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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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\textbf{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.

\textbf{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 \textit{et al.} (1995), Kedzieriski-Goncalves (1997), Yu-Koyama (1998), Shikazono \textit{et al.} (1998) and Goto \textit{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.

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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, \( \gamma \) is the vertex angle of fin, \( \beta \) is the helix angle of fins, \( n \) is the number of fins, and \( \eta_A \) 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>
<thead>
<tr>
<th>Table 1 Dimensions of Micro-fin Tube.</th>
</tr>
</thead>
<tbody>
<tr>
<td>(a) Present data</td>
</tr>
<tr>
<td>( d_o )</td>
</tr>
<tr>
<td>7.52</td>
</tr>
<tr>
<td>(b) Reference: Miyara's data</td>
</tr>
<tr>
<td>( d_o )</td>
</tr>
<tr>
<td>7.0</td>
</tr>
<tr>
<td>7.0</td>
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<tr>
<td>7.0</td>
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<tr>
<td>7.0</td>
</tr>
<tr>
<td>7.0</td>
</tr>
<tr>
<td>(c) Reference: Haraguchi's data</td>
</tr>
<tr>
<td>( d_o )</td>
</tr>
<tr>
<td>10.0</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Table 2 Experimental conditions. (a) Present data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tube Type</td>
</tr>
<tr>
<td>Refrigerant</td>
</tr>
<tr>
<td>( P_{in} ) (MPa)</td>
</tr>
<tr>
<td>( G ) (kg/(m²s))</td>
</tr>
<tr>
<td>( T_{in} ) (°C)</td>
</tr>
<tr>
<td>(b) Miyara's data</td>
</tr>
<tr>
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</tr>
<tr>
<td>( T_{in} ) (°C)</td>
</tr>
<tr>
<td>(c) Haraguchi's data</td>
</tr>
<tr>
<td>Tube Type</td>
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<tr>
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<tr>
<td>( P_{in} ) (MPa)</td>
</tr>
<tr>
<td>( G ) (kg/(m²s))</td>
</tr>
<tr>
<td>( T_{in} ) (°C)</td>
</tr>
</tbody>
</table>
3.1 Data Reduction for Frictional Pressure Drop

To obtain the fractional pressure drop through each subsection, the pressure drop due to momentum change, \( \Delta P_M \), is estimated from the following equation

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

where \( G \) is the mass velocity, \( x \) is the vapor quality, \( \rho \) is the density, \( \xi \) is the void fraction, and subscripts \( V \) and \( L \) denote vapor and liquid, respectively. The void fraction \( \xi \) 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.

\[
\xi = 0.81 \xi_{Smith} + 0.19 x^{100(\rho_v/\rho)^{0.05}} \xi_{Homo}
\]

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

Frictional pressure drop through each subsection, \( \Delta P_F \), is calculated from the measured static pressure drop, \( \Delta P_T \), and the pressure drop due to momentum change, \( \Delta P_M \), as

\[
\Delta P_F = \Delta P_T - \Delta P_M
\]

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

\[
\Phi_T = \frac{\Delta P_T}{\Delta x}, \quad X_H = \left( \frac{1-x}{x} \right)^{0.9} \left( \frac{P_V}{P_L} \right)^{0.5} \left( \frac{\mu_L}{\mu_V} \right)^{0.1}
\]

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

\[
\left( \frac{\Delta P_V}{\Delta x} \right) = -2 f_V \frac{G^2 x^2}{d \rho_V}
\]

where the friction coefficient, \( f_V \), is estimated by the Carnavos correlation proposed for single phase flow inside R133, Page 3

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

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

\[ \Phi_F = 1 + 1.2 \, Fr^{0.05} \, X^0.5 \]  

(7)

where \( Fr \) is the Froude number defined as

\[ Fr = \frac{G}{\sqrt{g \, d_1 \, \rho_f (\rho_L - \rho_f)}} \]  

(8)

where \( g \) is the gravity acceleration.

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(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

\[ Q = W_c C_{pc} \Delta T_c \]  

(9)

where \( W_c \) and \( C_{pc} \) are the mass flow rate and the specific isobaric heat of cooling water, and \( \Delta T_c \) is the temperature change of cooling water in each subsection. The heat transfer coefficient, \( \alpha \), and the Nusselt number, \( Nu \), are defined as

\[ \alpha = \frac{Q}{\pi d_1 \eta_A \Delta x (T_{sat} - T_{wi})} \]  

(10)

\[ Nu = \frac{\alpha d_1}{\lambda_L} \]  

(11)

where \( \eta_A \) is the area enlargement ratio, \( \Delta x \) 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

\[ Nu = \left( Nu_F^m + Nu_N^m \right)^{1/m} \]  

(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

\[ Nu_F = Re_L^* Pr_L \left[ T_1^* \right] \]  

(13)

where \( Re_L^* \) is the liquid Reynolds number, \( Pr_L^* \) is the liquid Prandtl number, \( T_1^* \) is the dimensionless temperature difference between the vapor-liquid interface and the tube wall, and their definitions are shown as
The wall shear stress $\tau_w$ in equations (14) and (15) can be expressed as

$$\tau_w = \frac{\Phi v^2 f_v}{2\rho_v}$$

(16)

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

$$Nu_F = \sqrt{\frac{f_v}{2} Re_L \Phi v} \left( \frac{\rho_L}{\rho_v} \right)^{0.5} \left( \frac{x}{1-x} \right) \left( \frac{Pr_L}{T_i^+} \right)$$

(17)

where $Re_L$ is the liquid Reynolds number defined as

$$Re_L = G(1-x)\frac{l_i}{\mu_L}$$

(18)

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

$$Nu_N = G(\eta_L)H(\xi)\left( \frac{Ga Pr_L}{Ph_L} \right)^{1/4}$$

(19)

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

$$Bo = \frac{(P-t)d_i g (\rho_L - \rho_v)}{\sigma}, \quad Ga = g \rho_L^2 d_i^3 \mu_L^2, \quad Ph_L = \frac{C_p L (T_{sat} - T_{wi})}{\Delta h_{LV}}$$

(20), (21), (22)

where $P$ is the pitch of micro-fin, $t$ is the width of micro-fin tip, $\sigma$ is the surface tension, and $\Delta h_{LV}$ is the latent heat. In equation (19), $G(\eta_L, Bo)$ 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 $\Phi v$ 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,
Table 3 Deviations between $Nu_{exp}$ and $Nu_{cal}$

<table>
<thead>
<tr>
<th>Experimental Data Source</th>
<th>Miyara</th>
<th>Haraguchi</th>
<th>Present study</th>
<th>Overall</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of original data</td>
<td>430</td>
<td>255</td>
<td>216</td>
<td>901</td>
</tr>
<tr>
<td>Residual</td>
<td>$Er A$</td>
<td>$Er B$</td>
<td>$Er A$</td>
<td>$Er B$</td>
</tr>
<tr>
<td>Present eqn.</td>
<td>0.050</td>
<td>0.086</td>
<td>-0.100</td>
<td>0.140</td>
</tr>
<tr>
<td>Cavallini et al.</td>
<td>0.003</td>
<td>0.330</td>
<td>-0.430</td>
<td>0.460</td>
</tr>
<tr>
<td>Kedzerski-Goncalves</td>
<td>-0.040</td>
<td>0.240</td>
<td>-0.280</td>
<td>0.290</td>
</tr>
<tr>
<td>Yu-Koyama</td>
<td>0.170</td>
<td>0.240</td>
<td>-0.060</td>
<td>0.140</td>
</tr>
<tr>
<td>Shikazono et al.</td>
<td>-0.590</td>
<td>0.620</td>
<td>-0.010</td>
<td>0.150</td>
</tr>
<tr>
<td>Goto et al.</td>
<td>-0.160</td>
<td>0.240</td>
<td>-0.250</td>
<td>0.260</td>
</tr>
</tbody>
</table>

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

where $n$ is number of experimental data.

$$H(\xi) = \xi + \left(10(1-\xi)^{0.1} - 8.9 \sqrt[3]{1-\xi}\right)$$

(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.

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

(24), (25), (26)

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

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

(27)

where

$$Nu_F = 2.12 \sqrt{f_p} \Phi_v \left( \frac{\rho_L}{\rho_v} \right)^{0.1} \left( \frac{x}{1-x} \right) Re_L^{0.5} Pr_L^{0.5}, \quad Nu_N = 1.98 \frac{H(\xi)}{\eta_A^{0.5} Bo^{0.1}} \left( \frac{Ga}{Pr_L \Phi_h} \right)^{0.25}$$

(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

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Subscript</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Bo$</td>
<td>Bond number</td>
<td>(-)</td>
<td></td>
</tr>
<tr>
<td>$d_i$</td>
<td>Average inner diameter of test Micro-fin tube</td>
<td>(m)</td>
<td></td>
</tr>
<tr>
<td>$c$</td>
<td>Water</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
Outer diameter of test Micro-fin tube \( d_o \) (m) \( \text{calc} \) Calculation
Friction coefficient \( f \) (–) \( \text{exp} \) Experiment
Fr \( Fr \) Froude number (–) \( F \) Forced convection
Refrigerant mass velocity \( G \) (kg m\(^{-2}\) s\(^{-1}\)) \( \text{in} \) Inlet
Gravitational acceleration \( g \) (m/s\(^2\)) \( L \) Liquid
Galileo number \( Ga \) (–)
Fin height \( h \) (m) \( \text{out} \) Outlet
Number of grooves \( n_g \) (–) \( \text{sat} \) Saturation
Nusselt number \( Nu \) (–)
Pressure \( P \) (Pa) \( wi \) Inside surface wall
Fin Pitch \( p \) (m) \( wo \) Outside surface wall
Phase change number \( Ph_L \) (–)
Prandtl number \( Pr \) (–)
Heat transfer rate of each test section \( Q \) (W)
Reynolds number \( Re \) (–)
Temperature \( T \) (K or \( ^\circ \)C)
Fin tip width \( t \) (m)
Vapor quality \( x \) (–)
Mass flow rate \( W \) (kg/h)
Heat transfer coefficient \( \alpha \) (W m\(^{-2}\) K\(^{-1}\))
Spiral angle \( \beta \) (rad)
Vertex angle of fin \( \gamma \) (rad)
Enlargement ratio of heat transfer surface area \( \eta_A \) (–)
Thermal conductivity \( \lambda \) (W m\(^{-1}\) K\(^{-1}\))
Dynamic viscosity \( \mu \) (kg m\(^{-1}\) s\(^{-1}\))
Void fraction \( \xi \) (–)
Density \( \rho \) (kg m\(^{-3}\))
Surface tension \( \sigma \) (N m\(^{-1}\))
Lockhart-Martinelli’s parameter \( X_H \) (–)
Effective heat transfer length of subsection \( \Delta x_H \) (m)
Axial length between neighboring pressure ports \( \Delta x \) (m)
Two-phase multiplier factor \( \Phi_r \) (–)

**REFERENCES**


Koyama, S., Chen, S., Kitano, R., Kuwashara, K., 2001, Experimental study on void fraction of two-phase flow


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