Numerical Analysis of an Air-cooled Ammonia Condenser with Plate Fins

Youngil Kim* and Byung Ha Kang*

Key Words : Air-cooled, Ammonia, Condenser, Numerical analysis, Plate fin

Abstract

Ammonia has been used as refrigerant for more than 100 years in absorption as well as in compression systems. Due to its poisonous and inflammable properties, however, its use has been mainly on heavy industrial plants in which regular maintenance is available. For these systems, condensers are generally water-cooled. This is suitable for large systems over 20RT but is not suitable for small systems. In order to apply ammonia for a small system, it is important to adopt an air-cooled condenser. In this study, simple numerical analysis of an air-cooled condenser for an ammonia refrigeration system has been carried out. The condenser is designed as horizontal tubes with plate fins attached at the outer surface to enhance the air-side heat transfer rate. Effect of fin shape and arrangement are studied in detail. Since the local heat transfer coefficient is highest at the leading edge, heat flux is highest at the edge and decreases along the distance. Conditions of inlet air are also varied in the study and condenser length that is required for full condensation is calculated. The results show that it is important to enhance both the air-side and internal heat transfer coefficients.

Nomenclature

A : area, m^2
d : diameter, m
f_p : fin pitch, m
f_t : fin thickness, m
G : mass flux, kg/(m^2 · s)
h : heat transfer coefficient, W/(m^2 · K)
j : j Colburn factor, $j = \mathrm{St}\,\mathrm{Pr}^{2/3}$
k_f : fin thermal conductivity, W/(m · K)
k_t : tube thermal conductivity, W/(m · K)
L : fin length, m
ℓ : tube length, m
Nu : Nusselt number
P : pressure, kPa; perimeter, m
P_c : critical pressure, kPa

* Thermal/Flow Control Research Center, Korea Institute of Science and Technology, 39-1 Hawolgok-dong, Seongbuk-gu, Seoul 136-791, Korea

Pe : Peclet number
Pr : Prandtl number
Re : Reynolds number
S : source term
S_1 : vertical tube pitch, m
S_2 : horizontal tube pitch, m
St : Stanton number, St=Nu/(Re · Pr)
U_o : overall heat transfer coefficient, $W/(m^2 \cdot K)$
V : velocity, m/s; volume, m^3
x : x coordinate
y : y coordinate
η_f : fin efficiency
η_o : fin effectiveness

Subscript

h : hydraulic
i : internal
L : liquid
o : external
TP : two-phase

1. Introduction

Ammonia has been used as refrigerant for more than 100 years in absorption as well as in compression systems[1~5]. Due to its high operating pressure, poisonous and inflammable properties, however, its use has been mainly on heavy industrial plants in which regular maintenance is available. For these systems, condensers are generally water-cooled. This is suitable for large systems over 20RT but is not suitable for small systems. In order to apply ammonia for a small system, it is important to adopt an air-cooled condenser.

In order to develop high efficiency air-cooled heat exchangers, numerous studies on heat transfer characteristics have been carried by experiment and numerical simulation.[6] Kays and London[7] have compiled heat transfer and pressure drop correlations for a large number of fin-tube heat exchangers based on their measurements. For staggered tubes with plate fins, heat transfer and friction correlations are proposed by several researchers. For a separate fin, Briggs and Young correlation[8] is suggested for heat transfer, and Robinson and Briggs correlation[9] for pressure drop. For continuous fins, Gray and Webbs correlation[10] based on measurements is widely used. These correlations based on experiment are difficult to generate since they require time and costly experiment setups. Recently, numerical simulation for predicting the performance of heat exchangers is becoming popular. Since the geometry of heat exchanger is rather complex, numerical simulation is usually done with commercial softwares such as Star-CD, Phoenics, Flow-3D, Fluent and etc. These softwares are convenient but require a great deal of effort and time to learn and use. Besides, commercially available softwares have difficulties solving problems that involve phase changes.

In this study, simple numerical analysis of an air-cooled condenser for a 5RT ammonia refrigeration system has been developed. This method calculates the 2-dimensional temperature distribution of fins and can be applied to fins of other shapes. With this simple numerical method, the effect of fin pitch, fin thickness, tube pitch, tube diameter, air velocity and air temperature is studied.

In general, water-cooled condensers are designed as horizontal or vertical tubes with cooling water flowing inside the tubes. Condensation of refrigerant occurs at the outer side of the tube as heat is rejected to the cooling water flowing inside. In an air-cooled condenser, however, condensation occurs inside the tubes and heat is rejected to the ambient air which passes across the tubes.

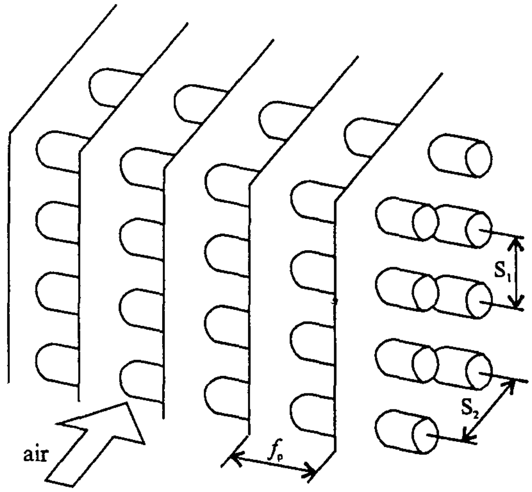

Fig.1 Air-cooled condenser

Since the inside condensation heat transfer coefficient is much greater than that of the air-side, thermal resistance occurs mainly on the outer surface. To increase the effective heat transfer area, plate fins are attached.

Air-cooled condenser chosen in this study is shown in Fig.1. Condensation of the working fluid occurs inside the tubes which are arranged horizontally and plate type fins are attached at the outer surface to increase the outer heat transfer area. Flow direction of air is also shown in the figure.

2. Analysis Model

For analysis, the heat exchanger is divided into two parts - outside and inside. On the outside, heat transfer occurs between air and fin/tubes and no phase change occurs.

Figure 2 shows the arrangement of tubes and plate fins. Air flows from left to right. Area for calculating temperature distribution is represented with hatched lines. Table 1 lists the heat exchanger specifications and conditions of air. Unless stated otherwise, default values in the table are used.

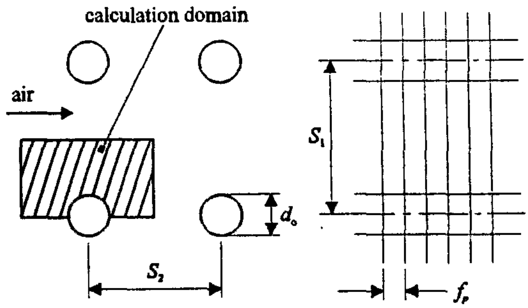

Fig.2 Arrangement of tubes and fins

Table 1 Simulation conditions

parameter	unit	default value	modified value
f_p	mm	4	2
f_t	mm	0.3	0.6
S_1 / S_2	mm	40 / 40	80 / 80
d_o / d_i	mm	14 / 10	20 / 16
V_{air}	m/s	1	5
T_{air}	℃	35	25

Overall heat transfer coefficient U_o between the fluid inside the tube and outer air can be represented as

$$\frac{1}{U_o}=\frac{A_o}{h_i A_i}+\frac{A_o \ln(d_o/d_i)}{2\pi k_t \ell}+\frac{1}{h_o} \tag{1}$$

$$h_o=\eta_o h_{air} \tag{2}$$

$$\eta_o=1-\frac{A_f}{A_o}(1-\eta_f) \tag{3}$$

$$\eta_f=\frac{\tanh(mL)}{mL} \tag{4}$$

$$m=\sqrt{\frac{h_{air}P}{k_f A}} \tag{5}$$

$$h_{air}=jGC_p\Pr^{-2/3} \tag{6}$$

External heat transfer coefficient h_o is obtained from the total heat transfer rate of the heat exchanger. Total heat transfer rate is the summation of heat transfer from the fins and base of the heat exchanger. This is calculated by integrating the convective heat transfer over the total fin and base area.

Heat transfer between air and fin becomes a conjugate heat transfer in which conduction and convection should be considered simultaneously. Considering these two effects, temperature distribution of the fin is calculated and the fin efficiency can be obtained from this result. When default values in Table 1 are used, the Reynolds number becomes 503. Therefore, the flow between two fins is assumed laminar. Laminar Nu number correlation[11] of a flow between two plates in which the effect of thermal entry length is considered, is given in Eqn. (7). Coefficients for Eqn. (7) are given in Table 2. Hydraulic diameter d_h is twice the distance between the fins.

$$Nu_x = \frac{\Sigma G_n \exp(-\lambda_n^2 x^+)}{\Sigma G_n/\lambda_n^2 \exp(-\lambda_n^2 x^+)} \tag{7}$$

$$x^+ = \frac{2x/d_h}{\text{Re Pr}} \tag{8}$$

Figure 3 shows the Nusselt number variation along the distance. The Nu number is large in the entry region in which the boundary layer is developing. At $x^+ = 0.03$, Nu number converges to a constant value. Substituting default values in Table 1 in Eqn. (8),

Table 2 Coefficients in Eqn. (7)

n	λ_n^2	G_n
0	15.09	1.717
1	171.3	1.239
2	498	0.952
$n \geq 3$	$\frac{16n}{\sqrt{3}} + \frac{20}{3\sqrt{3}}$	$2.68\lambda_n^{-1/3}$

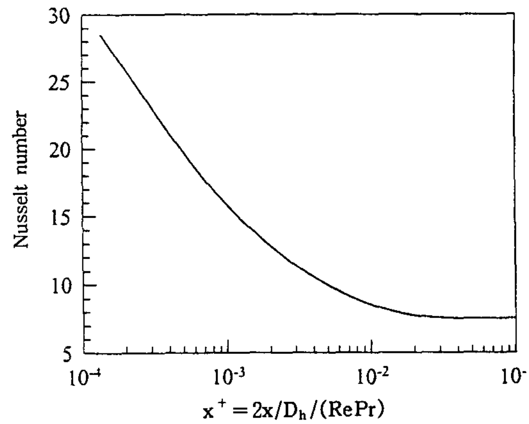

Fig.3 Nu number for a parallel plate

x becomes 42mm from the tip and after this length, heat transfer coefficient becomes a constant small value.

Flow around the tube is assumed a flow around a cylinder and Churchill and Bernstein correlation[12] is used.

For $10^2 < \text{Re} < 10^7$ and $\text{Pe} > 0.2$

$$\text{Nu} = 0.3 + \lambda \left[1 + \left(\frac{\text{Re}}{282000}\right)^{5/8}\right]^{4/5} \tag{9}$$

For $2\times10^4 < \text{Re} < 4\times10^5$ and $\text{Pe} > 0.2$

$$\text{Nu} = 0.3 + \lambda \left[1 + \left(\frac{\text{Re}}{282000}\right)^{1/2}\right] \tag{10}$$

where $\lambda = \dfrac{0.62\ \text{Re}^{1/2}\ \text{Pr}^{1/3}}{\left[1 + \left(\frac{0.4}{\text{Pr}}\right)^{2/3}\right]^{1/4}}$

Due to conduction and convection heat transfer, temperature distribution exists in the fin. Boundary which is in contact with the tube is assumed constant temperature while other boundaries are assumed symmetrical. For this steady 2-dimensional problem, the governing equation becomes

$$\frac{\partial}{\partial x}\left(k_f \frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k_f \frac{\partial T}{\partial y}\right) + S = 0 \tag{11}$$

S denotes heat generation due to convection.

$$S = h_x \Delta A(T_{air} - T)/\Delta V \qquad (12)$$

Convective heat transfer h_x is a function of distance and Eqn. (12) is solved by finite difference method.[13]

Ammonia flows inside the tube with phase change. For thermodynamic properties of ammonia, correlation by Harr and Gallagher[14] is used. For transport properties, correlations are derived by curve fitting the table values given by Liley and Desai.[15] Ammonia enters the condenser as superheated state (57.6℃, 16.5bar) and the temperature is decreased as desuperheating occurs. As the temperature reaches the saturation temperature, condensation begins to occur. Subcooling follows after completion of the condensation process. The whole process is shown in Fig.4. The pressure drop in the condenser is assumed negligible.

Desuperheating and subcooling processes

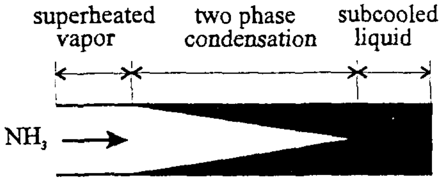

Fig.4 Condensation model

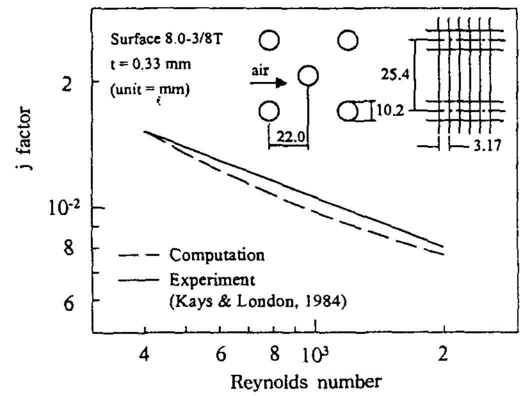

Fig.5 j factor for a plate fin

are assumed as a single phase flow. For a laminar flow, Nu=4.363 is used while for a turbulent flow, well known Dittus and Boelter correlation is used.

$$Nu = 0.023\ Re^{0.8}\ Pr^{0.4} \qquad (13)$$

For a two-phase flow, Shah's correlation is used.[16]

$$h_L = 0.023\, Re_L^{0.8}\, Pr_L^{0.4} \frac{k_L}{d} \qquad (14)$$

$$h_{TP} = h_L \left[X^{0.8} + \frac{3.8\, x^{0.76}\, X^{0.04}}{(P/P_c)^{0.38}} \right] \qquad (15)$$

where $X = 1 - x$.

3. Results and Discussions

Numerical results are compared with the experimental results by Kays and London[7] in Fig.5. Shape, dimension and arrangement of fins are shown in the figure. Eqn. (2) through (6) are used when converting calculated results to j-factors. Numerical results are smaller than those of the experiment but overall, they are in good agreement.

Figure 6 shows the temperature distribution of the fin. Temperature drop is large at the leading edge where the heat transfer coefficient is largest. Minimum temperature is observed at the upper left corner where the

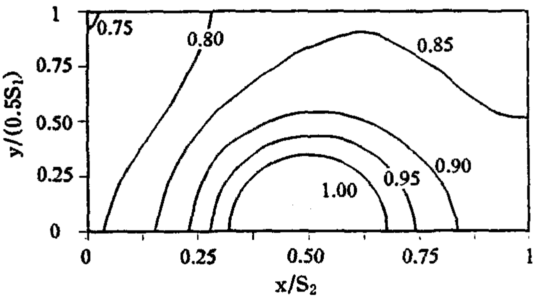

Fig.6 Temperature contour lines

heat transfer coefficient is greatest and the distance farthest from the tube. Fig.7 shows the heat transfer rate per unit area divided by the average value. The values are large near the leading edge and the maximum occurs at the lower left corner.

Figure 8 shows the effect of air velocity on Nu number and percentage of heat removal by fin. Fin efficiency increases as air velocity increases but due to noise and pressure drop, air velocity is usually constrained to a certain limit. For an air-cooled condenser, the facial air velocity is usually designed around 3m/s. In this case, the heat transfer from the fins is more than 90%. Therefore in order to achieve high performance, fin design and good maintenance of fins to minimize fouling are important.

Figure 9 shows that by decreasing the fin pitch from 4mm to 2mm, the condenser length is decreased significantly. This increases the heat exchanger efficiency without increasing the overall heat exchanger size but the pressure drop on the air-side will increase. Figure 10 shows the effect of fin thickness. By increasing the thickness from 3mm to 6mm, the performance is increased only slightly.

Figure 11 shows that by increasing the fin area, the required condenser length is decreased. This will, however, increase the overall size. In Fig.12, it shows that tube diameter has little effect on the performance. As the tube diameter gets smaller, the internal heat transfer coefficient is increased due to the increase of Reynolds number but the air-side heat transfer coefficient is decreased. These two factors offset each other somewhat. Internal pressure drop is increased as tube diameter gets smaller.

Air velocity has a significant effect on condenser length as shown in Fig.13. Increased air velocity gives better performance of the heat exchanger but both fan power and noise will increase. Figure 14 shows that as air temperature is decreased from 35℃ to 25℃,

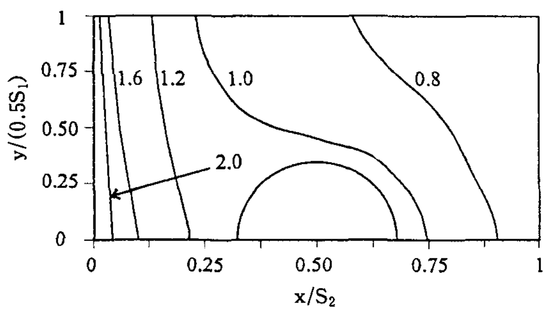

Fig.7 Heat flux contour lines

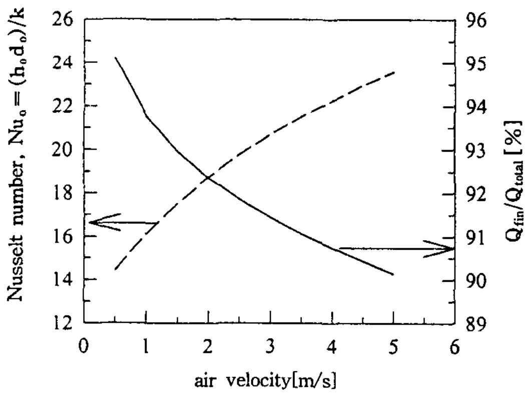

Fig.8 Effect of air velocity

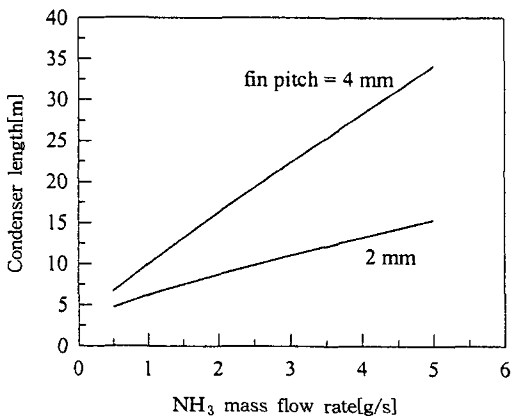

Fig.9 Effect of fin pitch

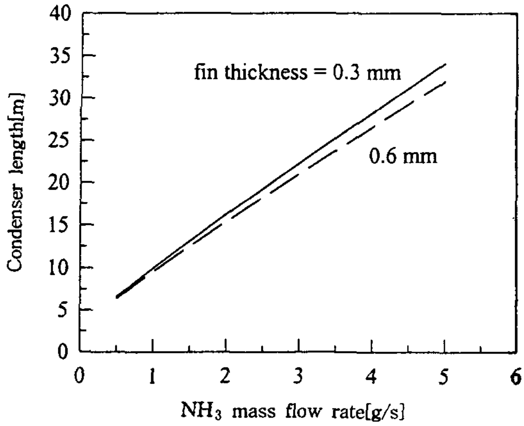

Fig.10 Effect of fin thickness

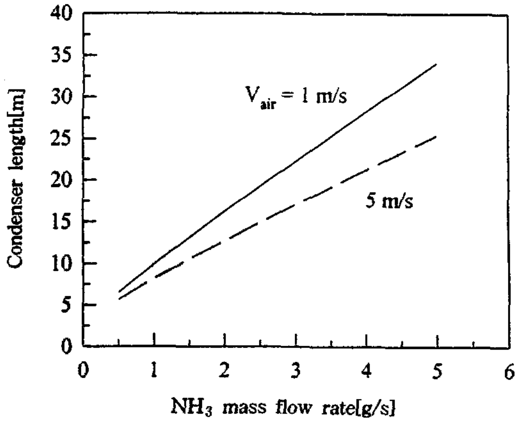

Fig.13 Effect of air velocity

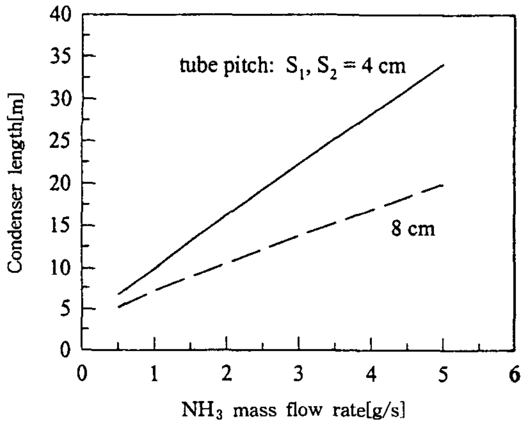

Fig.11 Effect of tube pitch

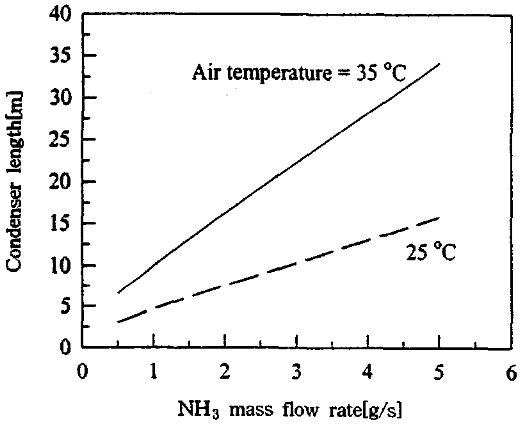

Fig.14 Effect of air temperature

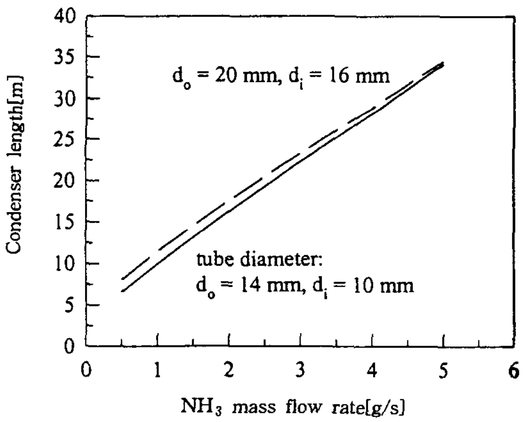

Fig.12 Effect of tube diameter

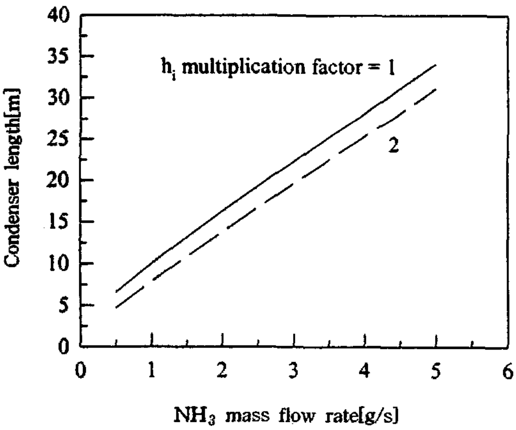

Fig.15 Effect of internal heat transfer coefficient

the condenser length is decreased significantly.

Recently, numerous studies are being done to increase the internal heat transfer coefficient.[6] In this particular case, condensation heat transfer is 67 times greater than that of the air-side. The air-side heat transfer area is about 24 times greater than that of the internal side. Therefore, internal overall heat transfer coefficient is about 2.8 times greater. Figure 15 shows the effect of internal heat transfer coefficient. If the internal heat transfer coefficient is doubled, condenser length is decreased 29% at 0.5g/s, and 9% at 5g/s. Overall performance is affected more in the low mass flow rate case since the internal heat transfer coefficient is small due to the small Re number.

4. Conclusions

Simple numerical analysis to predict the performance of an air-cooled condenser for an ammonia refrigeration system has been developed. This method has advantages with respect to time and cost when compared to experiment and commercial CFD softwares.

Fin efficiency is better when closer to the leading edge and this should be incorporated when designing fins. Condenser length depends significantly on fin pitch, tube pitch, air velocity and temperature. Tube pitch is related to the heat exchanger size. Fin pitch and air velocity are related to pressure drop and noise. Fin thickness and tube size have little effect on the condenser performance. Internal heat transfer coefficient becomes an important factor if the air-side area is increased substantially by attaching fins.

References

(1) Bogart, M., 1981, Ammonia Absorption Refrigeration in Industrial Processes, Gulf Publishing Company, Houston.

(2) Gosney, W. B., 1982, Principles of Refrigeration, Cambridge University Press, Cambridge.

(3) Alefeld, G. and Radermacher, R., 1994, Heat Conversion Systems, CRC Press, Boca Raton.

(4) Herold, K. E., Radermacher, R. and Klein, S. A., 1996, Absorption Chillers and Heat Pumps, CRC Press, Boca Raton.

(5) Lorentzen, G., 1988, "Ammonia : an Excellent Alternative", Int. J. Refrigeration, Vol. 11, pp. 248~252.

(6) Webb, R. L., 1994, Principles of Enhanced Heat Transfer, John Wiley & Sons, New York.

(7) Kays, W. M. and London, A. L., 1984, Compact Heat Exchangers, 3rd ed., McGraw-Hill, New York.

(8) Briggs, D. E. and Young, E. H., 1963, "Convection Heat Transfer and Pressure Drop of Flowing Across Triangular Pitch Banks of Finned Tubes", Chem. Eng. Progress Sym. Series, Vol. 59, No. 41, pp. 1~10.

(9) Robinson, K. K. and Briggs, D. E., 1966, "Pressure Drop of Air Flowing Across Triangular Pitch Banks of Finned Tubes", Chem. Eng. Progress Sym. Series, Vol. 62, No. 64, pp. 177~184.

(10) Gray, D. L. and Webb, R. L., 1986, "Heat Transfer and Friction Correlations for Plate Fin-and-Tube Heat Exchangers having Plain Fins", Proceedings of the 9th International Heat Transfer Conference, San Francisco, pp. 2745~2750.

(11) Kays, W. M. and Crawford M. E., 1993, Convective Heat and Mass Transfer, 2nd ed., McGraw-Hill, New York.
(12) Churchill, W. W. and Bernstein, M., 1977, "A Correlation Equation for Forced Convection from Gases and Liquids to a Circular Cylinder in Crossflow", J. Heat Transfer, Vol. 99, pp. 300~306.
(13) Patankar, S. V., 1991, Computation of Conduction and Duct Flow Heat Transfer, Innovative Research, Inc., Minnesota.
(14) Harr, L. and Gallagher, J., 1978, "Thermodynamic Properties of Ammonia", J. Phys. Chem. Ref. Data, Vol. 7, No. 3, pp. 635~792.
(15) Liley, P. E. and Desai, P. D., 1993, Thermophysical Properties of Refrigerants, SI ed., ASHRAE, Atlanta, Georgia.
(16) Shah, M. M., 1979, "A General Correlation for Heat Transfer during Film Condensation inside Pipes", Int. J. Heat and Mass Transfer, Vol. 22, pp. 547~556.