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An Experimental Study of the Airside Performance of Slit Fin-and-Tube Heat Exchangers under Dry and Wet Conditions

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Abstract

Water condensate accumulated on the surface of a fin-and-tube heat exchanger significantly affects its thermal and hydraulic performances. The purpose of this study is to investigate the effects of condensate retention on the air-side heat transfer performance and flow friction for various flow and geometric conditions. Total of twelve samples of slit and plate fin-and-tube heat exchangers are tested under dry and wet conditions. The thermal fluid measurements are made using a psychrometric calorimeter. Frontal air velocity varies in the range from 0.7 m/s to 1.5 m/s. Using the experimental data, presented are heat transfer coefficients in terms of Colburn *j*-factors and friction factors, and these data are compared with the existing correlations.

Key words: Fin-tube heat exchanger, Slit fin, Dry condition, Wet condition, Dehumidification

Nomenclature

A : total heat transfer area [m^2]
A_i : tube inside heat transfer area [m^2]
A_t : tube outside heat transfer area [m^2]
A_f : total fin heat transfer area [m^2]
A_{free} : minimum flow area of air [m^2]
b : slope of the enthalpy temperature curve of saturated air [KJ/kgK]
C_p : constant pressure specific heat [KJ/kg ℃]
d_h : hydraulic diameter [$4A_{free}L/A$, m]
d_i : tube inside diameter [m]
D_o : tube outside diameter [m]
D_c : extended tube outside diameter [m]
f : friction factor
G_{max} : mass flux of air flowing through the minimum flow area, A_{free} [kg/m^{2}s]
h : heat transfer coefficient [KW/ m^2 K]
i : enthalpy [kJ/kg]
j : Colburn j factor ($Nu/RePr^{1/3}$)
k : thermal conductivity [W/mK]
$\dot{m}$: mass flow rate [kg/s]
L : streamwise length of heat exchanger
N : number of tube row
P_l : longitudinal tube pitch [m]
P_t : transverse tube pitch [m]
P_f : fin pitch [m]
Re_{Dc} : Reynolds number based on outside tube diameter ($G_{max}D_c/\mu_a$)
V_{fr} : frontal velocity [m/s]
U : overall heat transfer coefficient [KW/m^2 K]
U_w : wet surface overall heat transfer coefficient [kg/m^{2}s]

Greek symbols

ΔD_o : tube diameter difference after expansion (D_c - D_o) [m]
ΔP : pressure drop [mmAq]
δ_f : fin thickness [m]
η : overall surface efficiency
η_f : fin efficiency
μ : density [kg/m^3]
ρ : thickness [m]

Subscript

a : air
c : contact between fin collar and tube
f : fin

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i : inlet
m : mean
o : outlet
r : tube side
w : water

1. Introduction

Fin-and-tube heat exchangers are frequently employed in air handling systems such as air conditioning heat exchangers, fan coils, cooling or heating coils. Water vapor in humid air condenses on the air-side surface of the heat exchanger when the surface temperature is below the dew point. The condensed water grows along with the flow direction and is extracted downstream through the cooling process. Some of the condensate may bridge between fins. Since heat and mass transfer with condensate occurs on the surface simultaneously, there are significant changes of heat transfer and friction characteristics, dependent on the amount and forms of condensation. The air flow across the heat exchanger interacts with water condensate, which makes the flow pattern more complex.

Recently, slit-fin with small size of tube diameter(7 mm) is widely used in household air conditioning heat exchanger applications. This is because the higher heat transfer performance and the smaller space area can be achieved when the tube diameter becomes smaller.

There are several previous investigations associated with the air side heat transfer performance of slit fin-tube heat exchangers, but most of the data are obtained for dry conditions[1-5]. Nakayama and Xu[5] presented test results of three samples of slit fin-tube heat exchangers and proposed a correlation based on their results. Wang et al.[1] tested twelve samples with various fin pitches and number of tube rows. Based on Nakayama and Xu[5]'s correlation, Wang et al.[1] proposed a new correlation and pointed out the applicable limit of Nakayama and Xu's correlation[5].

Although the slit-fin is widely used in fin-tube heat exchangers and has been studied previously, there are not sufficient data to be used for heat exchanger design, especially for wet conditions. The objective of this study is to present the air side heat and hydraulic performances of slit-fin patterns under dry and wet conditions and to examine available correlations by comparing them with the present results. Twelve samples of fin-tube heat exchangers are used in this study. These are plate and slit fin tubes of 7.0 mm in diameter with different fin pitches and number of tube rows. Geometric details of the heat exchanger samples are listed in Table 1.

2. Experimental apparatus and test condition

Fig. 1 shows a schematic diagram of the psychrometric 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 chamber. The suction type wind tunnel and volumetric flow rate measurement system for humid air consist of five nozzles, a fan, a motor, and an air-sampling unit.

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

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 General Purpose Interface Bus(GPIB) to the computer for data recording. The uncertainty of the calorimeter for the air side heat transfer rate measurement and heat balance between air and water are less than ±1.5% and 3%, respectively when the steady state conditions of air and water exit temperatures are achieved.

The tested fin patterns in this study are slit and plain fins. Each has three different fin pitches and two different tube rows(Table 1). For water flow, two

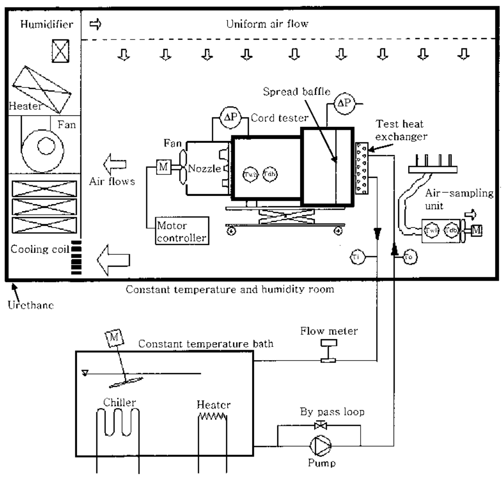


Fig. 1. Schematic diagram of test apparatus.

Table 1. Geometric details of heat exchanger samples.

Sample no	Dc [mm]	Fin shape	P_f [mm]	P_t [mm]	P_l [mm]	δ_f	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

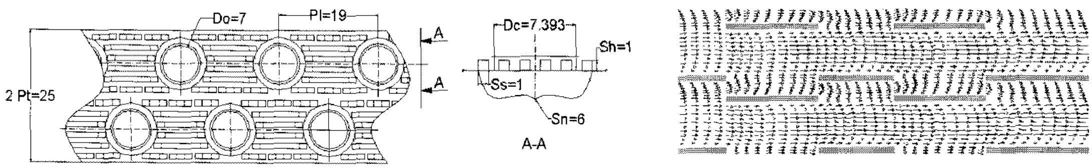


Fig. 2 (a) Details of the slit fin geometry. (b) Schematic of the airflow pattern of the slit fin geometry.

divided circuits are used for all heat exchangers and two pass flow configurations are used for two row heat exchangers, but three passes for three row heat exchangers.

Fig. 2(a) shows the slit fin pattern used in this study and Fig. 2(b) indicates flow pattern along the slit fin. The boundary layer develops at the leading edge of the fin and breaks down at the trailing edge. This flow pattern repeats at the subsequent fins and thus enhances the heat transfer. The test conditions are as follows;

Dry-bulb temperature of the air inlet:
20±0.5℃ (for dry operation)
27±0.5℃ (for wet operation)

Inlet relative humidity: 50%

Inlet water temperature:

60±0.5℃ (for dry operation)
5±0.5℃ (for wet operation)
Air velocity: 0.7-1.5 m/s (5 steps)
Water flow inside the tube: 0.78 m^3/h

3. Data reduction

3.1 Dry condition

Heat transfer rate of a heat exchanger in dry condition can be evaluated by an air enthalpy change or a water enthalpy change as follows:

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

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

The total heat transfer rate between the fluids, in the present work, is determined as an arithmetic average of these two values,

$$Q = \frac{Q_a + Q_w}{2} \tag{3}$$

Since the inlet and outlet temperatures of air and water are measured, a log-mean-temperature-difference(ΔT_{LM}) can be calculated. Then, the average overall heat transfer coefficient (U) can be determined from the following relationship,

$$Q = UA\Delta T_{LM} \tag{4}$$

In general, the total thermal resistance of a fin-tube heat exchanger is the sum of convective resistances both at the inside and outside tube fluids, conductive resistance through the tube wall and contact resistance between the tube and fin collar. In the present study, the conductive resistance is neglected because through the very thin copper tube the value is accounted for less than 1% of the total resistance. Then, the total thermal resistance can be expressed as follows:

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

In order to evaluate the air-side heat transfer coefficient from the above equation, several parameters need to be determined. Contact resistance may be determined by a correlation suggested by Sawai et al.[6] as follows:

$$\frac{h_c}{\delta_f} = 1.38\times10^{11}\Delta D_o + 1.62\times10^{7} \tag{6}$$

Tube side convective heat transfer coefficient can be evaluated by Gnielinski's[7] correlation as follows:

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

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

Overall surface efficiency has a relationship with fin efficiency,

$$\eta = 1 - \frac{A_f}{A}(1-\eta_f) \tag{8}$$

where the fin efficiency can be calculated by Schmidt's[8] correlation,

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

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

$$Re_{DC} = G_{\max} D_c / \mu_a \tag{10}$$

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

$$j = \frac{h_a Pr^{2/3}}{G_{\max} C_{p,a}} \tag{11}$$

$$f = \frac{\Delta P \rho_a d_h}{2G_{\max}^2 \cdot L} \tag{12}$$

3.2 Wet condition

When a heat exchanger is at the wet condition and its surface temperature is below the dew point, it is subjected to accumulation of condensation water. The analysis of this wet condition requires consideration of heat and mass transfer simultaneously. The approach used here for data reduction employs the mean different enthalpy method. Heat transfer rate of a heat exchanger in wet condition can be expressed as,

$$Q_w = \dot{m}_w C_p \Delta T_w \tag{13}$$

$$Q_a = Q_{lat} + Q_{sen} \tag{14}$$

Q_{lat} is latent heat and Q_{sen} is sensible heat. The total heat transfer rate between the fluids is again determined as an arithmetic mean of these two values. The average overall heat transfer coefficient is obtained from the following equation,

$$Q = U_w A \Delta i_m \tag{15}$$

For humid air, air enthalpy change Δi_m can be obtained as,

$$\Delta i_m = \frac{i_{a,i} - i_{a,o}}{\ln\left(\dfrac{i_{a,i} - i_{s,r}}{i_{a,o} - i_{s,r}}\right)} \tag{16}$$

In equation (16), $i_{s,r}$ is the saturated air enthalpy at tube side water temperature. The overall enthalpy transport coefficient for the wet condition is;

$$\frac{1}{U_w A} = \frac{b_{w,m}}{\eta_{wet} h_{wet} A} + \frac{b_r}{h_w A_i} + \frac{b_t \delta_f}{k A_f} \tag{19}$$

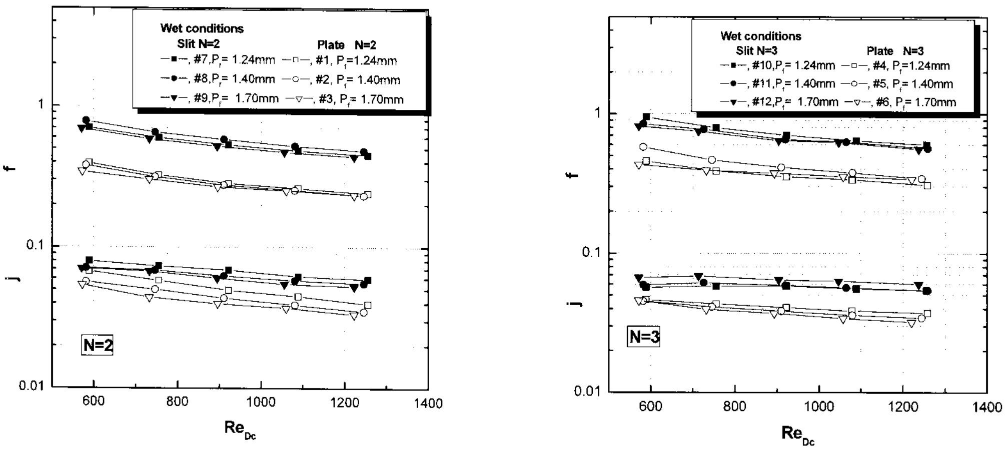
Here, b_r, b_t and $b_{w,m}$ are the saturated air enthalpy temperature slopes at the water, tube wall and water film temperature, respectively. The wet surface fin efficiency (η_{wet}) is calculated by Kim and Sim[9] method.

The uncertainty of contact thermal resistance and tube-side convective resistance are assumed to be less than ±5% in this calculation. Based on the error propagation analysis, uncertainties of water heat transfer rate, air heat transfer rate and air heat transfer coefficient are estimated to be less than ±3%, ±5% and ±6%, respectively.

4. Results and discussion

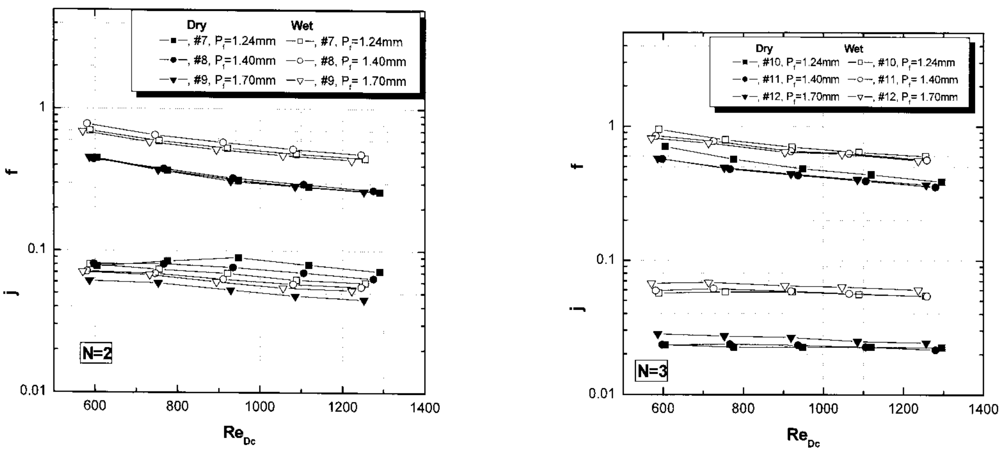
Fig. 3(a) and (b) present the values of Colburn j factor and friction factor f for wet conditions of the slit and plate fin-and-tube heat exchangers. The fin pitches of each fin type are 1.24, 1.4 and 1.7 mm. Reynolds numbers(Re_{DC}) based on tube outside diameter are in the range from 570 to 1260. As seen in the figure, the heat transfer coefficients and friction factors of slit fin are greater than those of plate fin. For all type of interrupted surface, the similar results were found in many previous investigations[1-5]. In this study, the heat transfer coefficients of slit fin with two and three rows are approximately 42% and 52% larger than those of plate fin, respectively. On the other hand, the friction factors of slit fin are approximately 90% and 80% larger than those of plate fin, respectively.

Fig. 4(a) and (b) illustrate the effects of the fin pitch of slit fin on the heat transfer and friction characteristic both for dry and wet conditions. As seen in the



(a) N=2 (b) N=3

Fig. 3. Comparison of j & f factors between slit and plain fin-and-tube heat exchangers in wet conditions.



(a) N=2 (b) N=3

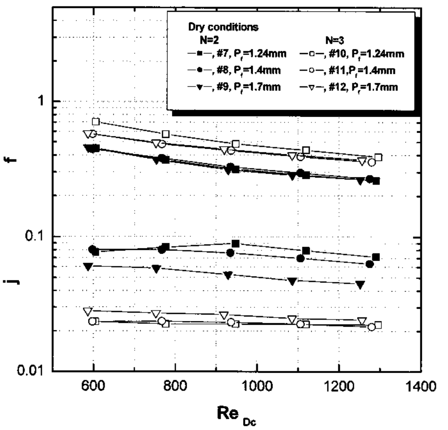
Fig. 4. Effect of fin-pitch on the air side performance for slit fin-tube heat exchanger in dry and wet conditions.

figure, the Colburn j-factors in all cases decrease slightly with increasing Reynolds number, but the friction factors decrease more steeply as the Reynolds number increases. The friction factors in the wet condition are much higher than those for dry conditions both for N=2 and 3 because the retention of condensate increases the flow resistance with the wet surface. On the other hand, the heat transfer coefficients for N=2 do not exhibit clear difference between dry and wet conditions, but for N=3 the values in wet conditions are considerably higher than those of dry conditions. This is because the short distance of fin length for N=2 is not sufficient for condensate formation and extraction. The differences of flow pattern between wet and dry condition may be explained by two reasons. First, when fin-and-tube heat exchanger is operated at the dehumidifying condition, the condensate retention may become severer as fin spacing decreases. Second, as the fin pitch decreases, the condensate may considerably damage flow pattern across the heat exchanger. As a consequence, the whirl flow is generated and it may produce extra friction loss as the fin spacing decreases. However, the effects of fin spacing on the heat transfer and friction characteristics are not significant both in dry and wet conditions. Du & Wang[2] 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.[3], 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.

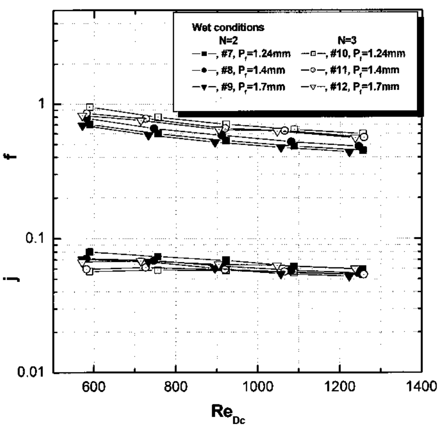
Fig. 5(a) illustrates the effects of the number of tube rows on the heat transfer and friction characteristics for dry conditions. The heat transfer coefficients of two row heat exchangers are higher than those of three rows for all fin pitches. The values increase with decreasing fin pitches and this 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[3] reported that the heat transfer coefficient drops sharply with the increase of tube row for Re_{Dc}< 1000. Mochizuki et al.[10] 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.

Fig. 5(b) shows the wet condition *j* and f factors for the same samples as those of Fig. 5(a). Since the condensate formation and extraction are well facilitated with three rows and this condensate acts as a heat transfer enhancement and friction increase mechanism, the *j* and *f* factors for three rows approach to the values of two rows.

In Fig. 6, the present data are compared with the available correlations in references [1,3,5] for dry conditions. Although heat exchanger parameters used in these references are different from our cases, the gen-

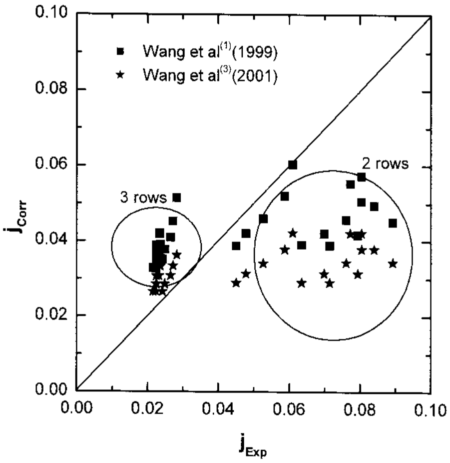


(a) Dry conditions

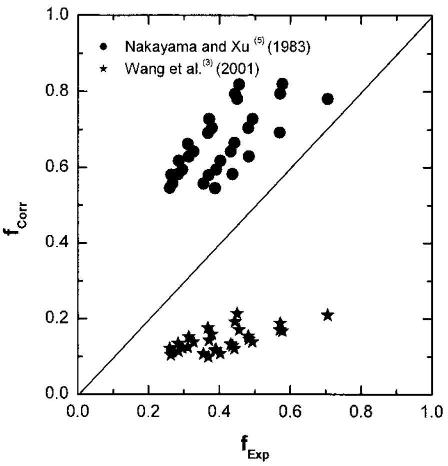


(b) Wet conditions

Fig. 5. Effect of number of row on the air-side performance for slit fin-and-tube heat exchanger.



(a) j factor



(b) f factor

Fig. 6. Comparison of present data with existing correlations for dry conditions.

eral trends are observable and comparable. Most of samples of previous studies have larger tube diameter, larger longitudinal pitch and larger transverse pitch than ours. Only Wang et al.[3] 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. 6, the j factors of Wang et al.[1](1999) and Wang et al[3] (2001) correlations overpredict in three row heat exchangers, but underpredict in two row cases. The f factors of Nakayama and Xu[5] correlation falls in higher range, but Wang et al.[3](2001) correlation underpredicts both for two and three row heat exchangers.

5. Conclusions

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

(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) For wet conditions, two row slit fin heat exchangers result in a lower friction factor than that for three rows, but the differences become smaller than the case of dry conditions, while the heat transfer coefficients do not reveal clear differences between two and three rows.

(4) The j factor correlations of Wang et al.[1](1999) and Wang et al[3] (2001) overpredict by 30% and 40% in three row heat exchangers, but underpredict by 35% and 25% in two row cases. However, the f factors of other studies do not represent similar trends.

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References

[1] Wang, C. C., Tao, W. H and 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. 595-603.

[2] Du, Y. J and 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.

[3] Wang, C. C., Lee, W. S and 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.

[4] Dejong, N. C and Jacobi, A. M., 1997, An experimental study of flow and heat transfer in parallel-plate arrays: local, row-by-row and surface average behavior, Int. J. Heat and Mass Transfer, Vol. 40, pp. 1365-1378.

[5] Nakayama, W and Xu. L. P., 1983, Enhanced fin for air cooled heat exchangers-heat transfer and friction correlation. 1st ASME/JSME. Thermal Engineering Joint Conference, pp. 495-502.

[6] Sawai, S., Hayashi, T., Ohtake, Y and Takei, T., 1969, Effects of mechanical bond between fin and tube on heat transfer, Refrigeration, vol.41, No. 502, pp. 15-21.

[7] Gnielinski, V., 1976, New equation for heat and mass transfer in turbulent pipe and channel flow, Int. Chem. Engineering, Vol. 16, No2, pp. 359-368.

[8] Schmidt, T. E., 1949, Heat transfer calculation for extended surface, J of ASHRAE, Refrigerating Engineering, Vol.4, pp. 351-357.

[9] Kim, N.H and Sim, Y.S., 2004, Reduction of the wet surface heat transfer coefficients from experimental data, Int. J. of Air-Conditioning and Refrigeration Korea, Vol. 12, pp. 37-39.

[10] Mochizuki, S, Yagi, Y and Yang, W.J., 1988, Flow pattern and turbulence intensity in stacks of interrupted parallel surfaces. Experimental Thermal and Fluid Science, 1, pp. 51-57.

[11] Wang, C. C., Lin, Y. T and Lee, C. J., 2000, Heat momentum for compact louver fin-and-tube heat exchangers in wet condition, Int. J. of Heat and Mass Transfer, Vol. 43, pp. 3443-3452.

[12] Wang, C. C., Webb, R. L and Chi, K. Y., 2000, Data reduction for air-side performance of fin-and-tube heat exchangers. Experimental Thermal and Fluid Science, Vol. 21, pp. 218-226.

[13] ASHRAE 33-87, 1987, Method of testing forced circulation air cooling and air heating coils.