대한설비공학회 2006하계학술발표대회 논문집 pp.627~632

06-S-100

An Experimental Study on Air-side Performance of Fin-and-Tube Heat Exchangers with Slit Fin

Keun-Sun Chang†, Thanh-Long Phan

Department of Mechanical Engineering, Sunmoon University, Chungnam, 336-840, Korea

ABSTRACT: An experimental study is conducted to investigate the effect of the tube row and fin spacing on the air-side heat transfer and friction characteristics of fin-tube heat exchangers with slit fin pattern. A total of twelve samples of fin-tube heat exchangers with the nominal tube diameter of 7 mm, transverse tube pitch of 19 mm and longitudinal tube pitch of 12.5 mm are tested. The thermal fluid measurements are made using a psychometric calorimeter. The raw data are reduced to the desired heat transfer coefficient in terms of j-factor and friction factor of f for various frontal air velocities, fin pitches and number of tube rows.

Key words: Heat exchangers, Slit fin, Dry conditions, Convection heat transfer

Nomenclature

A : total heat transfer area [m^2]
A_f : total fin heat transfer area [m^2]
A_{free} : minimum flow area of air [m^2]
d_h : hydraulic diameter [$4A_{free}L/A$, m]
D_c : extended tube outside diameter [m]
f : friction factor($\Delta P \rho_a d_h / 2 G_{max}^2 L$)
G_{max} : mass flux of air flowing through the minimum flow area, A_{free} [kg/m^2s]
j : the Colburn j factor ($Nu/RePr^{1/3}$)
L : streamwise length of a heat exchanger
P_l : longitudinal tube pitch [m]
P_t : transverse tube pitch [m]
P_f : fin pitch [m]
Re_{dh} : Reynolds number based on hydraulic diameter ($G_{max}.d_h/\mu_a$)
Re_{Dc} : Reynolds number based on outside tube diameter ($G_{max}.D_c/\mu_a$)
V_{fr} : frontal velocity [m/s]

Greek symbols

ΔP : pressure drop [mmAq]
μ : viscosity [kg/ms]
η : overall surface efficiency
η_f : fin efficiency

Subscript

a : air
f : fin
w : water

†Corresponding author
Tel.:+82-41-530-2336 ; fax:+82-41-530-2986
E-mail address: chang@sunmoon.ac.kr

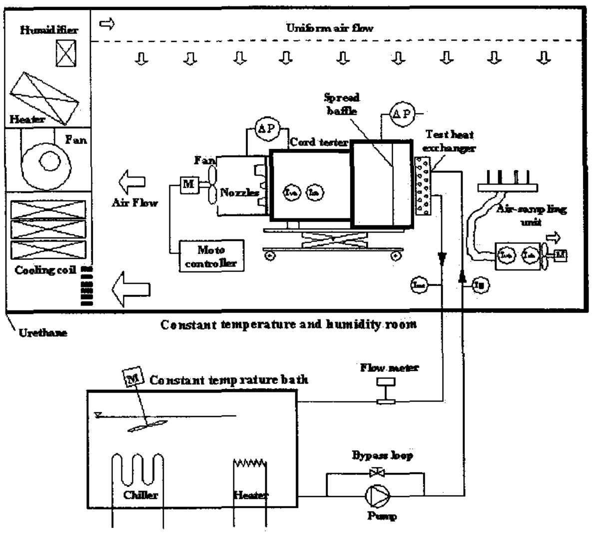
1. Introduction

Heat exchangers with interrupted fin surfaces are common in applications related to the air conditioning. Each of interrupted fin type has its own air flow pattern and different heat transfer and friction characteristics. The commonly used heat transfer enhanced interrupted fin patterns include wavy, louver, convex-louver, and slit fin (offset strip fin or parallel louver). The air side performance of the fin and tube heat exchanger is very important information since the dominant thermal resistance of fin and tube heat exchangers usually resides on the air side. In this study, the air side heat transfer and friction performances of fin and tube heat exchanger with slit fin are investigated.

For slit fin, there have been many investigations on the air side performance. The following review is not intended to exhaust, but rather to provide a background for the present study.

Naykayama and Xu[1] developed predictive correlation for the Colburn j factor and friction factor for slit fin and tube heat exchangers. They applied the appropriate heat transfer correlation to the each zone of fin that was divided into various regions. They also recommended that a slit fin and tube heat exchangers with less than four rows be used for an optimum design.

Wang et al.[2] (1999) tested twelve slit fin and tube heat exchangers which have various fin pitches and number of rows, and concluded that fin pitch has a strong effect on the performance of slit fin and tube heat exchangers. Both heat transfer coefficient and pressure drop decreased as fin pitch increased.



g. 1 Schematic of the experimental apparatus

Du and Wang[3] examined the effects of number of tube row on air side heat transfer for several heat exchangers. They concluded that the number of tube rows has a little effect on f factor and j factor, but these values decrease significantly with increasing number of tube row.

Wang et al.[4] (2001) studied heat exchangers with small tube diameter and denser fin pitch. The results revealed that the effect of the number of tube row on the heat and frictional performance for slit fin and tube

Table1 Geometric details of heat exchanger samples

Sample no	Dc [mm]	Fin shape	P_f [mm]	P_t [mm]	P_l [mm]	δ_f [mm]	N
1	7.34	plain	1.24	12.5	19	0.115	2
2	7.34	plain	1.4	12.5	19	0.115	2
3	7.34	plain	1.7	12.5	19	0.115	2
4	7.34	plain	1.24	12.5	19	0.115	3
5	7.34	plain	1.4	12.5	19	0.115	3
6	7.34	plain	1.7	12.5	19	0.115	3
7	7.34	slit	1.24	12.5	19	0.115	2
8	7.34	slit	1.4	12.5	19	0.115	2
9	7.34	slit	1.7	12.5	19	0.115	2
10	7.34	slit	1.24	12.5	19	0.115	3
11	7.34	slit	1.4	12.5	19	0.115	3
12	7.34	slit	1.7	12.5	19	0.115	3

heat exchangers is relatively small. Influence of the fin pitch on heat transfer performance is also relatively small for Re_{Dc}>1000.

Although the slit-fin heat exchangers were investigated previously, some results were quite different and the effect of parameter of slit fin and tube heat exchangers was not clear. The purpose of this study is to present the air side heat transfer and hydraulic performance of slit fin for dry conditions and to examine available correlation by comparing them with present results.

2. Experimental apparatus and test condition

Fig. 1 shows a schematic diagram of the psychometric calorimeter used in this experiment. It consists of a suction type wind tunnel, water circulation unit, air-sampling unit and data acquisition system. All apparatus is located in a constant temperature and humidity room. The suction type wind tunnel and volumetric flow rate measurement system for humid air consists of five nozzles, a fan, a motor, and an air-sampling unit.

The water circulation and control unit maintain the inlet condition of tube side at the desired value by regulating water flow rate and inlet temperature.

All data signals are collected and converted by a data acquisition system (a hybrid recorder). The data acquisition system then transmits the converted signals through GPIB interface to the computer for data recording.

The test conditions are as follows;

Dry-bulb temperature of the air inlet: 20±0.5 ℃

Inlet relative humidity: 50 %

Inlet water temperature: 60±0.5 ℃

Air velocity: 0.7-1.5 m/s (5 steps)

Water flow inside the tube: 0.78 m^3/h

3. Data reduction

Heat transfer rate of a heat exchanger can be evaluated by an air temperature change or a water temperature change as follows:

$$Q_a = \dot{m}_a C_{p,a}(T_{a,i} - T_{a,o}) \quad (1)$$

$$Q_w = \dot{m}_w C_{p,w}(T_{w,i} - T_{w,o}) \quad (2)$$

The total heat transfer is determined as:

$$Q = (Q_a + Q_w)/2 \quad (3)$$

The average overall heat transfer coefficient can be determined by:

$$Q = UA \triangle T_{LM} \quad (4)$$

In general, the total thermal resistance can be expressed as follows:

$$\frac{1}{UA} = \frac{1}{h_c A_t} + \frac{1}{h_w A_i} + \frac{1}{\eta h_a A} \quad (5)$$

Contact resistance may be determined by a correlation suggested by Sawai et al.[5] as follows:

$$\frac{h_c}{\delta_f} = 1.38 \times 10^{11} \triangle D_o + 1.62 \times 10^7 \quad (6)$$

Tube side convective heat transfer coefficient can be evaluated by Gnielinski's[6] correlation as follows:

$$N_w = \frac{(f_w/8)(Re_w - 1000)Pr_w}{1 + 12.7\sqrt{f_w/8}\,(\mathrm{Pr}^{2/3} - 1)} \quad (7)$$

$$f_w = (1.82 ln Re_w - 1.64)^{-2}$$

Overall surface efficiency has a relationship with fin efficiency,

$$\eta = 1 - (A_f/A)(1 - \eta_f) \quad (8)$$

where the fin efficiency can be calculated by Schmidt's correlation[7],

$$\eta_f = tanh\left(\frac{\beta D_c \phi}{2}\right) / \left(\frac{\beta D_c \phi}{2}\right) \qquad (9)$$

Reynolds number of air is defined based on hydraulic diameter as;

$$Re_{d_h} = G_{\max} d_h / \mu_a \qquad (10)$$

where the hydraulic diameter at the minimum free flow area is defined as follows:

$$d_h = 4 A_{free} L / A \qquad (11)$$

Reynolds number based on the outer diameter of the tube including fin collar (D_c) is,

$$Re_{Dc} = G_{\max} D_c / \mu_a \qquad (12)$$

Air heat transfer coefficient in terms of Colburn j factor and frictions factor can be expressed as follows;

$$j = \frac{h_a}{G_{\max} C_{p,a}} \mathrm{Pr}^{2/3} \qquad (13)$$

$$f = \frac{\triangle P \rho_a d_h}{2 G_{\max}^2 \cdot L} \qquad (14)$$

4. Results and Discussion

The experimentally determined values of Colburn j factor and friction factor f for dry conditions of the slit and plate fin-and-tube heat exchangers are shown in Fig. 2 and 3. The fin pitches of each fin type are 1.24, 1.4 and 1.7 mm. Reynolds numbers (Re_{Dc}) based on tube outside diameter are in the range from 570 to 1260. As seen in the figures, the heat transfer coefficients and friction factors of slit fin are greater than those of plain fin. For all type of interrupted surface, the similar results were found in many previous investigations. In this study, the heat transfer coefficients of slit fin with two and three rows are approximately 160 % and 16 % larger than those of plain fin, respectively. On the other hand, the friction factors of slit fin are approximately 80 % and 85 % larger than those of plain fin, respectively.

The effects of the fin pitch of slit fin on the heat transfer and friction characteristic are also shown in Fig. 2 and 3. The effects of fin spacing on the heat transfer and friction characteristics are not significant. The similar trend was also reported by some previous investigators. Du & Wang [3] tested 30 heat exchanger samples with slit fin and different tube diameters for the frontal velocities from 0.25 m/s to 7 m/s, and reported that for N=1 the heat transfer performance increases with the decrease of fin pitch, but for N>2 the effect is reversed. Wang et al.[4], who tested 6 heat exchangers with fin pitch of 1.2 mm and 1.8 mm and tube row of N=1, 2 and 3, reported that the effects of fin pitch on the heat transfer were not significant.

Fig. 4 illustrates the effects of the number of tube rows on the heat transfer and friction characteristics for dry conditions. The heat transfer coefficients of two row heat

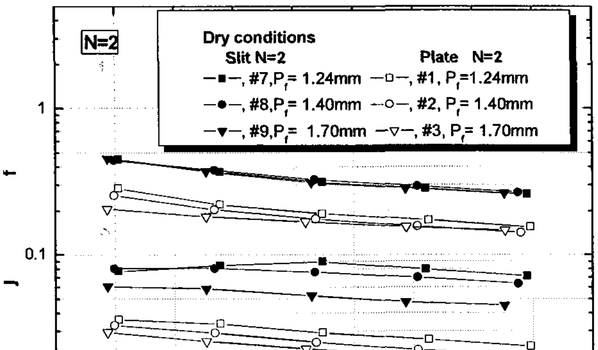


Fig. 2 Comparison of slit fins with plain fins, two-row configuration

exchangers are higher than those of three rows for all fin pitches. The values increase with decreasing fin pitches and this tendency is more evidenced for two row heat exchangers. With fin pitch of 1.2 mm (D_c=7.6 mm, P_l= 21 mm, P_t=12.7 mm), Wang et al[4] reported that the heat transfer coefficient drops sharply with the increase of tube row for Re_{Dc}< 1000. Mochizuki et al.[8] reported that steady laminar flow pattern prevailed throughout the core of slit (offset slit) geometry at the low Reynolds number region. This implies that heat transfer performance may be determinate significantly as the depth of core increases. However, it is certain that friction factors for three row heat exchangers are higher than those of two row cases and slightly increase with decreasing fin pitches.

In Fig. 5, the present data are compared with the available correlations in references [1,2,4] for dry conditions. Most of samples of previous studies have larger tube diameter, larger longitudinal pitch and larger transverse pitch than ours. Only Wang et al.[4] tested slit fin-and-tube heat exchangers with small longitudinal and transverse tube pitch and tube diameter similar to those of our cases. As seen in the Fig. 5, the j factors of Wang et al.[2](1999) and Wang et al[4] (2001) correlations overpredict in three row heat exchangers, but underpredict in two row cases. The f factors of Nakayama and Xu[1] correlation falls in higher range, but Wang et al.[4](2001) correlation underpredicts both for two and three row heat exchangers.

4. Conclusions

An experimental study was carried out for the heat transfer and friction characteristics of slit and plain fin-and-tube heat exchangers under dry conditions. The important conclusions made in this study are summarized as follow;

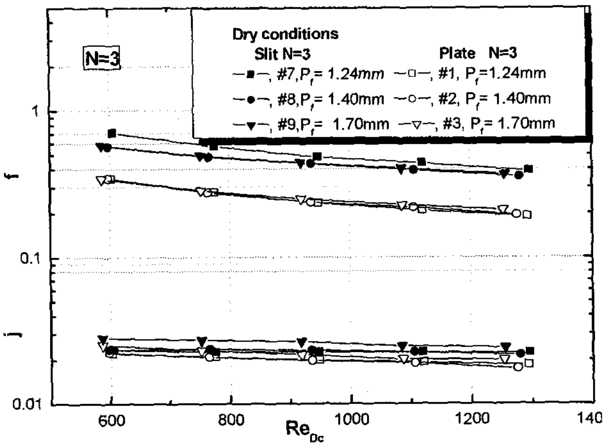


Fig. 3 Comparison of slit fins with plain fins, three-row configuration.

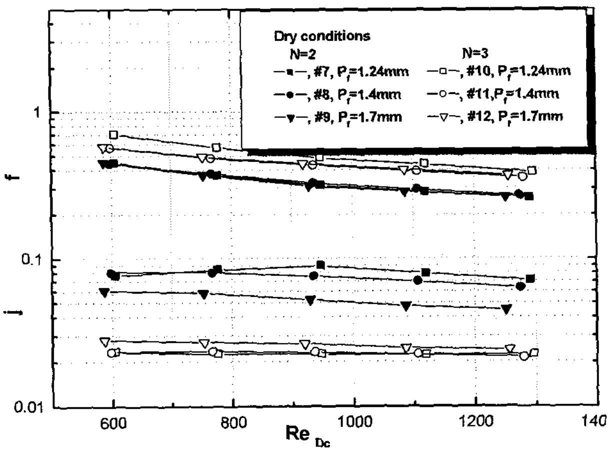
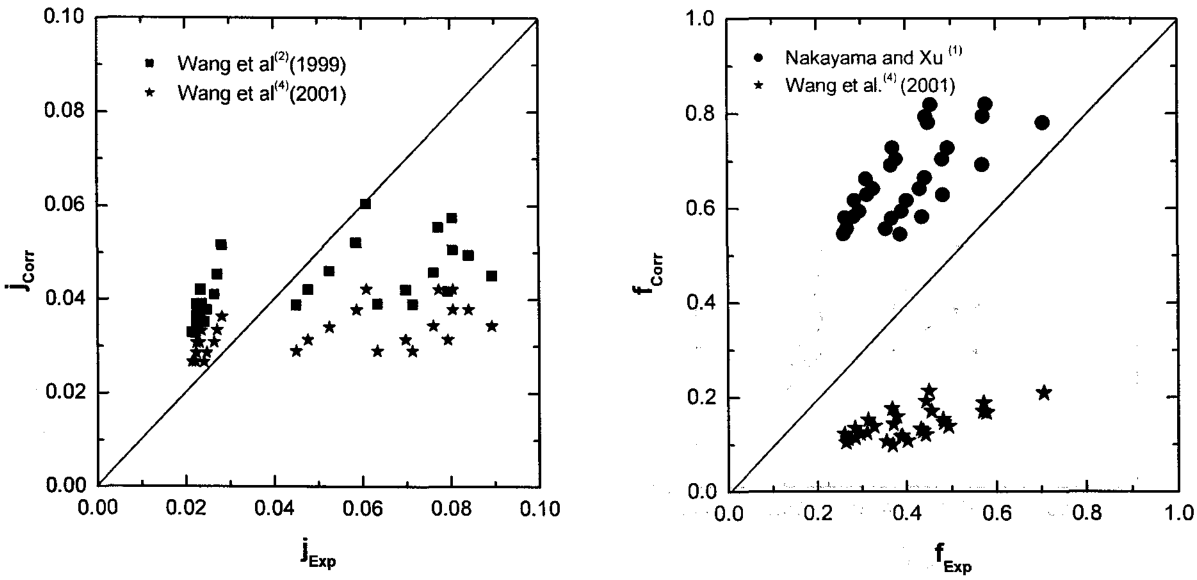


Fig. 4 Effect of tube row on the air side performance for slit fin-tube heat exchanger

(1) The effects of fin pitch on the heat transfer coefficient and friction factor are not significant both for slit and plain fin geometries.

(2) In dry conditions, two row slit fin heat exchangers exhibit a lower friction factor and higher heat transfer coefficient than those of three row cases.

(3) The j factor correlations of other researchers overpredict in three row heat exchangers, but underpredict in two row cases. However the f factors of other studies do not represent similar trends.



(a) j factor (b) f factor

Fig. 5 Comparison of present data with existing correlations for dry conditions.

Acknowledgement

This work was supported by RRC-ACCT at Sunmoon University and the authors would like to appreciate it.

References

1. Nakayama, W., Xu. L. P., 1983, Enhanced fin for air cooled heat exchangers-heat transfer and friction correlation. 1st ASME/JSME. Thermal Eng. Joint Conference, pp. 495-502.
2. Wang, C. C., Tao, W. H., Chang, C. J., 1999, An investigation of the airside performance of the slit fin-and-tube heat exchangers, Int. J. of Refrigeration, Vol. 22, pp. 595-60
3. Du, Y. J., Wang, C. C., 2000, An experimental study of the air-side performance of the superslit fin-and-tube heat exchangers, Int. J. of Heat and Mass Transfer, Vol. 43, pp. 4475-4482.
4. Wang, C. C., Lee, W. S., Sheu, W. J., 2001, A comparative study of compact enhanced fin-and-tube heat exchangers, Int. J. of Heat and Mass Transfer, Vol. 44, pp. 3565-3573.
5. Sawai, S., Hayashi, T., Ohtake, Y., Takei, T., 1969, Effects of mechanical bond between fin and tube on heat transfer, Refrigeration, vol.41, No. 502, pp. 15-21.
6. Gnielinski, V., 1976, New equation for heat and mass transfer in turbulent pipe and channel flow, Int. Chem. Eng., Vol. 16, No2, pp. 359-368.
7. Schmidt, T. E., 1949, Heat transfer calculation for extended surface, J of ASHRAE, Refrigerating Engineering, Vol.4, pp. 351-357.
8. S. Mochizuki, Y. Yagi, W. J. Yang,. 1988, Flow pattern and turbulence intensity in stacks of interrupted parallel surfaces. Experimental Thermal and Fluid Science, 1, pp. 51-57.