Available online at www.sciencedirect.com
ScienceDirect
International Journal of Refrigeration 30 (2007) 805–811
REVUE INTERNATIONALE DU FROID
INTERNATIONAL JOURNAL OF
refrigeration
www.elsevier.com/locate/ijrefrig

Experimental correlation of falling film condensation on enhanced tubes with HFC134a; low-fin and Turbo-C tubes

Yong Tae Kang[a,*], Hiki Hong[a], Young Soo Lee[b]

[a]School of Mechanical System Engineering, Kyung Hee University, Yong-In, Kyung-Gi 449-701, Republic of Korea
[b]Korea Energy Research Institute, Yusung, Daejeon 305-343, Republic of Korea

Received 10 October 2006; received in revised form 11 December 2006; accepted 12 December 2006
Available online 23 December 2006

Abstract

The objectives of this paper are to develop experimental correlations of heat transfer for enhanced tubes used in a falling film condenser, and to provide a guideline for optimum design of the falling film condenser with a high condensing temperature of 59.8 °C. Tests are performed for four different enhanced tubes; a low-fin and three Turbo-C tubes. The working fluid is HFC134a, and the system pressure is 16.0 bar. The results show that the heat transfer enhancement of low-fin tube, Turbo-C (1), Turbo-C (2) and Turbo-C (3) ranges 2.8–3.4 times, 3.5–3.8 times, 3.8–4.0 times and 3.6–3.9 times, respectively, compared with the theoretical Nusselt correlation. It was found that the condensation heat transfer coefficient decreased with increasing the falling film Reynolds number and the wall subcooling temperature. It was also found that the enhanced tubes became more effective in the high wall subcooling temperature region than in the low wall subcooling temperature region. This study developed an experimental correlation of the falling film condensation with an error band of ±5%.

Keywords: Refrigeration; Air conditioning; Condensation; Falling film; Tube; Enhanced; Experiment; Heat transfer; Correlation; R134a

HFC134a: corrélation expérimentale de la condensation d'un film tombant sur des tubes à surface augmentée

Mots clés : Réfrigération ; Conditionnement d'air ; Condensation ; Film tombant ; Tube ; Surface augmentée ; Expérimentation ; Transfert de chaleur ; Corrélation ; R134a

1. Introduction

Enhanced tubes with structured surface are strongly recommended to obtain high condensation performance in the shell side of the falling film condensers. There have been some researches on the performance improvement of condensation with the variation of materials and surface structure [1,2]. It is reported that the enhanced tubes such as low-fin and Turbo-C tubes provide a high heat transfer coefficient compared with the smooth tubes [3]. Hwang et al. [4] proposed that HFC134a be the best alternative to the conventional CFC12 on the enhanced tubes. Honda et al. [5] studied

* Corresponding author. Tel./fax: +82 31 201 2990.
E-mail address: ytkang@khu.ac.kr (Y.T. Kang).

0140-7007/$35.00
doi:10.1016/j.ijrefrig.2006.12.003

Nomenclature

A area (m^2)
C_p specific heat ($J\,kg^{-1}\,K^{-1}$)
d diameter (m)
f friction factor
g gravitational acceleration ($m\,s^{-2}$)
h heat transfer coefficient
k thermal conductivity ($W\,m^{-1}\,K^{-1}$)
L tube length (m)
$\dot{m}$ mass flow rate ($kg\,s^{-1}$)
Nu Nusselt number
p fin pitch (mm)
Pr Prandtl number, $\mu C_p/k$
Q heat transfer rate (W)
R_w wall resistance ($K\,W^{-1}$)
Re Reynolds number, Vd/ν
Re_f film Reynolds number
U overall heat transfer coefficient ($W\,m^{-2}\,K^{-1}$)
V velocity ($m\,s^{-1}$)
ΔP pressure drop (Pa)
ΔT_{LM} log mean temperature difference (K)
ΔT_{sub} wall subcooling temperature (K)

Greek symbols
ε non-dimensional geometric parameter, λ/p
λ fin height (mm)
μ viscosity (Pa s)
ν kinematic viscosity ($m^2\,s^{-1}$)
ρ density ($kg\,m^{-3}$)
Γ mass flow rate per unit length ($kg\,m^{-1}\,s^{-1}$)
ω non-dimensional parameter, gd_o^3/ν^2

Subscripts
c coolant
cal calculation
exp experiment
i inner
in inlet
l liquid
o outer
out outlet
ref refrigerant
sat saturation
w wall

the effect of fin geometry of horizontal tube finned tubes on the condensation performance with HFC134a. Yan and Lin [6] tested condensation heat transfer and pressure drop in a small pipe with HFC134a. Recently, enhanced tubes with three-dimensional fin geometry such as Turbo-C tubes have been used for performance improvement of condensers. Webb and Murawski [7] tested various enhanced tubes including low-fin and Turbo-C tubes, and found that the Turbo-C tube provided the highest heat transfer coefficient among the enhanced tubes tested. Jung et al. [8] measured condensation heat transfer coefficients of a plain tube, low-fin and Turbo-C tube for the low pressure refrigerants (CFC11 and HCFC123) and for the medium pressure refrigerants (CFC12 and HFC134a) at the condensation temperature of 39 °C, and also found that the Turbo-C tube provided the highest heat transfer coefficient among the tubes tested. However, most of the previous studies have focused on the performance comparisons of working fluids such as water and Ozone depleting CFC refrigerants. Few papers [7,8] tested a plain tube, a low-fin tube and a Turbo-C tube with R22, R407C, and R410A, and only few data are available for a Turbo-C tube with HFC134a which is generally used for the falling film condenser of turbo chillers. Therefore, it is required that the effects of geometric conditions of the enhanced tubes such as Turbo-C tubes on the condensation performance be evaluated for practical application to a real condenser design.

In the present study, three different Turbo-C tubes and a low-fin are tested at a high saturation temperature of 59.8 °C for application to the falling film condenser of turbo chillers. The film Reynolds number ranges from 120 to 330. The objectives of this study are to develop experimental correlations of the falling film condensation heat transfer for enhanced tubes used in the falling film condenser, and to provide a guideline for optimum design of the condenser for practical applications to a real turbo machine.

2. Experimental apparatus

Fig. 1 shows the experimental apparatus for the falling film condensation tests. The experimental setup consists of evaporator section, condenser section and the internal flow section. The working fluid is HFC134a, and the system pressure and the temperature were 16.0 bar and 59.8 °C, respectively. The internal flow section is installed at the coolant flow line to measure the internal flow characteristics of the enhanced tubes. The evaporator boils the refrigerant with the chilled water whose inlet temperature is controlled by a heater with the heat capacity of 30 kW. Fig. 2 shows the condenser test section with enhanced surfaces, called a low-fin and three Turbo-C tubes. In the falling film condenser, the cooling water (Ethylene Glycol 46.4%) flows inside the tubes and HFC134a refrigerant condenses on the outside of the enhanced tubes. The enhanced tubes of 1918 mm in length are placed in a row with five columns inside the condenser shell as shown in Fig. 2. The shell is fabricated of rolled steel (SS400), 267.4 mm in outer diameter, 6.6 mm thick and 1996 mm long. Five sight glasses are equipped for visual observation of the condensation. Fig. 3 shows the photographs of the low-fin and Turbo-C tubes

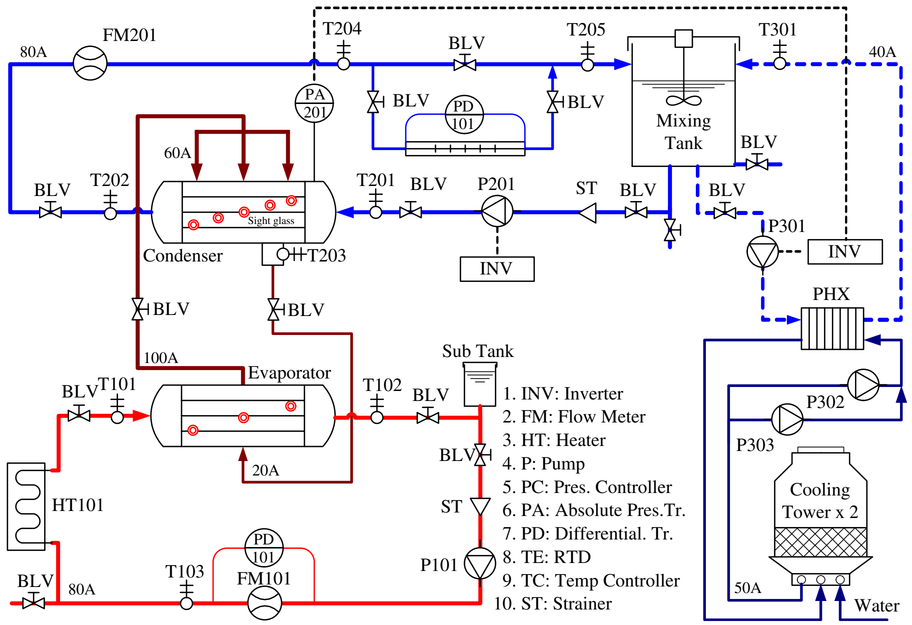

Fig. 1. Schematic diagram of experimental apparatus.

which have smooth internal surfaces as shown in the figure. The geometric details of the enhanced test tubes are summarized in Table 1. Initially, the refrigerant boils in the evaporator and moves up to the condenser as a gas state. The pipe from evaporator to condenser is made of carbon steel, 80 mm in diameter. The gasified refrigerant is well distributed to the top of the condenser by the three gas distributors. The gasified refrigerant is condensed on the enhanced tubes by releasing the condensing heat to the cooling water. The condensed liquid refrigerant falls down to the evaporator by gravity, through a pipe of 10 mm in diameter. The cooling water flows into the condenser through an electric valve which operates automatically to maintain the pressure of the refrigerant. To supply the chilled and cooling water with a required temperature, a mixing tank is installed, which has two rooms separated by a wall. It mixes the high-temperature cooling water from the condenser with the cold water (-20 °C) by controlling the flow rate, in order to maintain the cooling water inlet temperature constant. A reciprocating chiller (10RT) takes heat from the mixing tank.

The chilled and cooling water include 29.7 and 46.4% of ethylene glycol, respectively, to remove the crystallization problem at a low temperature. The mass flow rate of the chilled water was controlled by an inverter pump and was measured by a nozzle flow meter with differential pressure transmitter. The flow rate of the cooling water was measured by an electromagnetic flow meter. The repeatability for the present experiments was confirmed by conducting each case two times. The uncertainty in the measurement of

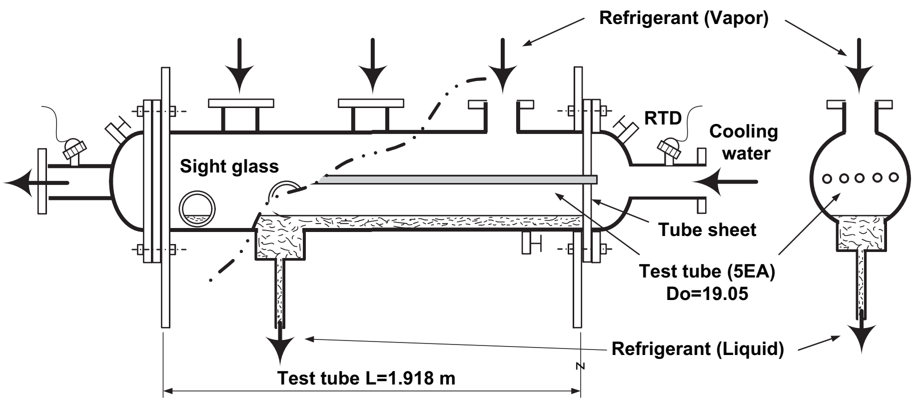

Fig. 2. Condenser test section.

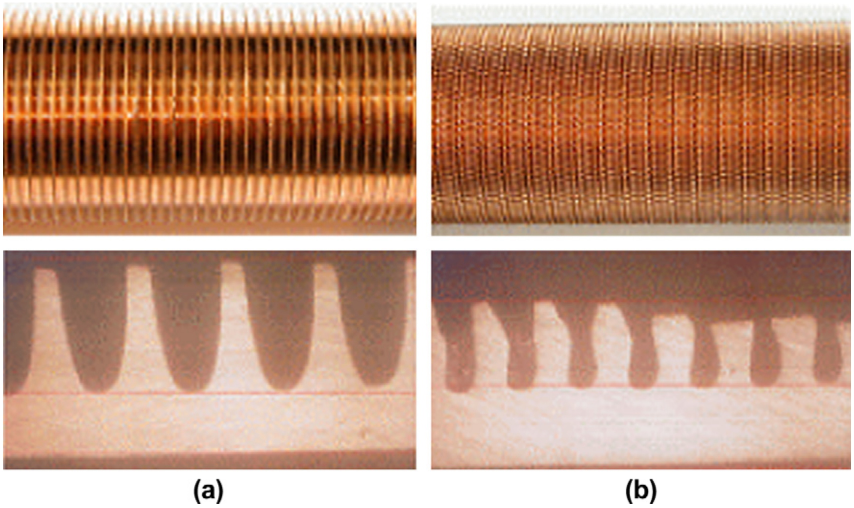

Fig. 3. Photographs of the cross section for each enhanced tube. (a) Low-fin tube. (b) Turbo-C tube.

mass flow rates was estimated as 0.13%. Temperatures were measured by RTD's, which were calibrated by a precision thermometer with a measurement error of ±0.03 °C in the temperature range of this experiment.

2.1. Data reduction

In the present experiments, the internal friction factor for the enhanced tubes was measured by using a single long tube with the length of 3.0 m, which had the internal flow section. The internal friction factor f_i is defined by the Darcy–Weisbach's definition [9] as follows;

$$\Delta P = f_i \frac{L}{d_i} \frac{1}{2} \rho V_i^2 \tag{1}$$

The friction factor for the present enhanced tubes with internal smooth surface was developed as follows;

$$f_i = \frac{96}{Re_i^{0.76}} \tag{2}$$

After obtaining the friction factor of Eq. (2), the following Gnielinski's correlation [10] is used to calculate the internal heat transfer coefficient;

Table 1
Geometric conditions of the enhanced tubes

Tube No.	Tube diameter d_o (mm)	Fin height λ (mm)	Wall thickness (mm)	Fin pitch p (mm)
Low-fin	19.05	1.42	0.69	0.98
Turbo-C (1)	19.05	0.79	1.00	0.61
Turbo-C (2)	19.05	1.03	0.68	0.61
Turbo-C (3)	19.05	1.13	0.79	0.61

$$Nu_i = \frac{(f_i/8)[Re_i - 1000]Pr_i}{1 + 12.7(f_i/8)^{1/2}[Pr_i^{2/3} - 1]} \tag{3}$$

Heat transfer to the cooling water, log mean temperature difference (LMTD), ΔT_{LM}, and, the overall heat transfer coefficients are calculated, respectively, as follows;

$$Q = \dot{m}_c C_{p,c}(T_{c,out} - T_{c,in}) = UA\Delta T_{LM} \tag{4}$$

$$\Delta T_{LM} = \frac{(T_{ref} - T_{c,in}) - (T_{ref} - T_{c,out})}{\ln\left(\dfrac{T_{ref} - T_{c,in}}{T_{ref} - T_{c,out}}\right)} \tag{5}$$

and

$$\frac{1}{UA} = \frac{1}{h_i A_i} + R_w + \frac{1}{h_o A_o} \tag{6}$$

In Eq. (6), R_w is the conduction resistance through the wall. In Eq. (6), the outside convection resistance was calculated based on d_o. d_o is the outside diameter measuring at the top of the fin height. The outside heat transfer area A_o is the nominal area based on the outside diameter. It is to be noted that for the low-fin and Turbo-C tubes, the nominal area is quite different from the actual area. However, it is generally accepted that the nominal area based on the outside diameter is used to calculate the heat transfer coefficient for the enhanced tubes because the actual heat transfer area cannot be measured accurately [11]. In the present study, the wall conduction resistance was found to be only less than 2% of the outside convection resistance in the calculation of the overall heat transfer coefficient. By combining Eqs. (3)–(6), the heat transfer coefficient for the falling film condensation on the enhanced tubes, h_o could be obtained. The

Nusselt number during the falling film condensation on the outer surface is defined as follows;

$$Nu_o = \frac{h_o}{k_l}\left(\frac{\nu_l^2}{g}\right)^{1/3} \tag{7}$$

Now, the wall subcooling temperature and the film Reynolds number are calculated as follows, respectively;

$$\Delta T_{sub} = T_{sat} - T_{w,o} = \frac{Q}{h_o A} \tag{8}$$

$$Re_l = \frac{4\Gamma_l}{\mu_l} \tag{9}$$

where

$$\Gamma_l = \frac{\dot{m}_{ref}/2}{L} \tag{10}$$

The experimental uncertainties in the measurement of the friction factor, f and the condensation heat transfer coefficient, h_o were estimated as 1.4 and 6.5%, respectively.

3. Results and discussion

Fig. 4 shows the comparison of the present experimental results and the Beatty and Katz's theoretical correlation [12] which is most widely used for the low-fin tube. The saturation temperature was kept constant at 59.8 °C during the experiment. The experimental data were satisfied with the theoretical correlation within ±20%. The experimental data were slightly higher than the correlation values because the condensate spreading effect between the fins and surface tension effect were not taken into account in the theoretical correlation. The surface tension plays an important role in spreading the condensate, and its effect on the heat transfer enhancement depends on the shape of the enhanced surfaces.

Fig. 5 shows the comparisons of the Nusselt condensation equation, the Beatty and Katz's theoretical correlation

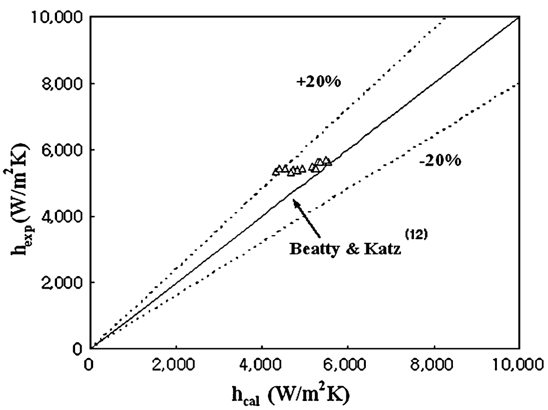

Fig. 4. Experimental comparison with the Beatty and Katz correlation for the low-fin tube.

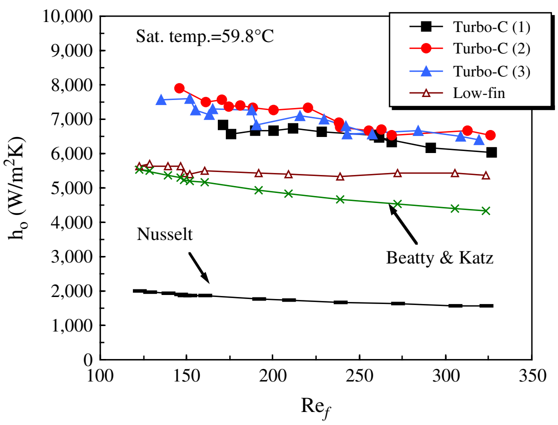

Fig. 5. Condensation heat transfer coefficient versus falling film Reynolds number.

and the present experimental results for the enhanced tubes. Of course, the Nusselt condensation equation is for a smooth surface and the Beatty and Katz's correlation is for the low-fin tubes. The condensation heat transfer coefficient is plotted as a function of the condensate film Reynolds number. There was not much variation in the Nusselt condensation equation while somewhat effect of the film Reynolds number on the condensation heat transfer coefficient was found in the Beatty and Katz's correlation and the present experimental results for the enhanced tubes. The condensation heat transfer coefficient decreased with increasing the film Reynolds number. This is because the condensate film becomes thicker leading to a high conduction resistance as the film Reynolds number increases in the laminar flow regime. Therefore, it could be concluded that the negative effect by the condensate film resistance was more dominant than the positive effect by the turbulent mixing of the liquid in the laminar region even for the enhanced tubes considered in the present study. The Reynolds number ranged from 120 to 330, which is much less than the transition Reynolds number of 1600–1800 for the falling film condensation on the smooth tube. It was also visualized through the sight glass that the uniform liquid film formed between the fins and the uniform liquid column did between the tubes.

Fig. 6 shows the condensation heat transfer coefficient as a function of the wall subcooling temperature, ΔT_{sub} for the enhanced tubes. Similar to Fig. 5, the condensation heat transfer coefficient decreased with increasing the wall subcooling temperature. This is also because of the effect of the condensate film resistance. As the wall subcooling temperature increases, the amount of condensate increases leading to a high conduction resistance. It was found that the Turbo-C tubes gave much higher heat transfer coefficient than the Low-fin tube. This is mainly because the Turbo-C tube has a three-dimensional shape as shown in Fig. 3, leading to a thin condensate film for a given amount of liquid while the Low-fin tube does a two-dimensional shape in

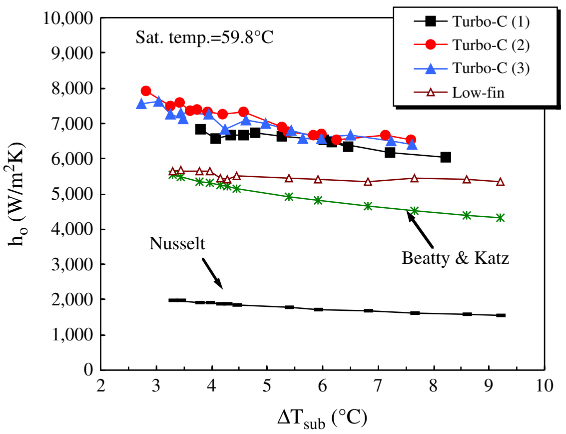

Fig. 6. Condensation heat transfer coefficients for each enhanced tube.

radial and axial directions. The other reason is that the condensate film formed on the Low-fin surface while condensate droplets formed on the Turbo-C surfaces. The similar uniform liquid columns were found between the tubes for both Turbo-C and the Low-fin tubes. The Turbo-C (1), Turbo-C (2) and Turbo-C (3) tubes gave about 18, 30 and 26% higher heat transfer coefficient than the Low-fin tube, respectively. The Turbo-C (2) with the fin height of 1.025 mm gave the highest heat transfer coefficient. This implies that the liquid holding phenomenon exists between the fins if the fin height increases too much. Therefore, it is suggested that an optimum fin height exist for a maximum heat transfer coefficient and guesses that the Turbo-C (2) tube falls on the condition.

Fig. 7 shows the condensation heat transfer enhancement ratio of the enhanced tubes to the Nusselt equation. h_{Nu} represents for the heat transfer coefficient based on the Nusselt condensation correlation. It was found that the enhancement ratio ranged 2.8–3.4 for the Low-fin tube, 3.5–3.8 for the Turbo-C (1), 3.8–4.0 for the Turbo-C (2) and 3.6–3.9 for the Turbo-C (3), respectively. It was also found that the enhancement ratio increased with increasing the wall subcooling temperature. This implies that the enhanced tubes become more effective in the high wall subcooling temperature region than in the low wall subcooling temperature region.

Based on the experimental results of falling film condensation, the following experimental correlation was developed with an error band of ±5% as shown in Fig. 8;

$$Nu_c = 0.148\, Re_f^{-0.201} Pr^{1.461} \omega^{0.063} \varepsilon^{0.179} \tag{11}$$

where

$$Re_f = \frac{4\Gamma}{\mu}, \quad \omega = \frac{g d_o^3}{\nu^2}, \quad \text{and} \quad \varepsilon = \frac{\lambda}{p}$$

The applicable ranges for the correlation are $120 \le Re_f \le 330$ and $1.31 \le \varepsilon \le 1.83$, respectively. In Eq. (11), Pr and ε were included to consider the effect of the refrigerant property variations depending on the saturation temperature change and the effect of the gravity force in the falling film condensation. The experimental correlation can be applied for condenser design of HFC134a with representative enhanced tubes such as Turbo-C and Low-fin tubes with a diameter of 19.05 mm.

4. Conclusions

The following conclusions are drawn from the present study.

1. Four different enhanced tubes including Turbo-C tubes were tested to apply to the falling film condenser in Turbo chillers. It was found that the condensation heat transfer coefficient decreased with increasing the

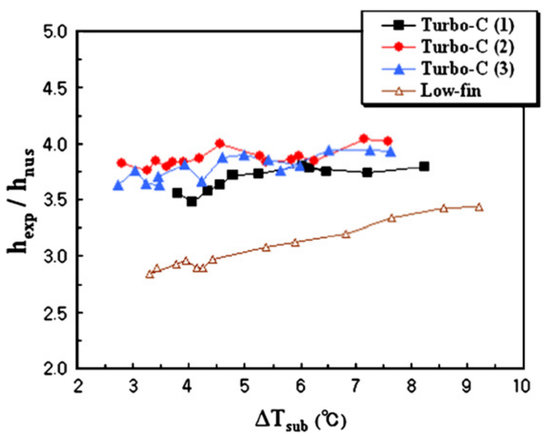

Fig. 7. Heat transfer enhancement factor versus the wall subcooling temperature.

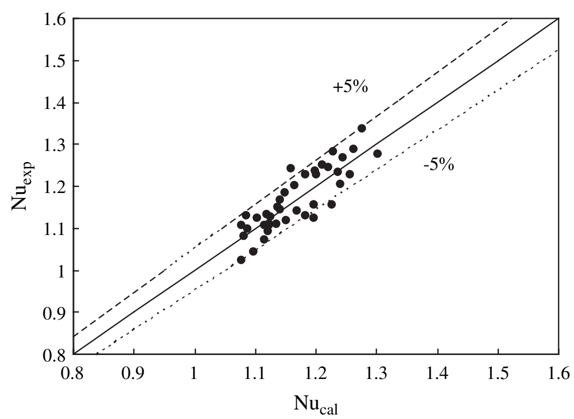

Fig. 8. Experimental correlation of the falling film condensation for each enhanced tube.

falling film Reynolds number and the wall subcooling temperature. It was concluded that the negative effect by the condensate film resistance was more dominant than the positive effect by the turbulent mixing of the liquid in the laminar region even for the enhanced tubes considered in the present study.
2. The Turbo-C (1), Turbo-C (2) and Turbo-C (3) tubes gave about 18, 30 and 26% higher heat transfer coefficients than the Low-fin tube, respectively. It was found that an optimum fin height existed for a maximum heat transfer coefficient for a given geometric conditions.
3. It was found that the heat transfer enhancement ratio ranged 2.8–3.4 for the Low-fin tube, 3.5–3.8 for the Turbo-C (1), 3.8–4.0 for the Turbo-C (2) and 3.6–3.9 for the Turbo-C (3), respectively. It was also found that the enhanced tubes became more effective in the high wall subcooling temperature region than in the low wall subcooling temperature region.
4. This study developed an experimental correlation for the falling film condensation with an error band of $\pm 5\%$. The correlation can be applied for falling film condenser design of HFC134a with enhanced tubes such as Turbo-C and Low-fin tubes.

Acknowledgement

This work was partially supported by Korea Energy Management Corporation Grant (2005-E-BD11-P-03-3-010-2005) and by MOCIE through EIRC program.

References

[1] H.H. Kim, Y.J. Baik, H.S. Ra, J.T. Park, S.R. Park, Experimental investigation of heat pump system using river water as a heat source, The Society of Air-Conditioning and Refrigerating Engineers of Korea (2002) 64–69.
[2] P.J. Marto, R.H. Nunn, Power Condenser Heat Transfer Technology, Hemisphere, Washington, 1981, pp. 287–372.
[3] D.S. Jung, S. Chae, D. Bae, G. Yoo, Condensation heat transfer coefficients of binary HFC mixtures on low fin and Turbo-C tubes, International Journal of Refrigeration 28 (2005) 212–217.
[4] S.M. Hwang, K.K. Kim, D. Jung, C.B. Kim, Condensation heat transfer coefficients of R22 alternative refrigerants on enhanced tubes, Journal of Society of Air-Conditioning and Refrigerating Engineers of Korea 23 (4) (1999) 459–469.
[5] H. Honda, N. Takata, H. Takamatsu, J.S. Kim, K. Usami, Condensation of downward-flowing HFC134a in a staggered bundle of horizontal finned tubes: effect of fin geometry, International Journal of Refrigeration 22 (2002) 3–10.
[6] Y.Y. Yan, T.F. Lin, Condensation heat transfer and pressure drop of refrigerant R-134a in a small pipe, International Journal of Heat and Mass Transfer 42 (1999) 697–708.
[7] R.L. Webb, C.G. Murawski, Row effect for R012 condensation on enhanced tubes, Journal of Heat Transfer 112 (1990) 768–776.
[8] D.S. Jung, C. Kim, S. Hwang, K. Kim, Condensation heat transfer coefficients of R22, R407C, and R410A on a horizontal plain, low fin, and Turbo-C tubes, International Journal of Refrigeration 26 (4) (2003) 485–491.
[9] Y.A. Çengel, Heat Transfer: A Practical Approach, McGraw-Hill, New York, 1998, pp. 605–616.
[10] V. Gnielinski, New equations for heat and mass transfer in turbulent pipe and channel flow, International Chemical Engineering 16 (2) (1976) 359–368.
[11] D. Jung, C.B. Kim, S. Cho, K. Song, Condensation heat transfer coefficients of enhanced tubes with alternative refrigerants for CFC11 and CFC12, International Journal of Refrigeration 22 (7) (1999) 548–557.
[12] K.O. Beatty, D.L. Katz, Condensation of vapors on outside of finned tubes, Chemical Engineering Progress 44 (1) (1948) 55–70.