Journal of Mechanical Science and Technology (KSME Int. J.), Vol. 19, No. 10, pp. 1957~1963, 2005 1957

Condensation Heat Transfer Coefficients of Flammable Refrigerants on Various Enhanced Tubes

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In this study, external condensation heat transfer coefficients (HTCs) of six flammable refrigerants of propylene (R1270), propane (R290), isobutane (R600a), butane (R600), dimethylether (RE170), and HFC32 were measured at the vapor temperature of 39℃ on a 1023 fpm low fin and Turbo-C tubes. All data were taken under the heat flux of 32~116 and 42~142 kW/m^2 for the low fin and Turbo-C tubes respectively. Flammable refrigerants' data obtained on enhanced tubes showed a typical trend that external condensation HTCs decrease with increasing wall subcooling. HFC32 and DME showed up to 30% higher HTCs than those of HCFC22 due to their excellent thermophysical properties. Propylene, propane, isobutane, and butane showed similar or lower HTCs than those of HCFC22. Beatty and Katz' correlation predicted the HTCs of the flammable refrigerants obtained on a low fin tube within a mean deviation of 7.3%. Turbo-C tube showed the best performance due to its 3 dimensional surface geometry for fast removal of condensate.

Key Words: External Condensation, Flammable Refrigerants, Hydrocarbons, Dimethylether, HFC32, Enhanced Tubes

Nomenclature

A : Area [m^2]
D : Diameter [m]
F : Property factor
fpi : Fins per inch
g : Gravitational acceleration [m/s^2]
h : Heat transfer coefficient [W/m^2K]
h_{fg} : Heat of evaporation [kJ/kg]
k : Thermal conductivity [W/m·K]
L : Length [m]
T : Temperature [℃ or K]
ΔT : Temperature difference [℃ or K]

Greek symbols

η : Fin efficiency
μ : Dynamic viscosity [Pa·s]
ρ : Density [kg/m^3]

Subscripts

B&K : Beatty and Katz
c : Characteristic
exp : Experimental
f : Saturated liquid phase or fin
g : Saturated vapor phase
Nusselt : Nusselt
o : Outside diameter
pre : Predicted
r : Root of fin

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1. Introduction

For the past decade, CFCs have been phased out due to ozone layer depletion and various alternative refrigerants have been proposed. Even though flammable refrigerants are environmentally friendly, they have not been used in normal refrigeration and air-conditioning applications because of a safety concern. These days, however, this trend becomes somewhat relaxed on account of an environmental mandate for clean technology. Thus, some of the flammable refrigerants have been applied to certain applications as a pure working fluid or as one of the components of mixed working fluids (Kruse, 1996; Jung et al., 2000). For the past decade, isobutane (R600a) has dominated the European refrigerator/freezer sector and is being used in Japan and Korea as well at this time (Kruse, 1996). At present, propane (R290) and propylene (R1270) are also proposed and actually used as working fluids in heat pumps for heating applications in Europe (IEAHPC, 2002). In fact, pure and mixed hydrocarbons offer such advantages as a low cost, availability, compatibility with the conventional mineral oil, and environmental friendliness (Kruse, 1996; IEAHPC, 2002). Jung et al.(1999b) showed that dimethylether (DME, RE170) is also a good refrigerant to replace CFC12. Even though DME is flammable, it has good thermodynamic properties as well as good compatibility with mineral oil and also offers excellent environmental properties of ODP=0 and GWP=5. Finally, HFC32 is commonly used these days as one of the components of some refrigerant mixtures such as R407C and R410A which are the popular alternatives for HCFC22.

For all refrigeration equipment, heat exchangers are the basic components and thus their efficiency should be increased for the overall improvement of the system efficiency. In this study, particularly condensation heat transfer characteristics of enhanced tubes which are currently used in the condensers of centrifugal and reciprocating chillers are studied experimentally with flammable refrigerants.

In general, tubes of coated and machined surfaces have been used as a means to increase heat transfer coefficients (HTCs) of various heat exchangers (Webb, 1994). Especially, low fin tubes of 2-dimensional (2-D) fin geometry were developed in the late 1940's and since then have been widely used in shell and tube type heat exchangers. In 1948, Beatty and Katz (1948) developed a correlation to predict the HTCs of low fin tubes for the first time. Their correlation was based upon the assumption that the condensate is not retained in between the fins and the surface tension force can be neglected with only the gravitational effect considered.

In 1980, Carnavos (1980) determined the optimum fin density for low fin tubes using CFC11 as a working fluid and in 1985, Yau et al.(1985) performed experiments with low fin tubes of various fin heights and space using CFC12 as a working fluid. In 1985, Rudy and Webb (1985) showed that the actual heat transfer area is reduced significantly due to the retention of the condensate in low fin tubes. In 1995, Kim et al. (1995) performed experiments using three low fin tubes with CFC11 and demonstrated that the retention of the condensate increases with the fin height. In 1996, Joo et al.(1997) measured HTCs of low fin tubes of various fin densities using CFC11 and its two alternatives and concluded that the low fin tube of 28 fins per inch (fpi) has the highest HTCs for all the fluids tested.

Recently, some enhanced tubes of 3 dimensional (3-D) fin geometry have been studied. Especially, the performance of a Turbo-C tube of a saw tooth type 3-D fins has mainly been examined. For example, in 1990, Webb and Murawski (1990) performed experiments on various enhanced tubes including a Turbo-C tube and a 26 fpi low fin tube using CFC11 and showed that the Turbo-C tube has the highest HTCs among all the tubes tested. In 2003, Jung et al.(2003) measured condensation HTCs of R22, R407C, and R410A on a low fin and Turbo-C tubes and found that the performance of Turbo-C tube is the best for R22 alternatives.

As revealed in the literature survey, most of the previous condensation heat transfer studies were done with such working fluids as water, n-pentane, and ozone depleting CFCs that are phased out by the Montreal protocol. Even though some recent studies examined the performance of HCFCs and HFCs, few data are available for the external condensation heat transfer characteristics of flammable refrigerants on enhanced tubes. Therefore, condensation heat transfer characteristics of a low fin and Turbo-C tubes will be examined in this study using propylene, propane, DME, isobutane, butane and HFC32 as working fluids. These data will prove to be helpful for the design of more energy efficient chillers using alternative refrigerants in the future.

In fact, this is a sequel paper to our previous works for the condensation heat transfer with CFC11, CFC12, HCFC22 alternatives on a horizontal plain, low fin and Turbo-C tubes (Jung et al., 1999a ; Jung et al., 2003) and with flammable refrigerants on a horizontal plain tube (Jung et al., 2004).

2. Experiments

In this work, external condensation HTCs of 6 flammable refrigerants are measured at the vapor temperature of 39℃ on a low fin tube of 1024 fins per meter (26 fins per inch) and a Turbo-C tube of nominal outside diameter of 18.9 mm using the same experimental apparatus with the same tube specimens described in Jung et al.(1999a ; 2003). Since Jung et al.(1999a) contains all the details of the test apparatus, tube specimen and its manufacture, measurements, experimental procedure, data reduction scheme, fouling effect, repeatability of data, data verification etc., they will not be presented again here. An interested reader is referred to Jung et al.(1999a) for the details. In this work, only brief description of the test facility, tube specifications, and uncertainties in the measurements will be reported.

Figure 1 shows the schematic diagram of the experimental apparatus. The facility is composed of the refrigerant and cooling water loops. The refrigerant vapor supplied to the test section was generated by an immersion heater of 3.5 kW in the boiler that was located at the bottom of the apparatus. The vapor generated was fed to the main test section through a connecting pipe and condensed via counter-current heat exchange with the cooling water flowing inside a test tube. The condensate as well as the uncondensed vapor went into a large capacity auxiliary condenser and were cooled there and finally returned to the bottom of the boiler. The cooling water for the test section and for the auxiliary condenser was supplied by two independent external chillers that were capable of controlling the temperature with an accuracy of 0.1℃.

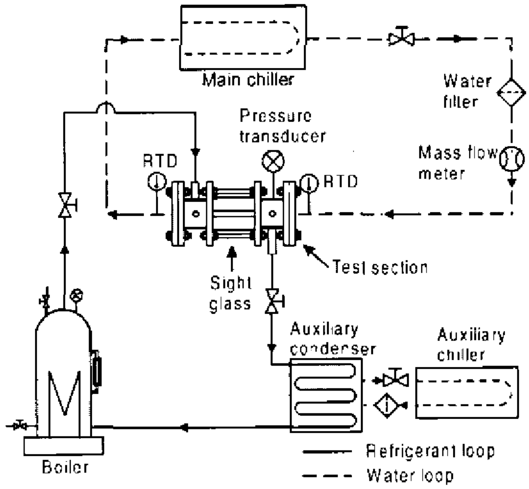

Fig. 1 Schematic diagram of the condensation heat transfer experimental apparatus

The main test section was made of a 80 mm inside diameter, 290 mm long stainless steel pipe with a 110 mm long sight glass installed in the middle to observe the condensation phenomenon. Both ends of the test section were flanged for easy mounting of the test tube. At both ends of the test section, nylon bushings of low thermal conductivity were tightly fastened on the tube so that heat transfer may occur only on the test tube. Finally, to reduce the heat loss to the surroundings, the whole apparatus including the test section was well insulated.

Vapor temperatures in the main test section and surface temperatures on the tube were measured by thermocouples calibrated against a resistance temperature detector (RTD) of 0.01℃ accuracy. The flow rate and temperature difference of the cooling water were measured by a mass flow meter of 0.2% accuracy and a set of RTDs of 0.01℃ accuracy respectively. Finally, a pressure transducer of 0.2% accuracy was calibrated and used to measure the pressure of the main test section. All measurements were taken for six testing fluids at the vapor temperature of 39℃ with wall subcooling of 3～8℃ under a heat flux of 32～116 and 42～142 kW/m^2 for the low fin and Turbo-C tubes respectively.

Table 1 Specifications of the low fin tube and Turbo-C tube

Tube type	Outside diameter(mm)	Fin height (mm)	Fin thickness		Fins/meter (Fins/inch)
			At tip (mm)	At base (mm)	
Low fin	18.9	1.214	0.252	0.576	1,024(26)
Turbo-C	18.9	0.760	0.250	0.350	1,654(42)

Table 1 lists the specifications of the low fin and Turbo-C tubes which are commercially used in refrigeration industry in Korea. The uncertainties occur during the measurements due to the uncertainties of the mass flow rate and temperature difference of the cooling water, wall temperature difference, and surface area etc. In this study, they were estimated by the method suggested by Kline and McClintock (1953) and turned out to be less than 7% for all data.

3. Results and Discussion

Figures 2 and 3 show the HTCs of various flammable refrigerants as a function of wall subcooling on a low fin and Turbo-C tubes respectively. In these figures, HTCs of HCFC22 and HFC134a from Jung et al.(1999a; 2003) are included for reference. Table 2 lists the HTCs of the various fluids for the enhanced tubes. The validity of the plain tube data and comparison with other data and correlations were discussed in our previous papers in detail (Jung et al., 1999a; 2003; 2004) and hence will not be discussed here.

Data of various flammable refrigerants of different vapor pressures obtained on enhanced tubes exhibited a typical trend that external condensation HTCs decrease as the wall subcooling increases. This trend was also shown with pure refrigerants on a plain tube (Jung et al., 1999a; 2003; 2004). For the low fin tube, HFC32 showed the highest HTCs which are approximately 30% higher than those of HCFC22. This is due to its excellent thermodynamic and transport properties of the liquid phase. HFC32 has the largest liquid density, density difference and second largest liquid thermal conductivity among the refrigerants tested, which have a significant effect on heat transfer as shown by the Nusselt correlation, equation (1).

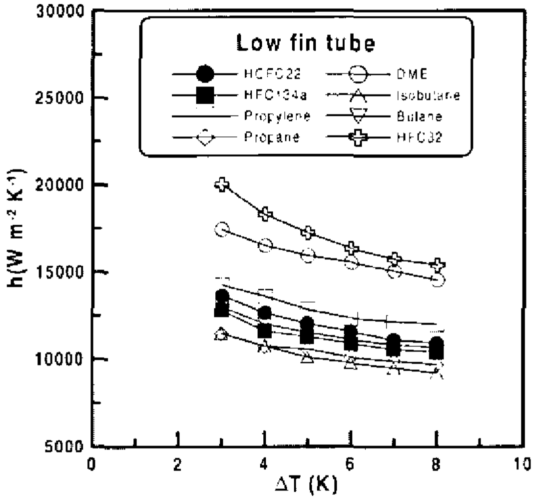

Fig. 2 Condensation HTCs of flammable refrigerants on a low fin tube

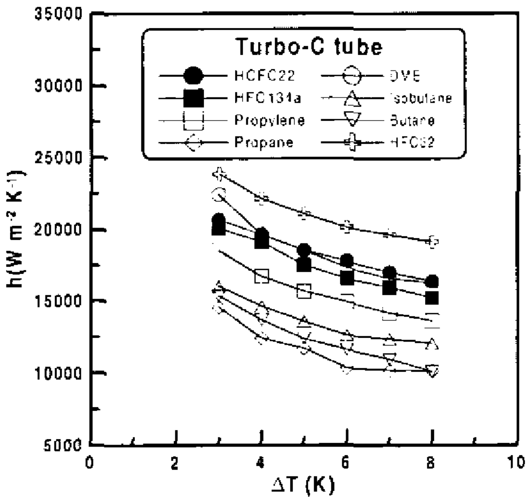

Fig. 3 Condensation HTCs of flammable refrigerants on Turbo-C tube

$$h_{Nusselt}=0.725\left[\frac{\rho_f(\rho_f-\rho_g)gk_f^3h_{fg}}{\mu_f\Delta TD}\right]^{1/4} \quad (1)$$

Table 2 Measured heat transfer coefficients of various refrigerants on two enhanced tubes

Tube	Refrigerant	Wall subcooling (K)					
		3.0	4.0	5.0	6.0	7.0	8.0
Low fin	Propylene	14248	13583	12843	12344	12149	12004
	Propane	11455	10723	10554	10106	9889	9686
	DME	17447	16508	15924	15547	15075	14546
	Isobutane	11464	10731	10104	9787	9496	9222
	Butane	12999	12023	11513	11123	10810	10667
	HFC32	20030	18290	17240	16334	15753	15413
Turbo-C	Propylene	18531	16713	15672	14923	14149	13607
	Propane	14552	12425	11720	10362	10198	10156
	DME	22425	19656	18525	17318	16591	16245
	Isobutane	16030	14635	13571	12617	12354	12054
	Butane	15353	13662	12392	11615	10958	10125
	HFC32	23846	22133	21098	20138	19623	19125

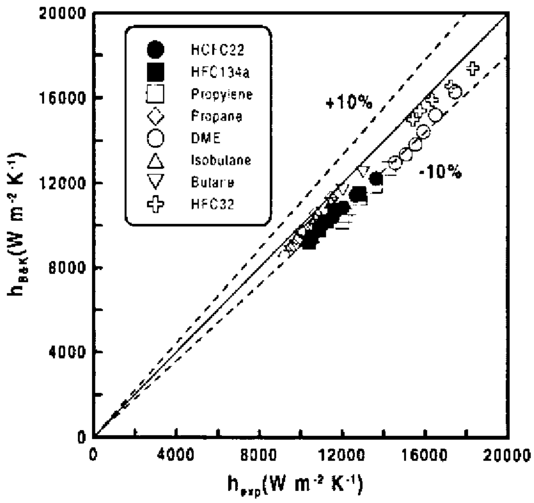

Fig. 4 Comparison between Beatty-Katz correlation and present data on a low fin tube

DME has the second highest HTCs which are approximately 24% higher than those of HCFC22 due also to its good thermodynamic and transport properties. Especially the liquid thermal conductivity of DME is the largest among the refrigerants tested. Propylene has 7% higher HTCs than those of HCFC22 while propane, isobutane, and butane have 15.0, 18.1, and 4.0% lower HTCs than those of HCFC22 respectively as shown in Fig. 2.

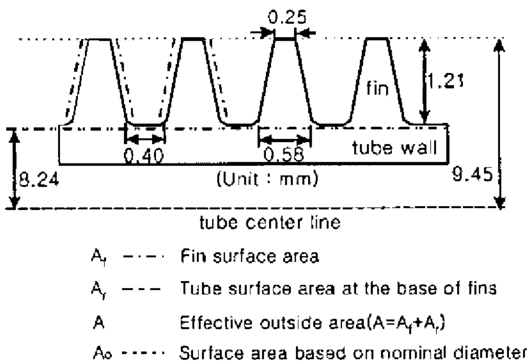

Fig. 5 Geometry of the low fin tube tested in this study

For Turbo-C tube, HFC32 again showed the highest HTCs which are 12.7% higher than those of HCFC22. DME has very similar HTCs as those of HCFC22. On the other hand, propylene, propane, isobutane, and butane have 17.8, 59.6, 35.6, and 49.6% lower HTCs than those of HCFC22 respectively as shown in Fig. 3.

Figure 4 shows the comparison between the experimental data obtained on a low fin tube and a correlation developed by Beatty and Katz (1948), equation (2). Once again, HCFC22 and HFC134a data obtained on the low fin tube in our previous works (Jung et al., 1999a ; 2003) are included for reference. Fig. 5 illustrates the detailed geometry of the low fin tube with specifications which are needed in equation (2).

$$h_{BK}=0.689F^{0.25}\left[\frac{A_r}{A}\frac{1}{D_r^{0.25}}+1.3\eta\frac{A_f}{A}\frac{1}{L_c^{0.25}}\right] \quad (2)$$

In equation (2), F is a constant as in equation (3) accounting for the effect of physical properties and L_c is a characteristic length defined by equation (4) and η is a fin efficiency which was assumed to be 1.0 for the low fin tube used in this study. As shown in Fig. 5, A, A_r, and A_f, are the effective outside area (0.05508 m^2), tube surface area at the root of the fins (0.01501 m^2), and fin surface area (0.04007 m^2) respectively. All properties needed in the calculation are obtained by REFPROP (1998).

$$F=\left(\frac{\rho_f^2 g k_f^3 h_{fg}}{\mu_f \Delta T}\right) \quad (3)$$

$$L_c=\frac{\pi(D_o^2-D_r^2)}{4D_o} \quad (4)$$

Experimental data of flammable refrigerants agreed well with the correlation within a mean deviation of 7.3%. Beatty and Katz' correlation showed a tendency of underestimating the data since it did not consider mainly the surface tension effect between the fins (Beatty and Katz, 1948). A similar phenomenon was observed with CFC11, CFC12, and HCFC22 alternative refrigerants in our previous studies (Jung et al., 1999a; 2003).

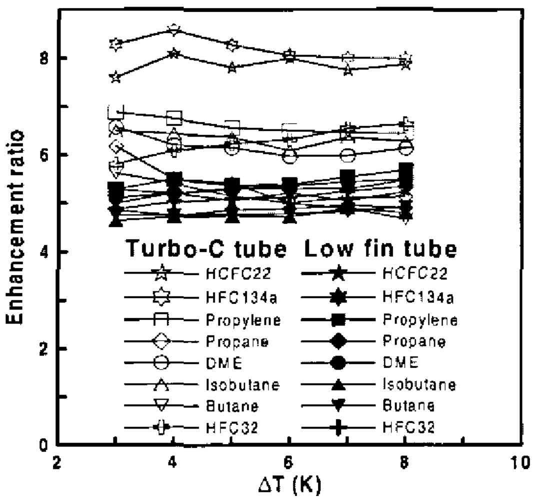

Fig. 6 Heat transfer enhancement ratios for low fin and Turbo-C tubes

Heat transfer enhancement ratios (HTERs) are obtained by dividing the HTCs of the enhanced tubes by those of the plain tube under the same condition. Fig. 6 shows the HTERs of the low fin and Turbo-C tubes as a function of wall subcooling. HTERs of the low fin and Turbo-C tubes are 4.6~5.7 and 4.7~6.9 respectively. The surface geometry of Turbo-C tube is 3 dimensional and hence the condensate is removed faster on this surface. That's why Turbo-C tube recorded the best heat transfer performance for all flammable refrigerants.

4. Conclusions

In this study, external condensation heat transfer coefficients (HTCs) of six flammable refrigerants of propylene (R1270), propane (R290), isobutane (R600a), butane (R600), dimethylether (RE170), and HFC32 were measured at the vapor temperature of 39℃ on a 1023 fpm low fin and Turbo-C tubes. All data were taken at the heat flux of 32~116 and 42~142 kW/m^2 for the low fin and Turbo-C tubes respectively. Based upon the test results, following conclusions can be drawn.

(1) Flammable refrigerants' data obtained on enhanced tubes showed a typical trend that external condensation HTCs decrease with increasing wall subcooling. No unusual behavior or phenomenon was observed for these fluids during experiments.

(2) HFC32 and DME showed 12~30 and 0.3~24% higher HTCs than those of HCFC22 due to their excellent thermophysical properties. On the other hand, propylene, propane, isobutane, and butane showed similar or lower HTCs than those of HCFC22.

(3) Beatty and Katz' correlation predicted the HTCs of the flammable refrigerants obtained on a low fin tube within a mean deviation of 7.3%.

(4) Heat transfer enhancement factors for the low fin and Turbo-C tubes were 4.6~5.7 and 4.7~6.9 respectively. Turbo-C tube of 3 dimen-

sional surface geometry had the best performance due to a fast removal of condensate.

Acknowledgments

This work was supported by INHA UNIVERSITY Research Grant.

References

Beatty, K. O. and Katz, D. L., 1948, "Condensation of Vapors on Outside of Finned Tubes," *Chemical Engineering Progress*, Vol. 44, No. 1, pp. 55~70.

Carnavos, T. C., 1980, "An Experimental Study: Condensing R-11 on Augmented Tubes," *ASME Paper*, No. 80-HT-54, pp. 54~60.

Int. Energy Agency's Heat Pump Center, Informative Fact Sheet: Hydrocarbons as Refrigerants in Residential Heat Pumps and Air-conditioners, 2002.

Joo, J. K., Cho, S., Jung, D. and Kim, C. B., 1997, "Condensation Heat Transfer Coefficients of CFC11 and its Alternative Refrigerants," *J. of Korean Society of Mechanical Engineers (B)*, Vol. 21, No. 6, pp. 830~840.

Jung, D., Kim, C. B., Cho, S. and Song, K., 1999a, "Condensation Heat Transfer Coefficients of Enhanced Tubes with Alterative Refrigerants for CFC11 and CFC12," *Int. J. Refrigeration*, Vol. 22, pp. 548~557.

Jung, D., Kim, C. B., Hwang, S. and Kim, K., 2003, "Condensation Heat Transfer Coefficients of R22, R407C, and R410A on a Horizontal Plain, Low Fin, and Turbo-C Tubes," *Int. J. Refrigeration*, Vol. 26, pp. 485~491.

Jung, D., Kim, C., Song, K. and Park, B., 2000, "Testing of Propane/Isobutane Mixture in Domestic Refrigerators," *Int. J. Refrigeration*, Vol. 23, pp. 517~527.

Jung, D., Park, B. and Lee, H., 1999b, "Evaluation of Supplementary/Retrofit Refrigerants for Automobile Air-conditioners Charged with CFC12," *Int. J. Refrigeration*, Vol. 22, pp. 558~568.

Jung, D., Song, K., Chae, S. and Bae, D., "Condensation Heat Transfer Coefficients of Flammable Refrigerants," *Int. J. Refrigeration*, Vol. 27, pp. 314~317.

Kim, N. H., Jung, I. K. and Kim, K. H., 1995, "An Experimental Study on the Condensation Heat Transfer of Low-finned Tubes," *Korean J. of Air-conditioning and Refrigeration Engineering*, Vol. 7, No. 2, pp. 298~309.

Kline, S. J. and McClintock, F. A., 1953, "Describing Uncertainties in Single-sample Experiments," *Mechanical Engineers*, Vol. 75, pp. 3~9.

Kruse, H., 1996, "The State of the Art of the Hydrocarbon Technology in Household Fefrigeration," *Proc. of the Int. Conferences on Ozone Protection Technologies, Washington (DC)*, pp. 179~188.

McLinden, M. O., Klein, S. A., Lemmon, E. W. and Peskin, A. P., 1998, NIST Thermodynamic and Transport Properties of Refrigerants and Refrigerant Mixtures — REFPROP Version 6.0.

Rudy, T. M. and Webb, R. L., 1985, "An Analytical Model to Predict Condensate Retention on Horizontal Integral-fin Tubes," *J. of Heat Transfer*, Vol. 107, pp. 361~368.

Webb, R. L. and Murawski, C. G., 1990, "Row Effect for R-12 Condensation on Enhanced Tubes," *J. of Heat Transfer*, Vol. 112, pp. 768~776.

Webb, R. L., 1994, "Principles of Enhanced Heat Transfer," John Wiley & Sons. Inc., New York, pp. 3~7.

Yau, K. K., Cooper, J. R. and Rose, J. W., 1985, "Effect of Fin Spacing on the Performance of Horizontal Integral Fin Condenser Tubes," *J. of Heat Transfer*, Vol. 107, pp. 377~383.