Journal of Mechanical Science and Technology (KSME Int. J.), Vol. 20, No. 8, pp. 1275~1283, 2006 1275

Condensation Heat Transfer Correlation for Smooth Tubes in Annular Flow Regime

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Condensation heat transfer coefficients in a 7.92 mm inside diameter copper smooth tube were obtained experimentally for R22, R134a, and R410A. Working conditions were in the range of 30-40℃ condensation temperature, 95-410 kg/m²s mass flux, and 0.15-0.85 vapor quality. The experimental data were compared with the eight existing correlations for an annular flow regime. Based on the heat-momentum analogy, a condensation heat transfer coefficients correlation for the annular flow regime was developed. The Breber et al. flow regime map was used to discern flow pattern and the Muller-Steinhagen & Heck pressure drop correlation was used for the term of the proposed correlation. The proposed correlation provided the best predicted performance compared to the eight existing correlations and its root mean square deviation was less than 8.7%.

Key Words : Smooth Tube, Condensation, Heat Transfer, Correlation, Convective

Nomenclature

A : Heat transfer surface area [m²]
c_p : Specific heat of fluid at constant pressure [J/kg K]
D : Tube diameter [m]
dP : Pressure drop [N/m²]
dz : Distance along the flow direction [m]
G : Mass flux [kg/m²s]
h : Heat transfer coefficients [W/m²K]
i : Enthalpy [J/kg]
k : Thermal conductivity of fluid [W/mK]
L : Test section length [m]
m : Mass flow rate [kg/s]
N : Total number of data [dimensionless]
Nu : Nusselt number [dimensionless]
Pr : Prandtl number [dimensionless]
Pr_τ : Turbulent Prandtl number [dimensionless]
Q : Heat transfer rate [W]
$\mathrm{Re}_{\mathrm{Eq}}$: Equivalent Reynolds number [dimensionless]
T : Temperature [℃]
u : Fluid average axial velocity [m/s]
u_τ : Turbulent friction or shear velocity [m/s]
x : Vapor quality [dimensionless]
y : Distance measured from the duct wall [m]

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y^+ : Wall coordinate [dimensionless]

Greek symbols

α : Thermal diffusivity [m^2/s]
β : Void fraction [dimensionless]
δ : Liquid film thickness [m]
Δ : Difference [dimensionless]
ε_h : Eddy thermal diffusivity for turbulent flow [m^2/s]
ε_m : Eddy kinematic viscosity for turbulent flow [m^2/s]
μ : Viscosity [kg/m s]
ν : Kinematic fluid viscosity [m^2/s]
ρ : Density [kg/m^3]
τ : Shear stress [N/m^2]

Subscripts

c : Critical
exp : Experimental
f : Refrigerant
g : Gas phase
i : Iinside
in : Inlet
l : Liquid phase
o : Outside
out : Outlet
pre : Pre-heater
pred : Predicted value
ts : Test section
w : Wall or water

1. Introduction

Due to the substitution of traditional refrigerants and the development of compact size heat devices, existing heat transfer correlations must be modified or new correlations should be proposed for new refrigerants with consideration of the characteristics of millimeter scale tubes. For smaller tubes, relative magnitudes of gravity, shear, and surface tension forces are significantly different from larger tubes and these differences play an important role. Most of promising alternatives are the mixture of two more refrigerants. Generally, the existing correlations do not show reliable prediction performances for the alternative refrigerants. According to recent studies, the Kosky & Staub correlation (1971) and the Dobson & Chato model (1998) were recommended by Cavallini et al.(2001) and Smit et al.(2002), respectively. Jung et al.(2004) compared six different models to their own experimental data and reported that those correlations showed similar predicted results. Above recent results show that it is needed to evaluate the existing correlations for alternative refrigerants in millimeter scale tubes. In this work, condensation heat transfer coefficients in a 7.92 mm inside diameter smooth tube were obtained experimentally for R22, R134a, and R410A. Acquired experimental data in the annular flow regime were compared with the eight existing correlations. A new condensation heat transfer correlation for the annular flow regime was proposed with the heat-momentum analogy.

2. Experiment

2.1 Experimental apparatus and procedures

Figure 1 shows the schematic diagram of the whole system. The experimental system was made up of two independent loops, i.e., a low temperature cooling water loop in annulus and a high temperature refrigerant loop inside the tube. The refrigerant loop was composed of a magnetic gear pump, a mass flow meter, a vapor quality control heater, and a chiller. The mass flow rate of the refrigerant was controlled by a magnetic gear pump and valves. A mass flow meter was located between a filter and a vapor quality control heater and measured the mass flow rate of the subcooled refrigerant. The subcooled refrigerant was heated by the vapor quality control heater to acquire a predetermined inlet vapor quality before entering the test section. Through the pre-tests with water, the amount of heat gained from the vapor quality control heater was correlated to the voltage input. The two-phase refrigerant entered the test section and was condensed. Afterwards, the refrigerant was subcooled by the chiller and went into a liquid receiver. The condensing pressures and temperatures of the refrigerants in the system were determined by adjusting the flow rates and the temperatures of the water in the chiller. Finally, the subcooled refrigerant was re-circulated through the refrigerant loop.

The cooling water loop was composed of a cooling water bath, a volumetric flow meter, and a pump. The cooling water was pumped to the circular-tube annulus. The heat of condensation was removed by constant temperature water from the temperature controlled water bath. The difference of temperature between the refrigerant and cooling water (averaged value of inlet and outlet temperatures) was maintained as a specific value (15± 0.5℃) during the experiments. The volumetric flow meter was used to measure the flow rates of the cooling water.

The test section is shown in Figure 2. The test section was a horizontal double-pipe heat exchanger. The refrigerant flowed in the inner tube and the cooling water flowed counter currently in the annulus removing the condensing heat of the refrigerant. The inner tube material is copper. The effective heat transfer length was 1000 mm. A 7.92 mm inside diameter copper tube and a 19 mm inner diameter acryl tube was used as a circular-tube annulus. A straight tube of 1700 mm in

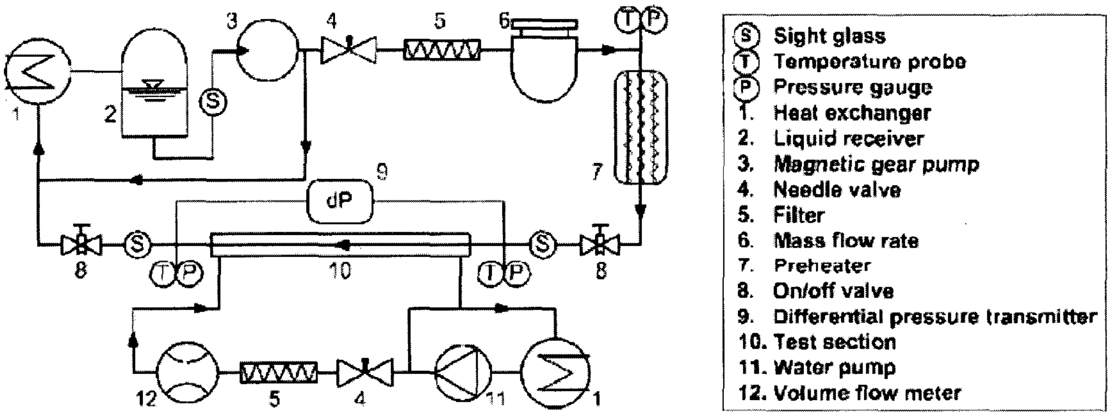

Fig. 1 Schematic diagram of test facility

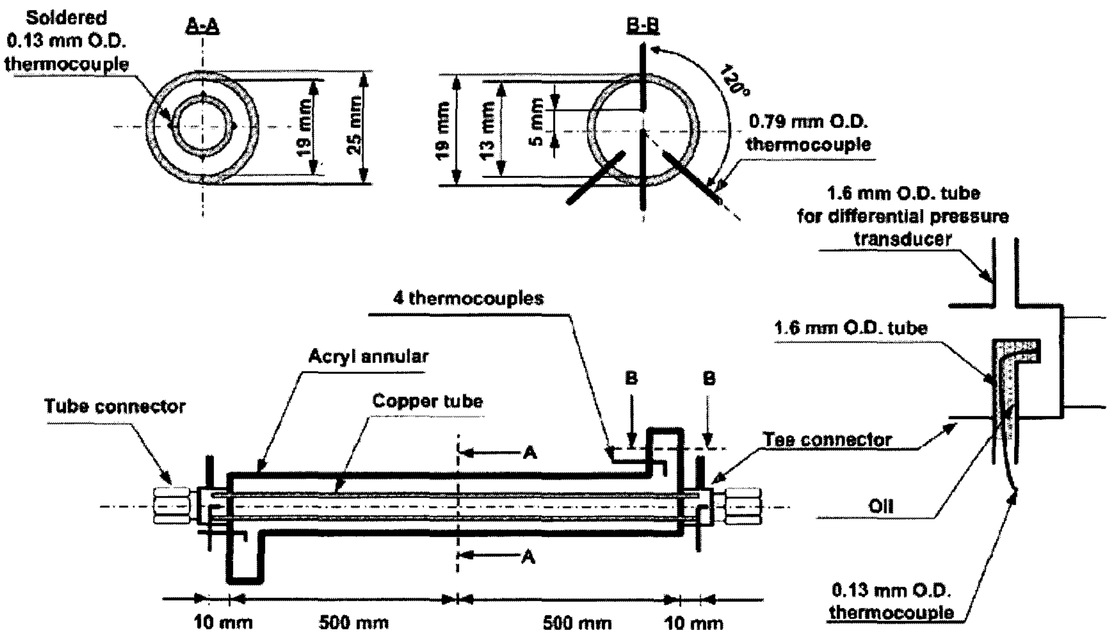

Fig. 2 Schematic diagram of test section

length was placed to stabilize the flow before the refrigerant entered the test section. The inlet and outlet temperatures of working fluids in the tube-in-tube heat exchanger were recorded by 4 T-type thermocouples. Another 4 T-type thermocouples were soldered on top, bottom, right and left sides of copper tube to measure the copper tube wall temperature. The location of 4 thermocouples was in the center of test section. The heat exchanger was well insulated with the glass fiber and rubber material of 15 cm radial thickness. For the pretest runs, the heat loss calculated from the energy balances between the inner side and the annulus narrows within 3% of the total heat transfer rate. The sight glasses were mounted at the inlet and outlet of the test section to visualize the flow. R22, R134a, and R410A were used as working fluids. Experimental data were obtained in the range of 95-410 (kg/m^2s), 0.15-0.85 vapor quality, and 30-40℃ condensation temperature. Uncertainties were $\pm 2.6\%$ for the mass flow rate of water, $\pm 2.3\%$ for the mass flux of refrigerant, $\pm 3.3\%$ for the heat flux of the test section, $\pm 4.1\%$ for the vapor quality, and $\pm 10.3\%$ for the local Nusselt number. Detail description for experimental apparatus and procedures can be found in reference (Han and Lee, 2005).

2.2 Heat transfer data reduction

By measuring the temperature and pressure in front of the vapor quality control heater (preheater) in subcooled state, enthalpy in front of the vapor quality control heater, i_{pre} (J/kg), could be obtained. Then, test section inlet enthalpy, $i_{ts,in}$ (J/kg), could be calculated from the heat input of the vapor quality control heater, Q_{pre} (W), and refrigerant mass flow rate, m_f (kg/s).

$$i_{ts,in} = i_{pre} + Q_{pre}/m_f \tag{1}$$

The condensing heat transfer rates in the test section, Q_{ts} (W), were obtained by the temperature difference in the cooling water, ΔT_w, and specific heat of fluid at constant pressure, $c_{p,w}$ (J/kg K).

$$Q_{ts} = m_w c_{p,w} \Delta T_w \tag{2}$$

Enthalpy at test section outlet, $i_{ts,out}$ (J/kg), was calculated from Eq. (3).

$$i_{ts,out} = i_{ts,in} - Q_{ts}/m_f \tag{3}$$

With the calculated enthalpy, test section inlet vapor quality, $x_{ts,in}$, was given by

$$x_{ts,in} = (i_{ts,in} - i_l)/i_{lg} \tag{4}$$

where, i_l is liquid phase enthalpy and i_{lg} is the enthalpy difference between liquid and vapor phase.

The change of refrigerant quality inside the tube, Δx, was evaluated from Eq. (5).

$$\Delta x = x_{ts,in} - x_{ts,out} = Q_{ts}/m_f \cdot i_{lg} \tag{5}$$

Then, the average of the quality in the test section, x_{ts}, was given by

$$x_{ts} = x_{ts,in} - \Delta x/2 \tag{6}$$

Average tube outside wall temperature, $T_{w,o}$, was obtained by taking the mean of the left, $T_{w,left}$ (K), right, $T_{w,right}$ (K), top, $T_{w,top}$ (K), and bottom, $T_{w,bottom}$ (K), wall temperatures.

$$T_{w,o} = (T_{w,top} + T_{w,bottom} + T_{w,left} + T_{w,right})/4 \tag{7}$$

Tube inside wall temperature, $T_{w,i}$ (K), could be obtained as follows

$$T_{w,i} = T_{w,o} + Q_{ts}\ln(D_o/D_i)/2\pi kL \tag{8}$$

where, D_o (m) is the tube outside diameter, D_i (m) is the tube inside diameter, k (W/m K) is the thermal conductivity of fluid, and L is the test section length.

Heat transfer coefficients, h (W/m^2 K), were calculated from heat transfer rate, the averaged value of inlet and outlet refrigerant temperature, T_f (K), and tube inside surface area, A (m^2).

$$h = Q_{ts}/A(T_f - T_{w,i}) \tag{9}$$

3. Results and Discussion

3.1 Evaluation of the existing correlations

Breber et al. (1980) proposed the simplest flow regime map which was dependent on the dimensionless vapor mass velocity and Lockhart-

Martinelli parameter (X_{tt}). With the Breber et al. flow regime map (Breber et al., 1980), heat transfer data in the annular flow regime were selected.

Table 1 shows the deviation of existing correlations. The r.m.s. means the root mean square deviation and the m.d. means the arithmetic mean deviations. Those deviations are defined as follows.

$$\text{m.d.}=\frac{1}{N}\sum\frac{data_{\text{pred}}-data_{\text{exp}}}{data_{\text{exp}}} \tag{10}$$

$$\text{r.m.s.}=\sqrt{\frac{1}{N}\sum\left(\frac{data_{\text{pred}}-data_{\text{exp}}}{data_{\text{exp}}}\right)^2} \tag{11}$$

Generally, the Tandon et al.(1995), the Dobson and Chato (19988), the Cavallini & Zecchin (1974), the Shah (1979), the Fujii (1995) correlations over predicted more than 15%. On the other side, the Akers et al.(1959) and the Kosky & Staub (1971) correlations slightly under predicted. The Kosky & Staub correlation (1971) showed the best predicting results among correlations. The Akers et al.(1959) correlation also showed relatively good predicting performance within 20% r.m.s. deviations.

Table 1 also shows the deviation for refrigerants. The Traviss et al. correlation (1972), Shah (1979), and Akers et al.(1959) showed the best results for R22, R134a, and R410A, respectively. For R22, The Cavallini & Zecchin (1974), the Dobson and Chato (1998), and the Tandon et al.(1995) correlations showed over predicted performance over 20%. The Akers et al.(1959)· and the Kosky & Staub (1971) correlations under predicted more than 10% for R134a. The most accurate model was different as refrigerant. Therefore, reliable condensation heat transfer correlation should be proposed.

3.2 Development of condensation heat transfer correlation

The annular flow regime is dominant flow pattern in condensers. Therefore, the prediction of heat transfer in the annular flow regime is significant to design heat exchangers. In this section, a semi-empirical condensation heat transfer correlation will be developed with the heat-momentum analogy. To apply the momentum-heat transfer analogy to the liquid layer, some reasonable assumptions were needed.

Assumptions

(1) The liquid film thickness is uniform around the tube periphery in the annular flow regime because of the greater influence of shear force.

(2) Since the vapor core is very turbulent, radial temperature gradients are neglected, and the temperatures in the vapor core and at the liquid-vapor interface are assumed to be equal to the saturation temperature.

(3) The flow in liquid phase is incompressible.

(4) The flow is steady.

Table 1 Deviation of correlations

Refrigerants	R22		R134a		R410A		Cavallini (2001)		Total	
Deviations (%)	m.d.	r.m.s.	m.d.	r.m.s.	m.d.	r.m.s.	m.d.	r.m.s.	m.d.	r.m.s.
Akers et al.(1959)	−7.9	9.7	−14.8	15.3	0.4	5.8	−5.0	15.9	−5.5	15.9
Shah (1979)	13.7	15.1	0.8	7.3	23.9	25.7	18.1	27.4	16.9	27.4
Traviss et al.(1972)	5.1	7.8	−4.6	8.8	12.3	17.0	11.6	23.1	9.9	23.1
Cavallini & Zecchin (1974)	21.9	23.3	8.5	12.2	33.7	34.8	25.4	33.0	24.6	33.0
Dobson & Chato (1998)	26.1	27.6	15.3	17.7	39.5	40.4	26.7	32.6	26.9	32.6
Kosky & Staub (1971)	−3.7	6.9	−11.6	13.3	3.7	7.8	−6.2	12.7	−5.5	12.7
Tandon et al.(1995)	55.2	56.9	50.3	51.6	43.2	44.0	51.6	53.9	51.0	53.9
Fujii (1995)	20.4	25.2	7.7	13.7	35.1	37.6	18.9	28.0	19.7	28.0
This work	−0.7	5.4	−0.9	7.1	1.1	5.9	−1.2	8.5	−0.9	8.5

(5) The liquid-vapor interface is smooth.

(6) Liquid droplet entrainment is neglected.

(7) The universal velocity distribution developed for steady and single-phase pipe flow is applicable to the annular film.

(8) Turbulent Prandtl number (Pr_τ) is equal to 0.85.

In the higher vapor velocity, there is appreciable entrainment of liquid. Physically this occurs because the vapor has a sufficiently high velocity to pick liquid up off the wall and transport it as droplets in the vapor core. At these high mass fluxes, the thickness of the liquid layer decreases due to entrainment, and consequently the heat transfer coefficient increases. According to Traviss et al. (1972), this effect is not as large as one might expect, because the main resistance to heat transfer occurs in the laminar sub-layer, and liquid removed from the turbulent zone does not increase the heat transfer coefficient in direct relation to the amount of liquid removed.

With the exception of high qualities and high mass fluxes, the liquid annulus is usually thicker at the bottom of the tube than at the top. If the flow is not too stratified, the overall effect of this asymmetry on heat transfer is nullified by the compensation effect of increased heat transfer in the upper half of the tube and decreased heat transfer in the lower half of the tube.

It is assumed that the momentum-heat transfer analogy is applicable to the liquid layer.

From the definition of eddy kinematic viscosity for turbulent flow, ε_m (m^2/s), and the viscosity coefficient, ν (m^2/s), the total apparent shear stress, τ (N/m^2), molecular plus turbulent, can be expressed as

$$\frac{\tau}{\rho}=(\nu+\varepsilon_m)\frac{\partial u}{\partial y} \tag{12}$$

where, ρ (kg/m^3) is the density, u is the fluid time average axial velocity (m/s), and y is the distance measured from the tube wall (m).

Heat flux can be written with temperature gradient, thermal diffusivity, α (m^2/s), and eddy thermal diffusivity for turbulent flow, ε_h (m^2/s).

$$\frac{Q}{A}=\rho c_p(\alpha+\varepsilon h)\frac{\partial T}{\partial y} \tag{13}$$

Then, introduce the non-dimensional liquid film thickness, δ^+ $(=u_\tau/\nu)$ and the non-dimensional flow velocity, $u^+(=u/u_\tau)$ where,

$$u_\tau=\sqrt{\frac{\tau_w}{\rho}} \tag{14}$$

$$\tau_w=\frac{D_i}{4}\left(\frac{dP}{dz}\right)_{fr} \tag{15}$$

Where, pressure gradient along the flow direction, $(dP/dz)_{fr}$ is obtained by using the Muller-Steinhagen & Heck pressure drop correlation (1986). Recently, Ould Didi et al. (2002) recommended the Muller-Steinhagen & Heck pressure drop correlation (1986). The liquid film thickness, $\delta(=(1-\beta)D_i/4)$ (m), was obtained from the Baroczy void fraction, β, correlation (1966). Koyama et al. (2004) recommended the Baroczy void fraction correlation (1966) by comparing with their experimental observation.

Then, Eq. (13) can be written as

$$\frac{1}{h}=\frac{T_f-T_{w,i}}{Q/A}=\int_0^{\delta^+}\frac{\nu}{(\alpha+\varepsilon_h)\rho c_p u_\tau}dy^+ \tag{16}$$

After the rearrangement of the Eq. (16) and Prandtl number,

$$\frac{1}{h}=\int_0^{\delta^+}\frac{dy^+}{(1/\mathrm{Pr}+\varepsilon_h/\nu)\rho c_p u_\tau} \tag{17}$$

Split the integral of Eq. (17) into two parts corresponding to the viscous sub-layer and fully turbulent region.

$$\frac{\rho c_p u_\tau}{h}=\int_0^{y_c^+}\frac{dy^+}{(\mathrm{Pr}^{-1}+\varepsilon_h/\nu)}+\int_{y_c^+}^{\delta^+}\frac{dy^+}{(\mathrm{Pr}^{-1}+\varepsilon_h/\nu)} \tag{18}$$

In the viscous sub-layer, ε_h/ν can be neglected and 1/Pr also be neglected in the fully turbulent region. ε_h can be expressed using the turbulent Prandtl number

$$\varepsilon_h=\varepsilon_m/\mathrm{Pr}_\tau \tag{19}$$

Eq. (12) can be rearranged for the fully turbulent region ($\varepsilon_m \gg \nu$).

$$\frac{\tau}{\rho}=\varepsilon_m\frac{\partial u}{\partial y} \tag{20}$$

Then,

$$\varepsilon_m=\frac{\tau}{\rho}\frac{\partial y}{\partial u}=\frac{\tau}{\rho}\frac{\nu}{u_\tau^2}\frac{\partial y^+}{\partial u^+} \tag{21}$$

For the fully turbulent flow, following universal velocity distribution which is called as the law of the wall can be used.

$$u^+=2.5\ln y^+ +5 \tag{22}$$

Thus du^+/dy^+ can be obtained as

$$\frac{du^+}{dy^+}=\frac{2.5}{y^+} \tag{23}$$

Then, Eq. (19) becomes

$$\frac{\varepsilon_h}{\nu}=\frac{\varepsilon_m}{\nu \mathrm{Pr}_\tau}=\frac{\tau}{\rho}\frac{1}{u_\tau^2}\frac{y^+}{2.5}\frac{1}{\mathrm{Pr}_\tau} \tag{24}$$

Because the liquid layer is very thin, assume that $\tau \approx \tau_w$ in the liquid layer, and then Eq. (24) can be reformed with Eq. (14) for the fully turbulent region.

$$\frac{\varepsilon_h}{\nu}=\frac{\varepsilon_m}{\nu \mathrm{Pr}_\tau}=\frac{y^+}{2.5}\frac{1}{\mathrm{Pr}_\tau} \tag{25}$$

With Eq. (25) for the fully turbulent region, Eq. (18) could be expressed as

$$h=\frac{\rho C_p u_\tau}{\mathrm{Pr} y_d^+ +2.125\ln(\delta^+/y_d^+)} \tag{26}$$

Kays and Crawford (1993) suggested that turbulent Prandtl number is 0.85 for the fully turbulent region.

Finally, y_c^+ was correlated by using Prandtl number, Pr, as follows

$$y_c^+=2.8552+5.6081/\ln \mathrm{Pr} \tag{27}$$

Deviations of proposed correlation were shown in Table 1. Regardless of refrigerant, this newly proposed correlation predicted well within ±10% r.m.s. deviation. Except R410A, this correlation showed the best predicting results. Total r.m.s. deviation was 8.5%. In Fig. 3, experimental Nusselt numbers and corresponding predicted values by the proposed correlation at specific flow condition are shown in terms of equivalent Reynolds number in annular flow regime. As stated above, experimental data in annular flow regime were selected by using Breber et al. (1980) flow pattern map. The definitions of Nu and $\mathrm{Re_{Eq}}$ are as follows

$$\mathrm{Nu}=\frac{hD}{k} \tag{28}$$

$$\mathrm{Re_{Eq}}=\frac{GD[(1-x)+x(\rho_l/\rho_g)^{0.5}]}{\mu_l} \tag{29}$$

Because $\mathrm{Re_{Eq}}$ increases with vapor quality and mass flux, it can be found that heat transfer coefficient (Nu) increases with vapor quality and mass flux in Fig. 3. Figure 3 also shows the predicted values by the proposed correlation at the identical experimental condition. By the comparison of the experimental data and the predicted values, it can be noticed that the correlation works properly according to variables like physical properties, refrigerants, mass flux, and vapor quality.

Figure 4 shows the comparison between predicted condensation heat transfer coefficients by this proposed correlation and experimental data. Nearly all data were placed within ±20%. Additionally, recent experimental data from Cavallini

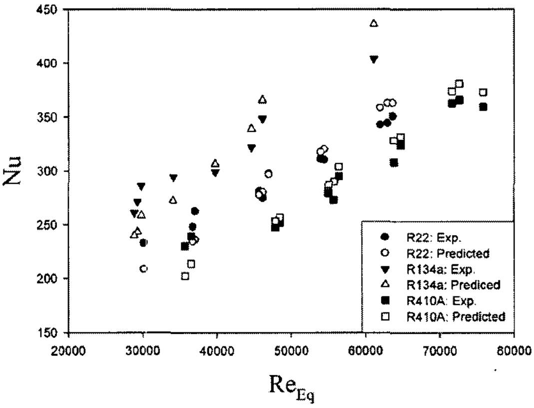

Fig. 3 Nu versus $\mathrm{Re_{Eq}}$ in annular flow regime

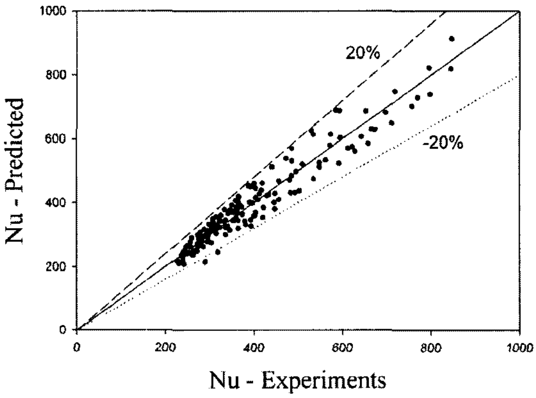

Fig. 4 Comparison between experimental data and predicted value

et al. (2001) were compared to correlations. The present correlation shows the best predictability

4. Conclusions

Condensation heat transfer coefficients in a 7.92 mm outside diameter smooth tube were obtained experimentally. R22, R134a, and R410A were used as working fluids. Experimental data were obtained in the range of 30-40℃ condensation temperature, 95-410 kg/m^2s mass flux, and 0.15-0.85 vapor quality. Experimental data were compared with the eight existing correlations for the annular flow regime.

Based on the heat-momentum analogy, a condensation heat transfer coefficients correlation in the annular flow regime was proposed. The Breber et al. flow regime map (1980) was used to classify the flow pattern and the Muller-Steinhagen & Heck pressure drop correlation (1986) was used to predict the pressure drop term in the correlation. Suggested correlation in this work was compared to experimental data and it showed improved predicting performance than the existing correlations. Its root mean square deviation was less than 8.5%.

Acknowledgments

This work was supported by the Korea Research Foundation Grant funded by the Korean Government (MOEHRD). (KRF-2004-214-D00020).

References

Akers, W. W., Deans, H. A. and Crosser, O. K., 1959, "Condensing Heat Transfer Within Horizontal Tubes," *Proc. Symp. Series Chem. Eng.*, Vol. 55, pp. 171~176.

Baroczy, C. J., 1966, "A Systematic Correlation for Two-Phase Pressure Drop," *Chemical Engineering Progress Symposium Series*, 62, pp. 232~249.

Breber, G., Palen, J. W. and Taborek, J., 1980, "Prediction of Horizontal Tube Side Condensation of Pure Components Using Flow regime Criteria," *J. Heat Transfer*, Vol. 102, pp. 471~476.

Cavallini, A. and Zecchin, R., 1974, "A Dimensionless Correlation for the Heat Transfer in Forced Convective Condensation," *Proc. 5th Int. Heat Transfer Conference*, Vol. 3, pp. 309~313.

Cavallini, A., Censi, G., Del Col, D., Doretti, L., Longo, G. A. and Rossetto, L., 2001, "Experimental Investigation on Condensation Heat Transfer and Pressure Drop of new HFC Refrigerants (R134a, R125, R32, R410A, R236ea) in a Horizontal Smooth Tube," *Int. J. of Refrigeration*, Vol. 24, pp. 73~87.

Dobson, M. K. and Chato, J. C., 1998, "Condensation in Smooth Horizontal Tubes," *J. Heat Transfer*, Vol. 120, pp. 193~213.

Fujii, T., 1995, "Enhancement to Condensing Heat Transfer-New Developments," *J. Enhanced Heat Transfer*, Vol. 2, pp. 127~138.

Han, D. and Lee, K. J., 2005, Experimental Study on Condensation Heat Transfer Enhancement and Pressure Drop Penalty Factors in four Microfin Tubes," *Int. J. of Heat and Mass Transfer*, Vol. 48, No. 18, pp. 3804~3816.

Jung, D., Cho, Y. and Park, K., 2004, "Flow Condensation Heat Transfer Coefficients of R22, R134a, R407C, and R410A Inside Plain and Microfin Tubes," *Int. J. of Refrigeration*, Vol. 27, pp. 25~32.

Kays, W. M. and Crawford, M. E. 1993, *Convective Heat and Mass Transfer*, chap. 13, McGraw-Hill, New York.

Kosky, P. G. and Staub, F. W., 1971, "Local Condensing Heat Transfer Coefficients in the Annular Flow Regime," *AIChE J.*, Vol. 17.

Koyama, S., Joodong, L. and Ryuuichirou, Y., 2004, "An Investigation on Void Fraction of Vapor-Liquid Two-Phase Flow for Smooth and Microfin Tubes with R134a at Adiabatic Condition," *Int. J. of Multiphase Flow*, Vol. 30, No. 3, pp. 291~310.

Müller-Steinhagen, H. and Heck, K., 1986, "A Simple Friction Pressure Drop Correlation for Two-Phase Flow in Pipes," *Chem Eng Process*, Vol. 20, pp. 297~308.

Ould Didi, M. B., Kattan, N. and Thome, J. R., 2002, "Prediction of Two-Phase Pressure Gradients of Refrigerants in Horizontal Tubes," *Int. J. of Refrigeration*, Vol. 25, pp. 935~947.

Shah, M. M., 1979, "A General Correlation for Heat Transfer During Film Condensation Inside Pipes," *Int. J. of Heat and Mass Transfer*, Vol. 22, pp. 547～556.

Smit, F. J., Thome, J. R. and Meyer, J. P., 2002, Heat Transfer Coefficients During Condensation of the Zeotropic Refrigerant Mixture HCFC-22/HCFC-142b," *J. Heat Transfer*, Vol. 124, pp. 1137～1146.

Tandon, T. N., Varma, H. K. and Gupta, C. P., 1995, "Heat Transfer During Forced Convection Condensation Inside Horizontal Tube," *Int. J. Refrigeration*, Vol. 18, pp. 210～214.

Traviss, D. P., Rohsenow, W. M. and Baron, A. B., 1972, "Forced Convection Condensation Inside Tubes: A Heat Transfer Equation for Condenser Design," *ASHRAE Trans.*, Vol. 79, pp. 157～165.