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Evaluation of Condensation Heat Transfer Correlations for Microfin Tubes

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Abstract

The feature of six existing condensation heat transfer correlations for microfin tubes were evaluated with the consideration of vapor quality, mass flux, geometries, and various refrigerants. The Kosky and Staub [15] and the Jaster and Kosky [16] correlations for smooth tube were used for the evaluation of the heat transfer enhancement factor (*EF*). For the prediction of zeotropic mixtures, most correlations show discrepancy with previous measurements. The Yu and Koyama [4] and the Shikazono et al. [8] correlations do not consider spiral angle effect. The Han and Lee [10] correlation shows fin height growth deteriorates heat transfer. Experimental verification to develop reliable condensation heat transfer correlation for microfin tubes is still needed with the consideration of geometrical effects and working conditions.

Key words: Microfin tube; Condensation; Heat transfer; Correlations; Geometrical effect

Nomenclature

c_p	specific heat of fluid at constant pressure [J/kg K]
D_i	maximum inside tube diameter [m]
e	fin height [m]
F	dimensionless parameter [-]
G	mass flux [kg/m^{2}s]
Ga	Galileo number [-]
H	function of void fraction [-]
h	heat transfer coefficients [W/m^{2}K]
n	number of fins [-]
Nu	Nusselt number [-]
P	pressure [Pa]
Ph	phase changer number [-]
Pr	Prantle number [-]
R	fin geometry parameter [-]
Re	Reynolds number [-]
Rx	enhancement factor [-]
T_c	condensation temperature [°C]
T^+	dimensionless temperature [-]
t_w^+	temperature difference in the groove [-]
x	vapor quality [dimensionless]

Greek symbols

β	spiral angle [°]
δ	liquid film thickness [m]
ε	surface area enhancement as compared to a smooth tube
Φ	two phase multiplier [-]
γ	apex angle of fins [°]
ρ	density [kg/m^3]
τ	shear stress [Pa]

Subscript

b	natural convective
cr	critical
e	based on equivalent diameter
Eq	equivalent
f	forced convective
h	based on hydraulic diameter
v	vapor
w	wall

1. Introduction

Microfin tubes have received a lot of interests because they provide heat transfer augmentation with

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marginal pressure drop increase. Microfin tubes are usually made of copper and have outside diameter between 4 to 15 mm, a single set of 50-70 spiral fins with spiral angle between 6 to 30°, and a fin height between 0.1 to 0.25 mm. The heat transfer augmentation is induced by heat transfer area enhancement, liquid film disturbance by fins, etc.

Since late 1990s, remarkable researches including the development of heat transfer models for microfin tubes have been performed and a couple of review papers [1, 2] have been published. The review papers only provide comparison results between measurements and correlations. Systematic evaluation of geometrical effects like fin height, spiral angle, and number of fins, and working conditions like vapor quality, mass flux, and refrigerants have not discussed. In this study, those upper mentioned parameters are evaluated with the previous measurements by other researchers. These results may be useful to improve the existing correlation by the correction of discrepancies and develop new models. Additionally, the characteristics of the existing correlations enable heat exchanger designers to select proper models in specific situations.

2. Condensation heat transfer coefficients correlations for microfin tubes

Cavallini et al. [1] developed condensation heat transfer coefficient (h) correlation for microfin tubes by modifying the Cavallini and Zecchin equation [3] for smooth tubes. Additional non-dimensional parameters with the effect of heat transfer area augmentation, inertia force, and surface tension were introduced. Yu and Koyama [4] modified the Haraguchi et al. [5] model for condensation in a smooth tube. They considered the simultaneous contribution of the forced convective effect and the natural convective effect on heat transfer mechanism. The void fraction is estimated by the Smith [6] correlation, which was originally developed for the smooth tube. Kedzierski and Goncalves [7] presented a correlation by the regression of their own data. They used the hydraulic diameter to account for the geometry effects. They consider a combination of liquid vapor interface mixing and turbulent mixing near the wall as important. However, surface tension drainage and swirl effects are considered as little influence on the heat transfer enhancement. Shikazono et al. [8] developed a general asymptotic analytical model. It has a similar form to that of the Yu and Koyama model [4]. Chamra et al. [9] developed a semi-empirical correlation for pure refrigerants. They modified the theoretical analysis of turbulent film condensation inside a smooth tube. Based on the heat-momentum analogy, Han and Lee [10] presented a correlation with their own pressure drop correlation for microfin tubes. The summary of the upper mentioned correlations are shown in table 1.

Table 1. The summary of the condensation heat transfer correlations for microfin tubes.

Cavallini et al. [1] [symbol: △]

$$\mathrm{Nu} = 0.05\,\mathrm{Re}_{Eq}^{0.8}\,\mathrm{Pr}_l^{1/3}\,R^2 F^{0.26}$$

Yu and Koyama [4] [symbol: ○]

$$\mathrm{Nu} = \left(\mathrm{Nu}_f^2 + \mathrm{Nu}_b^2\right)^{0.5}$$

$$\mathrm{Nu}_f = 0.152\left(0.3 + 0.1\,\mathrm{Pr}_l^{1.1}\right)\mathrm{Re}_{le}^{0.68}\,\frac{\Phi_v}{X_{tt}}$$

$$\mathrm{Nu}_b = 0.725\left(\frac{d}{d_e}\varepsilon\right)^{-1/4} H\left(\frac{Ga_e\,\mathrm{Pr}_l}{Ph}\right)^{1/4}$$

Kedzierski and Goncalves [7] [symbol: ▼]

$$\mathrm{Nu} = 2.256\,\mathrm{Re}_{h,l}^{0.303}\,Ph^{-0.232x}\,\mathrm{Pr}^{0.393}\left(\frac{P}{P_{cr}}\right)^{-0.578x^2}$$

$$\left[\log_{10}\left(\frac{P_{cr}}{P}\right)\right]^{-0.474x^2}\left[\frac{\rho_l - \rho_v}{x\rho_l - (1-x)\rho_v}\right]^{2.531x}$$

Shikazono et al. [8] [symbol: ●]

$$\mathrm{Nu} = \left[\mathrm{Nu}_f^2 + (f\mathrm{Nu})^2\right]^{0.5}$$

$$\mathrm{Nu}_f = 0.0152\left(1 + 0.6\,\mathrm{Pr}_l^{0.8}\right)\mathrm{Re}_{lr}^{0.77}\,\frac{\Phi_v}{X_{tt}}$$

$$\mathrm{Nu}_b = 0.725\left(\frac{d}{d_e}\varepsilon\right)^{-1/4}\left(\frac{Ga_e\,\mathrm{Pr}_l}{Ph}\right)^{1/4}$$

Chamra et al. [9] [symbol: ■]

$$h = \frac{0.208\rho_l c_p\left(\tau_w/\rho_l\right)^{0.224} Rx^{1.321}}{T^+}$$

Han and Lee [10] [symbol: □]

$$h = \frac{\rho_l c_p\left(\tau_w/\rho_l\right)^{0.5}}{t_w^+ + 2.1335\ln(\delta/e)}$$

3. Evaluation of correlations by previous researchers

Wang and Honda [2] compared existing condensation h correlations with their collected experimental data (six tubes with the tube inside diameter of 6.46-8.88 mm, the fin height of 0.16-0.24 mm, helix angle of groove of 12-20° and six refrigerants - R11, R123, R134a, R22, R410A), and reported that the Yu and Koyama [4] correlation showed the best predicting performance among empirical correlations.

Cavallini et al. [1] collected data from 18 different sources that covers the fin height of 0.15-0.635, the helix angle of groove of 0-30°, the diameter of 6.14-14.18 mm, and the mass flux of 80-910 kg/m^{2}s. Their data composed of nine different working fluids like R22, R134a, R12, R113, R407C, R32/R134a (30/70), R502, R125, R410A. According to their comparison, the Cavallini et al. [1] and the Yu and Koyama [4] show a mean absolute percentage deviation of about 35 and 39%, respectively.

García-Valladares [11] compared the correlations of Cavallini et al. [1], Kedzierski and Goncalves [7], and Yu and Koyama [4] with two different researchers' data set. He reported that additional work was needed to develop a generalized heat transfer correlation for microfin tubes.

Chamra et al. [9] compared their collected pure refrigerant (R22, R12, R134a) data with the Cavallini et al. [1], the Kedzierski and Goncalves [7], the Yu and Koyama [4], and their own correlation. They reported that their own correlation showed the best predicting performance. Their collected data composed of the outside diameter of 8-16 mm, the mass flux of 40-850 kg/m^{2}s, the fin height of 0.12-0.3 mm, and the helix of grooves of 7-30° from 17 different sources.

According to Han and Lee [10], their own correlation produced the best prediction for their experimental results. However, the best predicting correlation was varied as each tube type and refrigerant like their pressure drop comparison result.

4. Effects of geometry, working condition, and refrigerants on correlations

In this section, heat transfer coefficients (h) are compared in a similar way as was done for the pressure drop analysis. Fig. 1(a) to 1(c) show the effect of geometries on h. In Fig. 1(a), fin height effect on heat transfer is shown at 500 kg/m^{2}s. All correlations except the Han and Lee [10] produce an increasing tendency with fin height. Surface area increases with fin height. Therefore, the heat transfer coefficients may increase with fin height. However, there are conflicting reports for the effect of

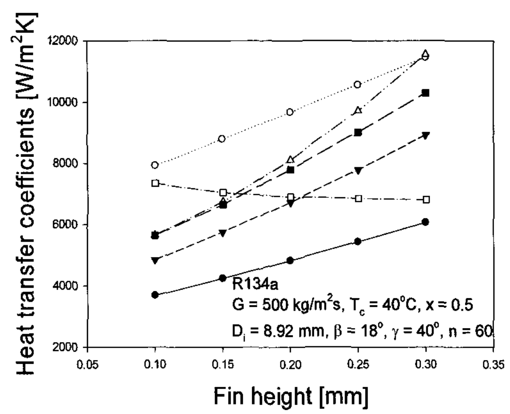

Fig. 1. (a) The effect of fin height on h.

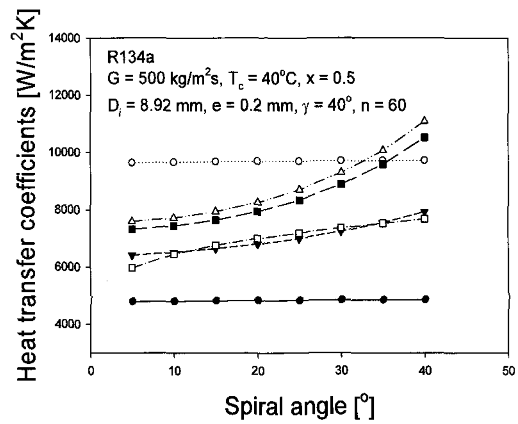

Fig. 1. (b) The effect of spiral angle on h.

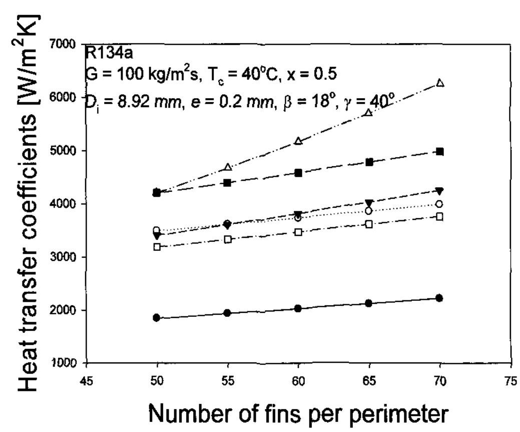

Fig. 1. (c) The effect of number of fins on h.

the fin height on the heat transfer coefficients according to Newell and Shah [12] review.

Figure 1 (b) shows the effect of spiral angle on the heat transfer coefficient at 500 kg/m^{2}s. All correlations except the Yu and Koyama [4] and the Shikazono et al. [8] produce increasing trend with spiral angle. Newell and Shah [12] reported that microfin tubes with smaller spiral angle had greater heat transfer than tubes with larger spiral angle. Oh and Bergles [13] reported that tubes with 18° spiral angle were the most effective for heat transfer enhancement at 50 kg/m^{2}s and 6° spiral angle showed the maximum heat transfer coefficients at 200 kg/m^{2}s. This implies that an optimum spiral angle exists for different mass flux.

The effect of the number of fins per perimeter is shown in Fig. 1 (c) at 100 kg/m^{2}s. The heat transfer surface area increases with increasing the number of fins. Therefore, the *h* may show the increasing tendency with the number of fins. All models show this predicted phenomenon.

The variation of *EF* with mass flux and vapor quality is demonstrated in Fig. 2 (a) and Fig. 2(b). The *EF* is defined as the ratio of *h* for the microfin tube and that for the smooth tube at the same working conditions (condensation temperature, mass flux, vapor quality) and maximum inside diameter. The *EF* includes heat transfer area augmentation. As smooth tube reference, the Kosky and Staub [14] and Jaster and Kosky [15] smooth tube heat transfer correlation was used for the annular flow regime and for the stratified flow regime, respectively. Cavallini et al. [16] recommended those two correlations by comparing the experimental data. Censi et al. [17] reported that the heat transfer enhancement factor of a micro-fin tube showed a convex tendency with mass flux. They obtained maximum heat transfer enhancement at 200 kg/m^{2}s in a 7.69 mm inside diameter with 0.23 mm fin height, 13° spiral angle, and 60 fins. After reaching the maximum value, the *EF* decreased with mass flux. This trend is appeared in all correlations. Especially, the Yu and Koyama [4] correlation shows the converging trend with mass flux.

Figure 2 (b) shows the variation of the *EF* with vapor quality at 500 kg/m^{2}s. Censi et al. [17] reported that the *EF* increased with vapor quality. Eckels and Tesene [18] reported that the *EF* decreased with vapor quality for higher mass fluxes (near 600 kg/m^{2}s) but increased with vapor quality for lower mass fluxes (near 250 kg/m^{2}s). The Shikazono et al. [8] correlation always predicts lower *h* values than those of smooth tube. Another interesting point is that the *EF* in lower vapor quality region is smaller than the area enhancement. A similar result was also reported by Censi et al. [17]

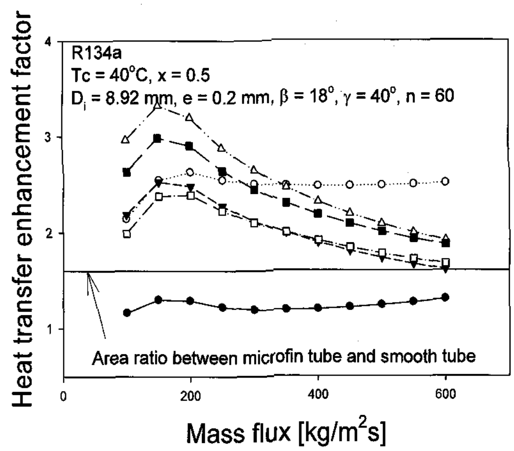

Fig. 2. (a) *EF* vs. *G*.

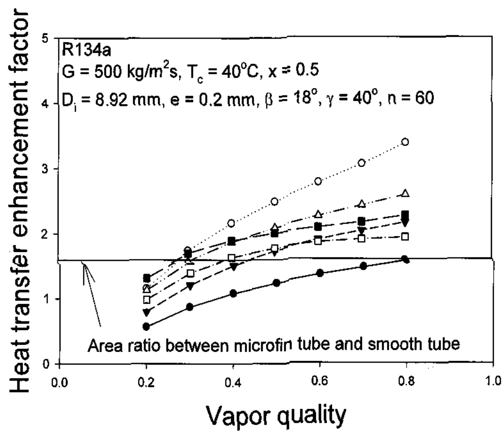

Fig. 2. (b) *EF* vs. vapor quality.

Figure 3 shows the comparison between the *h* of R410A and that of R22. Eckels and Tesene [18] reported that R134a had the highest heat transfer performance followed by R22, R410A, and R407C. Jung et al. [19] reported that *h* of R134a were similar to those of R22 while *h* of R407C and R410A were 23-53% and 10-21% lower than those of R22. Except Kedzierski and Goncalves [7] and Han and Lee [10] correlations, the others were developed for only pure refrigerants. Therefore, the Silver [20] and Bell and Ghaly [21] correction method were applied for the zeotropic mixtures (R410A and R407C) heat transfer coefficients of the Cavallini et al. [1], the Yu and Koyama [4], the Shikazono et al. [8], and the Chamra et al. [9] correlations. For R134a, all correlations except the Cavallini et al. [1] correlation match well with the previous studies. Their predicted results showed 10% lower *h* for R134a than those of R22.

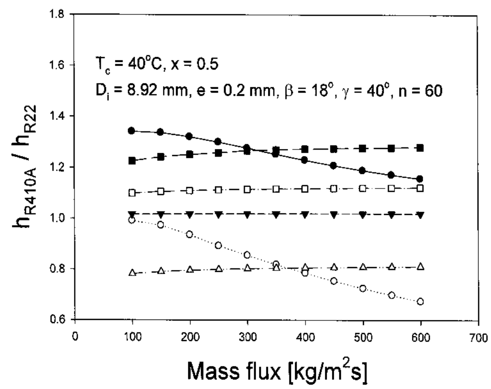

Fig. 3. (a) The comparison of *h* between R410A and R22.

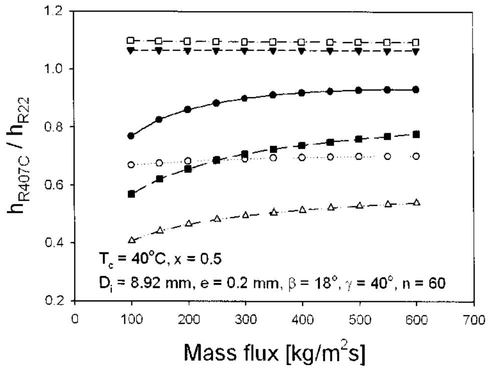

Fig. 3. (b) The comparison of *h* between R407C and R22.

For R410A, all correlations except the Yu and Koyama [4] and the Cavallini et al. [1] produce higher *h* for R410A than that of R22. The Kedzierski and Goncalves [7] produce almost same values. The Han and Lee [10] correlation shows 10% higher *h* for R410A than those for R22. The Shikazono et al. [8] and the Chamra et al. [9] correlations predict 15-30% higher results for R410A than R22. The ratios between R410A and R22 that are predicted by the Shikazono et al. [8] and the Yu and Koyama [4] correlations decrease with mass flux. For 407C, the Yu and Koyama [4] and the Cavallini et al. [1] correlation showed approximately 30% and 50-60% lower *h* for R407C than those of R22. Shikazono et al. [8] and Chamra et al. [9] produce 10-20% and 25-30% decreased values for R407C than those of R22. On the other hand, Kedzierski and Goncalves [7] and Han and Lee [10] correlations predict about 10% higher *h* for R407C than those for R22. The ratios between R407C and R22 that are predicted by Cavallini et al. [1], the Shikazono et al. [8], and the Chamra et al. [9] correlations increase with mass flux. The heat transfer coefficients of zeotropic mixtures can be qualitatively predicted by the Cavallini et al. [1] and the Yu and Koyama [4] models using Silver [20] and Bell and Ghaly [21] correction method.

5. Conclusions

The characteristics of existing condensation heat transfer correlations for microfin tubes were investigated with vapor quality, mass flux, geometries, and refrigerants.

The Han and Lee [10] heat transfer correlation produces decreasing heat transfer coefficients with fin height. The Yu and Koyama [4] and the Shikazono et al. [8] correlations did not consider the effect of spiral angle. All correlations do not show the characteristic of spiral angle that was reported by Oh and Bergles [13]. The Shikazono et al. [8] correlation always predicts lower heat transfer coefficients for microfin tubes than those of smooth tubes. All correlations embodied the mass flux effect on heat transfer enhancement factor successfully like the report by Censi et al. [17]. Even though the Shikazono et al. [8] and the Chamra et al. [9] correlations utilize the Silver [20] and Bell and Ghaly [21] correction method for the zeotropic mixtures, their prediction results do not show qualitative accordance with the trend of Eckels and Tescene [18] measurement. The Kedzierski and Goncalves [7] and the Han and Lee [10] correlation show 10% higher value for R407C than that of R22. The Cavallini et al. [1] model shows qualitative conformity for the prediction of zeotropic mixtures using the Silver [20] and Bell and Ghaly [21] correction method.

Because the physical understanding of microfin tubes is not enough, it is difficult to expect appropriate heat transfer correlations. It is natural that all correlations have some conflicting aspects to measurements. Therefore, systematic experimental investigations on the geometric characteristics and the effect of working condition like vapor quality, mass flux, and refrigerant should be accompanied to the development of condensation heat transfer model for microfin tubes.

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References

[1] Cavallini, A., Del Col, D., Doretti, L., Longo G. A. and Rossetto L., 2000, Heat transfer and pressure drop during condensation of refrigerants inside horizontal enhanced tubes, Int. J. of Refrig., Vol. 23, pp 4-25.

[2] Wang, H. S. and Honda, H., 2003, Condensation of refrigerants in horizontal microfin tubes: comparison of prediction methods for heat transfer, Int. J. Refrig., Vol. 26, pp. 452-460.

[3] Cavallini, A. and Zecchin, R. 1974, A dimensionless correlation for the heat transfer in forced convective condensation, 5th Int. Heat Transfer conference, Vol. 3, pp. 309-313.

[4] Yu, J. and Koyama, S., 1998, Condensation heat transfer of pure refrigerants in microfin tubes, proc. int. Refrig. Conf. at Purdue, pp. 325-330.

[5] Haraguchi, H., Koyama, S., Fujii, T., 1994, Condensation of refrigerants HCFC22, HFC134a and HCFC123 in a horizontal smooth tube, JSME Trans., Vol. 60, pp. 2111-2116.

[6] Smith, S. L., 1971, Void fraction in two-phase flow: A correlation based on equal velocity head model, Heat and Fluid flow, Vol. 1, pp. 22-39.

[7] Kedzierski, M. A. and Gonclaves, J. M., 1999, Horizontal convective condensation of alternative refrigerants within a micro-fin tube, J. Enhanced Heat Transfer, Vol. 6, pp. 161-178.

[8] Shikazono, N., Itoh, M., Uchida, M. Fukushima, T. and Hatada, T., 1998, Predictive equation proposal for condensation heat transfer coefficient of pure refrigerants in horizontal microfin tubes, Trans. JSME, Vol. 64, pp. 196-203.

[9] Chamra, L. M., Mago, P. J., Tan, M., and Kung, C., 2005, Modeling of condensation heat transfer of pure refrigerants in micro0fin tubes, Int. J. of Heat and Mass Transfer, Vol. 48, pp. 1293-1302.

[10] Han, D. and Lee, K. J., 2005, Experimental study on condensation heat transfer enhancement and pressure drop penalty factors in 4 microfin tubes, Int. J. of Heat and Mass Transfer, accepted.

[11] García-Valladares, O., 2003, Review of in0tube condensation heat transfer correlations for smooth and microfin tubes, Heat transfer engineering 24, no. 4, pp. 6-24.

[12] Newell, T. A., and Shah, R. K., 2001, An assessment of refrigerant heat transfer, pressure drop, and void fraction effects in microfin tubes, Int. J. of HVAC and R Research, Vol. 7, no. 2, pp. 125-153.

[13] Oh, S., and Bergles, A., 2002, Visualization of the effects of spiral angle on the enhancement of in-tube flow boiling in microfin tubes, ASHRAE Trans., Vol. 108, Part 2, pp. 509-515.

[14] Kosky, P. G. and Staub, F. W., 1971, Local condensing heat transfer coefficients in the annular flow regime, AIChE J. Vol. 17.

[15] Jaster, H., and Kosky, P. G., 1976, Condensation heat transfer in a mixed flow regime, Int. J. of Heat and Mass Transfer, vol. 19, pp. 95-99.

[16] Cavallini, A., Censi, G., Del Col, D., Doretti, L., Longo, G. A., and Rossetoo, L., 2001, Experimental investigation on condensation heat transfer and pressure drop of new HFC refrigerants (R134a, R125, R32, R410A, R236ea), Int. J. of Refrigeration, Vol. 24, pp. 73-87

[17] Censi, G., Doretti, L., Rossetto, L, and Zilio, C., 2003, Flow pattern visualization during condensation of R134a inside horizontal microfin and smooth tubes, Int. Congress of Refrigeration, Washington, D.C., USA.

[18] Eckels, S. J., and Tesene, B. A., 1999, A comparison of R-22, R-134a, R-410a, and R-407c condensation performance in smooth and enhanced tubes: Part I, Heat Transfer, ASHRAE Trans., Vol. 105, pp. 442-452.

[19] Jung, D., Cho, Y., and Park, K., 2004, Flow condensation heat transfer coefficients of R22, R134a, R407C, and R410A inside plain and micro-fin tubes, Int. J. of Refrigeration, Vol. 27, pp. 25-32.

[20] Silver, L., 1947, Gas cooling with aqueous condensation, Trans. Int. Chem. Eng. Vol. 25, pp. 30-42.

[21] Bell, K. J. and Ghaly, M. A., 1973, An approximate generalized design method for multicomponent/partial condenser, AIChE., Symp. Ser., Vol. 69. pp. 72-79.