International Journal of Air-Conditioning and Refrigeration Vol. 16, No. 3 (2008) 100~107

Airside Performance of Convex Louver Fin-and-Tube Heat Exchangers

Nae-Hyun Kim†, Wang-Kyu Oh*

Department of Mechanical Engineering, University of Incheon 177 Dohwa-Dong, Nam-Gu, Incheon, 402-749, Korea

**Graduate School, University of Incheon 177 Dohwa-Dong, Nam-Gu, Incheon, 402-749, Korea*

(Received June 30, 2008; Revision received August 29, 2008; Accepted September 19, 2008)

Abstract

The heat transfer and friction characteristics of heat exchangers having convex louver fins are experimentally investigated, and the results are compared with those of wave fin counterparts. Eighteen samples (nine convex louver fin samples and nine wave fin samples) which had different fin pitches (1.81 mm to 2.54 mm) and tube rows (one to four) were tested. The convex angle was 11.7°. The j factors are insensitive to fin pitch, while f factors increase as fin pitch increases. The effect of fin pitch on f factor is more significant for the wave fin compared with the convex louver fin. It appears that the complex fin pattern of the convex louver fin induces intense mixing of the flow, and thus reduces the effect of fin pitch. Both the j and f factors decrease as the number of tube row increases. However, as the Reynolds number increases, the effect of tube row diminishes. Comparison of the convex louver fin j factors with those of wave fin reveals that convex louver fin j factors are 18% to 29% higher than those of wave fin. The f factors are 16% to 34% higher for the convex louver fin. The difference increases as fin pitch decreases. Existing correlation fails to adequately predict the present data. More data is needed for a general correlation of the convex louver fin geometry.

Key words: Heat exchanger, Heat transfer Coefficient, Pressure drop, Fin-and-tube, Convex louver fin

Nomenclature

A : heat transfer area (m^2)
A_c : minimum free flow area (m^2)
A_t : heat transfer area at the mid-plane of the tube wall (m^2)
c_p : specific heat (J $kg^{-1}s^{-1}$)
D_c : tube diameter including fin collar thickness (m)
D_i : tube-side diameter (m)
D_r : maximum tubeside diameter (to the fin root) (m)
f : friction factor, see Eq. (19) (dimensionless)
f_i : tube-side friction factor, see Eq. (10) (dimensionless)
G : mass flux (kg m^{-2})
h : heat transfer coefficient (W m^{-2} K^{-1})
j : Colburn j factor, $\frac{h_o}{\rho_a V_{max} c_{pa}} \mathrm{Pr}_a^{2/3}$ (dimensionless)
k_t : thermal conductivity of the tube (W m^{-1} K^{-1})
k_w : thermal conductivity of water (W m^{-1} K^{-1})
N : number of tube row or number of data (dimensionless)
NTU : number of transfer units, see Eq. (2) (dimensionless)
P_f : fin pitch (m)
P_t : transverse tube pitch (m)
P_l : longitudinal tube pitch (m)
Pr : Prandtl number (dimensionless)
r_c : tube radius including fin collar (m)
R_{eq} : equivalent radius (m)
Re_{Dc} : Reynolds number based on D_c, $\frac{\rho_a V_{max} D_c}{\mu_a}$ (dimensionless)
Re_{Di} : tube-side Reynolds number (dimensionless)
s : fin spacing (m)
t : tube wall thickness (m)
T : temperature (K)
t_f : fin thickness (m)
U : overall heat transfer coefficient (W m^{-2} s^{-1})
V_{max} : maximum airside velocity (m s^{-1})

† Corresponding author. Tel.: +82 32 770 8420, Fax.: +82 32 770 8410
E-mail address: knh0001@incheon.ac.kr

Greek letters

ΔP : pressure loss (Pa)
η : fin efficiency, see Eq. (12) (dimensionless)
η_o : surface efficiency, see Eq. (11) (dimensionless)
ρ : density (kg m^{-3})
μ : dynamic viscosity (kg m^{-1} s^{-1})
σ : contraction ratio of the cross-sectional area (dimensionless)

Subscripts

2 : subscript of NTU, see Eq. (2)
a : air
c : fin collar
exp : experimental
i : tube side
in : inlet
f : fin
m : mean
o : airside
out : outlet
p : prediction
t : tube wall
w : water

1. Introduction

Fin-and-tube tube heat exchangers have been used for heat exchange between gases and liquids for many years. In a forced convective heat transfer between gas and liquid, the controlling thermal resistance is on the gas-side, and specially configured fins have been used to improve the gas-side performance. Webb and Kim[1] and Wang[2] provide recent progress on this issue. Of the many enhanced geometries, convex louver fin is known to provide the most significant heat transfer enhancement[1].

A sketch of the convex louver fin is shown in Fig. 1. The convex louver fin is made by cutting and bending the sides of the wave fins. The resultant shape is a combination of wave and louver fin geometries. Although there are extensive experimental study related to wave fin geometry[3-8] and louver fin geometry[9-11], very limited data are available for the convex louver fin geometry. Hatada et al.[12] tested one row sample having 16 mm diameter tubes. The convex angle was 12.5°. Compared with the plain fin counterpart, the heat transfer coefficient of the convex louver fin sample was 2.5 times higher, and the pressure drop was 3.2 times higher. Wang et al.[13,14] investigated the effect of fin pitch and tube rows on j and f factors of the convex louver geometry. Nine samples having 10.06 mm diameter tubes[13] and seven samples having 8.54 mm diameter tubes[14] were tested. For all the samples, the convex angle was 15.5°. They reported that the trends of the j and f factor were similar to those of the wave fin geometry. The j factors were independent of fin pitch. The row effect on the j factors was relatively weak compared with that of the plain fin geometry. The friction factors were independent of the number of tube rows. The j factors of the convex louver fin samples showed 21 to 41% increase and f factors showed 60 to 72% increase compared to the corresponding wavy fin samples. A correlation was proposed based on their own data.

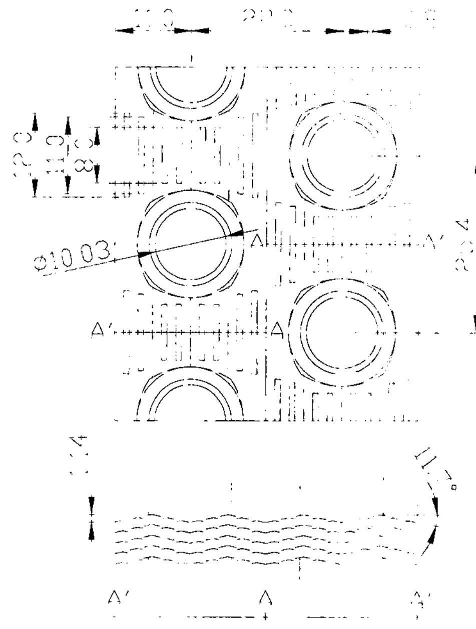

Fig. 1. Drawing of the convex louver fin geometry tested in this study (unit: mm).

The literature survey reveals that only 15.5° convex angle samples were tested for the convex louver geometry. Definitely, more data are needed, especially for samples having convex angles other than 15.5°. In the present study, nine samples having 11.7° convex angle were tested, and the results are compared with wavy fin counterparts. Data are also compared with the existing correlation.

2. Experiments

2.1 Heat exchanger samples

A total of eighteen heat exchangers (nine convex louver fin and nine wave fin) were tested in the present study. All the samples had same wave angle of 11.7°. The height and the width of the samples were 250 mm and 400 mm respectively. Detailed dimensions of the fin patterns are illustrated in Fig. 1. It is shown that 13% of the fin surface is louvered. The waffle height of both the convex louver and the wave fin was 1.14 mm. For all the heat exchanger samples, P_t was 25.4 mm, P_l was 22.0 mm, and D_c was 10.03 mm. The D_c was determined by measuring the tube outer diameter (including fin collar) from the samples. The fin pitch was varied from 1.81 mm to 2.54 mm, and the number of tube row was varied from one to four. For all the heat exchangers, smooth tubes were used at the tube-side, and the tubes were circuited to cross-counter configuration with single inlet and outlet.

2.2 Test apparatus and procedures

A schematic drawing of the apparatus is shown in Fig. 2. It consists of a suction-type wind tunnel, water circulation and control units, and a data acquisition system. The apparatus is situated in a constant temperature and humidity chamber. The airside inlet condition of the heat exchanger is maintained by controlling the chamber temperature and humidity. The inlet and outlet dry and wet bulb temperatures are measured by the sampling method as suggested in ASHRAE Standard 41.1[15]. A diffusion baffle is installed behind the test sample to mix the outlet air. The waterside inlet condition is maintained by regulating the flow rate and inlet temperature of the constant temperature bath situated outside of the chamber. Both the air and the water temperatures are measured

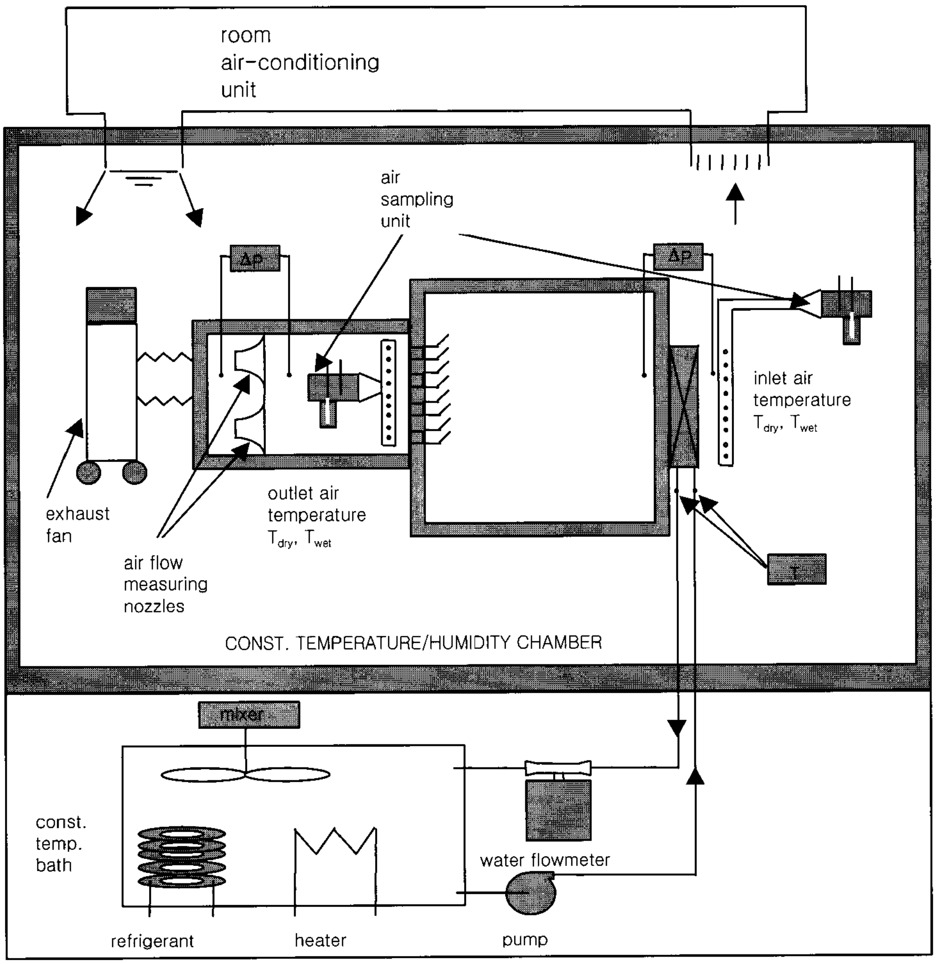

Fig. 2. Schematic drawing of the test setup.

Table 1. Uncertainty analysis.

Parameter	Max. Uncertainties
Temperature	± 0.1 K
Differential pressure	± 1.0 Pa
Water flow rate	± 1.5×10^{-6} m^3/s
Re_{Dc}	± 2%
f	± 10 %
j	± 12%

by pre-calibrated RTDs (Pt-100 Ω sensors). Their accuracies are ± 0.1°C. The water flow rate is measured by a positive displacement type flow meter, whose accuracy is ± 0.0015 liter/s. The airside pressure drop across the heat exchanger is measured using a differential pressure transducer. The air flow rate is measured using a nozzle pressure difference according to ASHRAE Standard 41.2[16]. The accuracy of the differential pressure transducers is ± 1.0 Pa.

During the experiment, the water temperature was held at 45°C. The chamber temperature was maintained at 21°C with 60% relative humidity. Experiments were conducted varying the frontal air velocity from 0.3 m/s to 3.5 m/s. The energy balance between the airside and the tube-side was within ± 3%. The discrepancy increased as the air velocity decreased. All the data signals were collected and converted by a data acquisition system (a hybrid recorder). The data were then transmitted to a personal computer for further manipulation. An uncertainty analysis was conducted following ASHRAE Standard 41.5[17], and the results are listed in Table 1. The major uncertainty on the friction factor was the uncertainty of the differential pressure measurement (± 10%), and the major uncertainty on the heat transfer coefficient (or j factor) was that of the tube-side heat transfer coefficient (± 10%). The uncertainties decreased as the Reynolds number increased.

2.3 Data reduction

For the cross-counter configuration of the present study, appropriate equations for the heat exchanger analysis are given by Taborek[18]. The UA value is obtained from the following equations.

$$UA = (\dot{m}c_p)_{air}NTU_2 \tag{1}$$

$$NTU_2 = -2\ln(1-K) \tag{2}$$

The K is obtained from the following equations.

$$2\text{ row}: \frac{K}{2}+(1-\frac{K}{2})\exp(2KR)=\frac{1}{1-RP} \tag{3}$$

$$3\text{ row}: \begin{aligned}&K[1-\frac{K}{4}-RK(1-\frac{K}{2})]\exp(KR)\\&+(1-\frac{K}{2})^2\exp(3KR)=\frac{1}{1-RP}\end{aligned} \tag{4}$$

$$4\text{ row}: \begin{aligned}&\frac{K}{2}(1-K+\frac{K^2}{4})+K(1-\frac{K}{2})\\&+[1-\frac{RK}{8}(1-\frac{K}{2})\exp(2KR)]\\&+(1-\frac{K}{2})^3\exp(4KR)=\frac{1}{1-RP}\end{aligned} \tag{5}$$

$$R=\frac{T_{w,in}-T_{w,out}}{T_{air,out}-T_{air,in}} \tag{6}$$

$$P=\frac{T_{air,out}-T_{air,in}}{T_{w,in}-T_{air,in}} \tag{7}$$

For the one row configuration, a cross-flow $\varepsilon-NTU$ equation was used. For the experimental range, $0.18\le P\le 0.87$ and $0.23\le R\le 1.72$. The airside heat transfer coefficient is obtained from the following equation.

$$\frac{1}{\eta_o h_o A_o}=\frac{1}{UA}-\frac{1}{h_i A_i}-\frac{t}{k_t A_t} \tag{8}$$

The total surface area A_0 of both the convex louver and the wave fin is an actual heat transfer area (not a projected area). The resulting A_0 of the samples were 2.1 % larger than those having plain fins. For the tube-side heat transfer coefficients, Gnielinski's Eq.[19] was used.

$$h_i=\frac{k_w}{D_i}\frac{(f_i/8)(Re_{Di}-1000)Pr_w}{1+12.7(f_i/8)^{1/2}(Pr_w^{2/3}-1)} \tag{9}$$

$$f_i=(0.79\ln Re_{Di}-1.64)^{-2} \tag{10}$$

For an accurate assessment of the airside heat transfer coefficient, it is important to minimize the tube-side thermal resistance. Throughout the experiment, the tube-side thermal resistance was less than 10% of the total thermal resistance.

The surface efficiency η_o is obtained from Eq. (11).

$$\eta_o=1-\frac{A_f}{A_o}(1-\eta) \tag{11}$$

The fin efficiency is given by Schmidt[20] as

$$\eta = \frac{\tanh(mr_c\phi)}{mr_c\phi} \tag{12}$$

where

$$m = \sqrt{\frac{2h_o}{k_f t_f}} \tag{13}$$

$$\phi = (\frac{R_{eq}}{r_c} - 1)[1 + 0.35\ln(\frac{R_{eq}}{r_c})] \tag{14}$$

$$\frac{R_{eq}}{r_c} = 1.28\frac{P_t}{r_c}(\frac{\sqrt{(P_t/2)^2 + P_l^2}}{P_t} - 0.2)^{0.5} \quad : N = 1 \tag{15}$$

$$\frac{R_{eq}}{r_c} = 1.27\frac{P_t}{r_c}(\frac{\sqrt{(P_t/2)^2 + P_l^2}}{P_t} - 0.3)^{0.5} \quad : N > 1 \tag{16}$$

Wang et al.[21] suggests that the above Schmidt[20] fin efficiency equation yields negligible error for practical applications. With Eqs. (8) to (16), an iterative procedure is needed to obtain the airside heat transfer coefficient h_o. In addition, Eq. (8) suggests that the thermal contact resistance between fins and tubes is included in h_o. In the figures, heat transfer coefficients are presented as j factors versus Re_{Dc}.

$$Re_{Dc} = \frac{\rho_a V_{max} D_c}{\mu_a} \tag{17}$$

$$j = \frac{h_o}{\rho_a V_{max} c_{pa}} Pr_a^{2/3} \tag{18}$$

All the fluid properties are evaluated at an average air temperature. The core friction factor is calculated from the measured pressure drop.

$$f = \frac{A_c}{A_o}\frac{\rho_m}{\rho_{in}}[\frac{2\Delta P \rho_{in}}{(\rho_m V_{max})^2} - (1+\sigma^2)(\frac{\rho_{in}}{\rho_{out}} - 1)] \tag{19}$$

In Eq. (19), the entrance and the exit loss coefficients are neglected following the suggestion by Wang et al.[21].

3. Results and discussions

The effect of fin pitch on j and f factors of both the convex louver fin and the wave fin samples are shown in Figs. 3 to 5 for different tube row. The fin pitch was varied from 1.81 mm to 2.54 mm. These figures show that the effect of fin pitch on j factors is

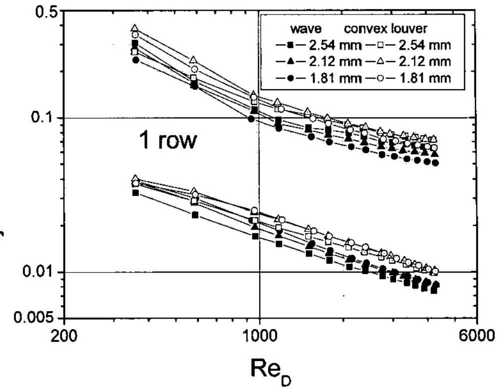

Fig. 3. Effect of fin pitch on the j and f factors for one row samples.

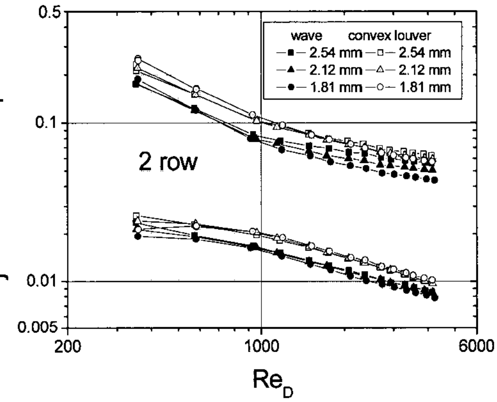

Fig. 4. Effect of fin pitch on the j and f factors for two row samples.

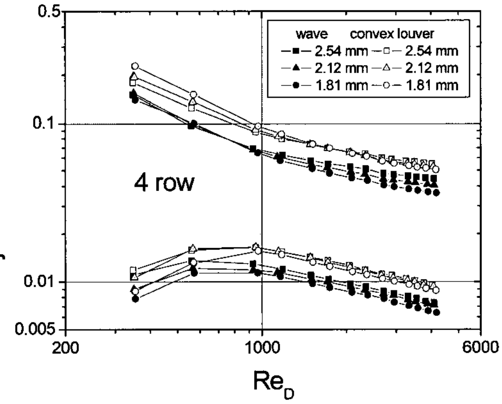

Fig. 5. Effect of fin pitch on the j and f factors for four row samples.

not significant both for the convex louver fin and for the wave fin. The independence of j factor on fin pitch has been noticed by many investigators for convex louver fin[13], wavy fin[3,5,14] and plain fin configurations[22]. Figures 3 to 5 show that f factor increases as the fin pitch increases. The increase of f factor with the fin pitch has also been reported by many researchers[3-6, 14, 23]. Torikoshi et al.[24] performed a three dimensional unsteady numerical computation for one row plate fin-and-tube heat exchangers having various fin pitches (from 1.7 mm to 3.0 mm for D_c = 10 mm). They reported that the downstream flow pattern was strongly affected by the fin pitch. As the fin pitch increased, the downstream flow field became more unsteady, which apparently increased the pressure drop of the heat exchanger. However, the flow field in the region between the fins remained steady even at the largest fin pitch. The heat transfer on the fin surface was also unaffected by the fin pitch, which yielded fin-pitch-independent j factors. Although the numerical study is limited to a plain fin configuration, similar arguments may apply to other configurations. Figures 3 to 5 shows that the effect of fin pitch on f factor is more significant for the wave fin compared with the convex louver fin. It appears that the complex fin pattern of the convex louver induces intense mixing of the flow, and thus reduces the effect of fin pitch on f factor.

For a large tube row sample as shown in Figure 5, the j factor curve shows a fall-off at low Reynolds numbers. The fall-off of the j factor curve at low Reynolds numbers has also been reported by Rich[23] and Wang et al.[3]. Wang et al.[3] argued that standing vortices, which formed behind the tubes at low Reynolds numbers, might be responsible for the decrease of the heat transfer coefficient. One other possible explanation could be that, at small Reynolds numbers of a large tube row sample, NTU values of the sample get large, and even a small uncertainty on the heat transfer measurement yields significant error on the j factor. Comparison of the convex louver fin j factors with those of wave fin reveals that convex louver fin j factors are 18% to 29% higher than those of wave fin. The f factors are 16% to 34% higher for the convex louver fin. The difference increases as the fin pitch decreases.

The effect of tube row on the j and f factor is shown in Fig. 6 for P_f = 2.12 mm. Both convex louver fin and wave fin data are shown. These figures show that both j and f factors are significantly influenced by the number of tube row. The j factors decrease as the number of tube row increases. However, as the Reynolds number increases, the effect of tube row diminishes. For fin-and-tube heat exchangers, air flows through channels formed by narrow-spaced fins and intermediate tubes. For a channel flow, the heat transfer is the largest at the inlet of the channel, and decreases downstream. Thus, we may expect larger j factor for samples having smaller number of rows (having short channels). However, at a sufficiently large Reynolds number, the turbulence generated by the tubes governs the heat transfer process, and the effect of tube row diminishes. Rich[23], Wang et al.[3,4] report the same trend for the j factor. Fig. 6 shows that the f factor also decreases as the number of tube row increases. Same arguments as those provided for the j factor may apply to the f factor. Jang and Chen[25], Min et al.[26] reported the same trend for the f factor. However, minimal dependency of the f factor on the number of tube row was reported by Wang et al.[3,4]

The literature reveals only one heat transfer and pressure drop correlation applicable to convex louver fin-and-tube heat exchangers. Wang et al.[14] developed a correlation based on their own data[13, 14]. One thing to be noted is that the convex angle is not a parameter in Wang et al. correlation, probably because all the samples had the same convex angle of 15.5°. The convex angle of the present samples is 11.7°. It is expected that the present data are overpredicted by the correlation because the correlation is based on samples having higher convex angle. The present j and f factors are compared with the predictions in Fig. 7. Figure 7 shows that the correlation overpredicts the present j factors. However, it underpredicts the f factors. The j factors are 32% ~ 72%

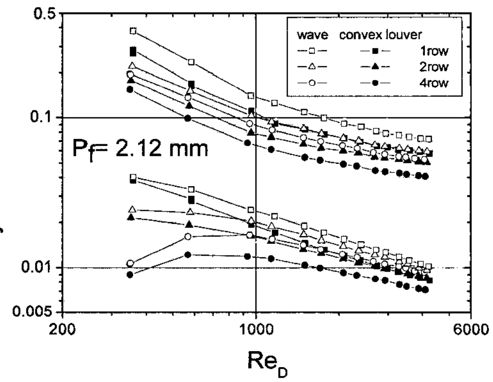

Fig. 6. Effect of tube row on the j and f factors.

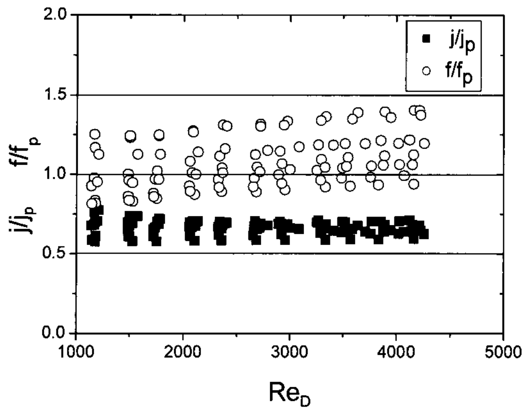

Fig. 7. The present convex louver data compared with the predictions by Wang et al.[14].

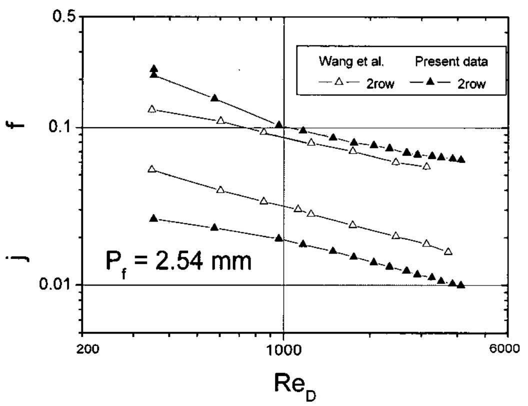

Fig. 8. The present convex louver two row data compared with those of Wang et al.[8].

(average 51%) overpredicted and the f factors are − 23% ~ 29% (average 8.3%) underpredicted. To illustrate the trend, the present two row data are compared with those of Wang et al.[13] in Fig. 8. Both samples have approximately the same tube diameter (10.06 mm for Wang et al. sample and 10.03 mm for the present sample) and the same fin pitch (2.54 mm for both samples). Figure 8 shows that the j factors of Wang et al. are 63 % higher than those of the present sample. However, the f factors are 14% lower. Generally, comparison of the fin-and-tube heat exchanger data with those from other sources is rather difficult due to the variation of fin surface geometries. Minimal differences may significantly affect the thermal performance. In addition, the fin-tube attachment depends on the manufacturing process, introducing additional thermal resistance, which is difficult to evaluate. The contact resistance is usually included in the airside heat transfer coefficient, which adds uncertainty on the comparison of the airside heat transfer coefficients.

4. Conclusions

In this study, the heat transfer and friction characteristics of the heat exchangers having convex louver fins were experimentally investigated, and the results are compared with wave fin counterparts. Eighteen samples (nine convex louver fin and nine wave fin) which had different fin pitches (1.81 mm to 2.54 mm) and tube rows (one to four) were tested. The convex angle was 11.7°. Data are compared with the existing correlation. Listed below are major findings.

1) The j factors are insensitive to the fin pitch, while f factors increase as the fin pitch increases. The effect of fin pitch on f factor is more significant for the wave fin compared with the convex louver fin. It appears that the complex fin pattern of the convex louver induces intense mixing of the flow, and thus reduces the effect of fin pitch.
2) Both the j and f factors decrease as the number of tube row increases. However, as the Reynolds number increases, the effect of tube row diminishes.
3) Comparison of the convex louver fin j factors with those of wave fin reveals that convex louver fin j factors are 18% to 29% higher than those of wave fin. The f factors are 16% to 34% higher for the convex louver fin. The difference increases as the fin pitch decreases.
4) Existing correlation fails to adequately predict the present data. More data is needed for a general correlation of the convex louver fin geometry.

References

[1] Webb, R. L. and Kim, N.-H., 2005, Principles of enhanced heat transfer, Taylor and Francis Pub.
[2] Wang, C. C., 1999, On the Airside Performance of Fin-and-Tube Heat Exchangers, in Heat Transfer Enhancement of Heat Exchangers, S. Kakac et al. Kluwer Academic Press, pp. 141-162.
[3] Wang, C. C., Fu, W. L. and Chang, C. T., 1997, Heat Transfer and Friction Characteristics of Typical Wavy Fin-and-Tube Heat Exchangers, Exp. Thermal Fluid Science, Vol. 14, pp. 174-186.
[4] Wang, C. C., Lin, Y. T., Lee, C. J. and Chang, Y. J., 1999, Investigation of Wavy Fin-and-Tube Heat Exchangers: A Contribution to Databank, Experi-

mental Heat Transfer, Vol. 12, pp. 73-89.

[5] Wang, C. C., Chang, Y. J., and Chiou, N. F., 1999, Effects of Waffle Height on the Airside Performance of Wavy Fin-and-Tube Heat Exchangers, Heat Transfer Engineering, Vol. 20, No. 3, pp. 45-56.

[6] Saiz Jabardo, J. M., Bastos Zoghbi Filho, J. R. and Salamanca, A., 2006, Experimental Study of the Airside Performance of Louver and Wave Fin-and-Tube Coils, Exp. Thermal Fluid Science, Vol. 30, pp. 621-631.

[7] Wongwises, S. and Chokeman, Y., 2004, Effect of Fin Thickness on Airside Performance of Herringbone Wavy Fin-and-Tube Heat Exchangers, Heat Mass Transfer, Vol. 41, pp. 147-154.

[8] Chokeman, Y. and Wongwises, S., 2005, Effect of Fin Pattern on the Airside Performance of Herringbone Wavy Fin-and-Tube Heat Exchangers, Heat Mass Transfer, Vol. 41, pp. 642-650.

[9] Chang, W. R., Wang, C. C., Tsai, W. C. and Shyu, R. J., 1995, Airside Performance of Louver Fin Heat Exchangers, Proceedings of the 4th ASME/JSME Thermal Engineering Joint Conference, Vol. 4, pp. 367-372.

[10] Bae, J. H., Park, M. H. and Lee, J. H., 1999, Local Flow and Heat Transfer of a 2-Row Offset Strip Fin-Tube Heat Exchanger, J. Enhanced Heat Transfer, Vol. 6, pp. 13-29.

[11] Wang, C. C., Chang, Y. P., Chi, K. Y. and Chang, Y. J., 1998, An Experimental Study of Heat Transfer and Friction Characteristics of Typical Louver Fin and Tube Heat Exchangers, Int. J. Heat Mass Transfer, Vol. 41, pp. 817-822.

[12] Hatada, T., Ueda, U., Oouchi, T. and Shimizu, T., 1989, Improved Heat Transfer Performance of Air Coolers by Strip Fins Controlling Air Flow Distribution, ASHRAE Trans., Vol. 95, Pt. 2, pp. 166-170.

[13] Wang, C. C., Chen, P. Y. and Jang, J. Y., 1996, Heat Transfer and Friction Characteristics of Convex-Louver Fin-and-Tube Heat Exchangers, Experimental Heat Transfer, Vol. 9, pp. 61-78.

[14] Wang, C. C., Tsai, Y. M., and Lu, D. C., 1998, Comprehensive Study of Convex Louver and Wavy Fin-and-Tube Heat Exchangers, J. Thermophysics Heat Transfer, Vol. 12, No. 3, pp. 423-430.

[15] ASHRAE Standard 41.1, 1986, Standard Method for Temperature Measurement.

[16] ASHRAE Standard 41.2, 1987, Standard Method for Laboratory Air-Flow Measurement.

[17] ASHRAE Standard 41.5, 1975, Standard Measurement Guide, Engineering Analysis of Experimental Data.

[18] Taborek, J., 1998, F and θ Charts for Cross-Flow Arrangements, in Heat Exchanger Design Handbook, G. F. Hewitt, Section 1.5.3, Begell House Inc.

[19] Gnielinski, V., 1976, New Equation for Heat and Mass Transfer in Turbulent Pipe and Channel Flow, Int. Chem. Eng., Vol. 16, pp. 359-368.

[20] Schmidt, T. E., 1949, Heat Transfer Calculations for Extended Surfaces, J of ASRE, Refrigeration Engineering, Vol. 4, pp. 351-357.

[21] Wang, C. C., Webb, R. L. and Chi, K. Y., 2000, Data Reduction for Airside Performance of Fin-and-Tube Heat Exchangers, Experimental Thermal Fluid Science, Vol. 21, pp. 218-226.

[22] Rich, D. G., 1973, The Effect of Fin Spacing on the Heat Transfer and Friction Performance of Multi-Row, Plate Fin-and-Tube Heat Exchangers, ASHRAE Trans., Vol. 79, Pt. 2, pp. 137-145.

[23] Rich, D. G., 1975, The Effect of the Number of Tube Rows on Heat Transfer Performance of Smooth Plate Fin-and-Tube Heat Exchangers, ASHRAE Trans., Vol. 81, Pt. 1, pp. 307-317.

[24] Torikoshi, K., Xi., G. N., Nakazawa, Y. and Asano, H., 1994, Flow and Heat Transfer Performance of a Plate Fin and Tube Heat Exchanger (First Report: Effect of Fin Pitch), Heat Transfer 1994, Proceedings of the 10th Int. Heat Transfer Conf., Vol. 4, pp. 411-416.

[25] Jang, J.-Y. and Chen, L.-K., 1997, Numerical Analysis of Heat Transfer and Fluid Flow in a Three-Dimensional Wavy Fin-and-Tube Heat Exchanger, Int. J. Heat Mass Transfer, Vol. 40, No. 16, pp. 3981-3990.

[26] Min, C.-K., Cho, J.-P., Oh, W.-K. and Kim, N.-H., 2004, Heat Transfer and Pressure Drop Characteristics of Heat Exchangers Having Plain Fins Under Dry and Wet Conditions, Korean J. Air-Conditioning and Refrigeration, Vol. 16, No. 3, pp. 218-229.