube. Therefore, in the present study, the Yu-Koyama correlation is modified based on the void fraction correlation proposed for micro-fin tube and the correlation for frictional pressure drop expressed by equation (7).

From the turbulent liquid film theory, the forced convective condensation term is expressed as

$$Nu_F = Re_L^*\ Pr_L \Big/ T_i^+ \tag{13}$$

where Re_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, and their definitions are shown as

$$Re_L^* = \frac{\rho_L \sqrt{\tau_w / \rho_L}\ d_i}{\mu_L} \quad , \quad T_i^+ = \frac{\rho_L C p_L \sqrt{\tau_w / \rho_L}\left(T_{sat} - T_{wi}\right)}{Q / \left(\eta_A \pi\, d_i\ \Delta z_H\right)} \qquad (14), (15)$$

The wall shear stress τ_w in equations (14) and (15) can be expressed as

$$\tau_w = {\Phi_V}^2 \frac{f_V}{2} \frac{G^2 x^2}{\rho_V} \qquad (16)$$

Substituting equations (14), (15) and (16) into equation (13), the following equation is obtained.

$$Nu_F = \sqrt{\frac{f_V}{2}}\, Re_L\, \Phi_V \left(\frac{\rho_L}{\rho_V}\right)^{0.5} \left(\frac{x}{1-x}\right) \left(\frac{Pr_L}{T_i^+}\right) \qquad (17)$$

where Re_L is the liquid Reynolds number defined as

$$Re_L = G(1-x) d_i / \mu_L \qquad (18)$$

On the other hand, the natural convective condensation term, Nu_N, is assumed to be expressed as

$$Nu_N = G(\eta_A, Bo) H(\xi) \left(\frac{Ga\, \mathrm{Pr_L}}{Ph_\mathrm{L}}\right)^{1/4} \qquad (19)$$

where Bo, Ga and Ph_L are the Bond number, the Galileo number and the phase change number, respectively, which are defined as

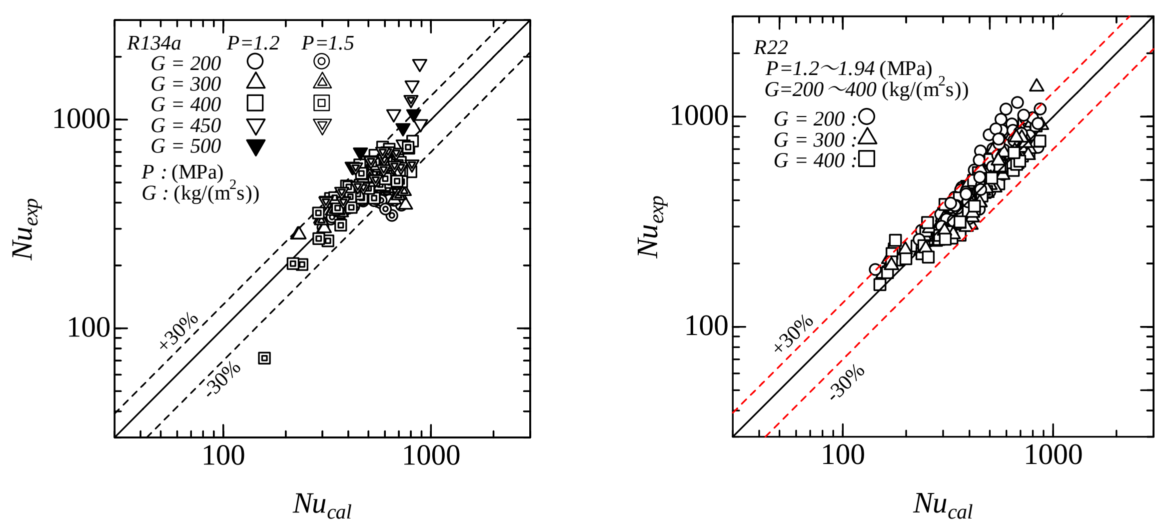
$$Bo = (P - t)\, d_i\ g\, (\rho_L - \rho_V)/\sigma \quad , \quad Ga = g\, \rho_L^2\, d_i^3 / \mu_L^2 \quad , \quad Ph_L = C_{PL}(T_{sat} - T_{wi})/\Delta h_{LV} \qquad (20), (21), (22)$$

where P is the pitch of micro-fin, t is the width of micro-fin tip, σ is the surface tension, and Δh_{LV} is the latent heat. In equation (19), $G(\eta_A, Bo)$ is the function expresses effects of area enlargement and surface tension of liquid in groove between micro-fins, and $H(\xi)$ is the function which expresses effect of thick liquid film flowing at the bottom of tube.

To obtain the optimum correlation for heat transfer coefficient in micro-fin tube, additional assumptions are introduced as

(1) Functions of Φ_V and f_V in equation (17) are given by equation (7) and the Carnavos correlation, respectively.

(2) Function $H(\xi)$ in equation (19) is the same as that of smooth tube, that is,



(a) Present data (b) Miyara's data

Fig. 2 Comparison of Nu_{exp} and Nu_{cal}

Table 3 Deviations between Nu_{exp} and Nu_{cal}

Experimental Data Source	Miyara		Haraguchi		Present study		Overall	
Number of original data	430		255		216		901	
Residual	*Er A*	*Er B*	*Er A*	*Er B*	*Er A*	*Er B*	*Er A*	*Er B*
Present eqn.	0.050	0.086	0.100	0.140	0.030	0.160	0.059	0.119
Cavallini *et al.*	0.003	0.330	-0.430	0.460	-0.140	0.210	-0.137	0.329
Kedzerski-Goncalves	-0.040	0.240	-0.280	0.290	0.040	0.140	-0.092	0.226
Yu-Koyama	0.170	0.240	-0.060	0.140	0.010	0.160	0.072	0.195
Shikazono *et al.*	-0.590	0.620	-0.010	0.150	-0.130	0.220	-0.327	0.400
Goto *et al.*	-0.160	0.240	-0.250	0.260	-0.130	0.200	-0.175	0.235

$$Er\ A = \frac{1}{n}\sum_{k=1}^{n}\frac{Nu_{cal,k} - Nu_{exp,k}}{Nu_{cal,k}} \quad , \quad Er\ B = \frac{1}{n}\sum_{k=1}^{n}\frac{ABS\left(Nu_{cal,k} - Nu_{exp,k}\right)}{Nu_{cal,k}}$$

where n is number of experimental data.

$$H(\xi) = \xi + \left\{10(1-\xi)^{0.1} - 8.9\right\}\sqrt{\xi}\left(1-\sqrt{\xi}\right) \tag{23}$$

Therefore, the parameter Pr_L / T_i^+ in equation (17) and the function $G(\eta_A, Bo)$ in equation (19) should be determined together with the exponent m in equation (12),. Using the experimental data of present study, Miyara and Haraguchi, optimum value of exponent m, optimum functions of Pr_L / T_i^+ and $G(\eta_A, Bo)$ are determined by the trail and error. The results are as follows.

$$m = 2 \quad , \quad \frac{Pr_L}{T_i^{\,+}} = \frac{5.0}{Re_L^{\,0.5}} \quad , \quad G(\eta_A, Bo) = \frac{1.98}{\eta_A^{\,0.5}\ Bo^{0.1}} \qquad (24), (25), (26)$$

The final correlation for condensation heat transfer coefficient in the micro-fin tubes is summarized as

$$\mathrm{Nu} = \left(\mathrm{Nu_F}^{2} + \mathrm{Nu_N}^{2}\right)^{1/2} \tag{27}$$

where

$$Nu_F = 2.12\sqrt{f_V}\ \Phi_V \left(\frac{\rho_L}{\rho_V}\right)^{0.1}\left(\frac{x}{1-x}\right) Re_L^{\ 0.5}\ Pr_L^{\ 0.5} \quad , \quad Nu_N = 1.98\frac{H(\xi)}{\eta_A^{\ 0.5} Bo^{0.1}}\left(\frac{Ga\ \ \mathrm{Pr}_L}{Ph_L}\right)^{0.25} \qquad (28), (29)$$

Fig. 2 shows the comparison of experimental data and predicted values using the present correlation (equation (27)), where Figs. (a) and (b) illustrate results of the present data and Miyara's data, respectively. In both figures, most of experimental data agree well with predicted values within the deviation of 30 %.

Table 3 shows the deviation between experimental data and predicted values using the present and the previous correlations. $Er\ A$ and $Er\ B$ values of the present correlation are the smallest among correlations in Table 3.

5. CONCLUSIONS

New correlations for the condensation heat transfer coefficient and frictional pressure drop of refrigerant condensing in micro-fin tube are developed using experimental data obtained by the present authors, Miyara and Haraguchi.

(1) The correlation for frictional pressure drop is developed using the void fraction correlation in micro-fin tube and the Carnavos correlation.

(2) The correlation for heat transfer coefficient is developed by modifying the Yu-Koyama correlation.

MAIN NOMENCLATURE

Bo	Bond number	(–)	**Subscripts**	
d_i	Average inner diameter of test Micro-fin tube	(m)	c	Water

d_o	Outer diameter of test Micro-fin tube	(m)	*cal*	Calculation
f	Friction coefficient	(–)	*exp*	Experiment
Fr	Froude number	(–)	*F*	Forced convection
G	Refrigerant mass velocity	(kg m^{-2} s^{-1})	*in*	Inlet
g	Gravitational acceleration	(m/s^2)	*L*	Liquid
Ga	Galileo number	(–)	*N*	Natural convection
h	Fin height	(m)	*out*	Outlet
n_g	Number of grooves	(–)	*sat*	Saturation
Nu	Nusselt number	(–)	*V*	Vapor
P	Pressure	(Pa)	*wi*	Inside surface wall
p	Fin Pitch	(m)	*wo*	Outside surface wall
Ph_L	Phase change number	(–)		
Pr	Prandtl number	(–)		
Q	Heat transfer rate of each test section	(W)		
Re	Reynolds number	(–)		
T	Temperature	(K or ^oC)		
t	Fin tip width	(m)		
x	Vapor quality	(–)		
W	Mass flow rate	(kg/h)		
α	Heat transfer coefficient	(W m^{-2} K^{-1})		
β	Spiral angle	(rad)		
γ	Vertex angle of fin	(rad)		
η_A	Enlargement ratio of heat transfer surface area	(–)		
λ	Thermal conductivity	(W m^{-1} K^{-1})		
μ	Dynamic viscosity	(kg m^{-1} s^{-1})		
ξ	Void fraction	(–)		
ρ	Density	(kg m^{-3})		
σ	Surface tension	(N m^{-1})		
X_{tt}	Lockhart-Martinelli's parameter	(–)		
Δz_H	Effective heat transfer length of subsection	(m)		
Δz	Axial length between neighboring pressure ports	(m)		
Φ_V	Two-phase multiplier factor	(–)		

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ACKNOWLEDGEMENT

The authors acknowledge gratefully Professor Akio Miyara of Saga University in providing his very useful experimental data.