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Technical Note

Airside performance of fin-and-tube heat exchangers in dehumidifying conditions – Data with larger diameter

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ABSTRACT

This study presents the airside performance of the fin-and-tube heat exchangers having plain fin geometry with a larger diameter tube (D_c = 15.88 mm) under dehumidifying condition. A total of nine samples of heat exchangers subject to change of the number of tube row and fin pitch are made and tested. It is found that the effect of fin pitch on the sensible j factor is, in general, diminished with the rise of tube row. However, there is a unique characteristic of fin pitch at a shallow tube row, the heat transfer performance is first increased at a wider pitch but a further increase of fin pitch lead to a falloff of heat transfer performance due to interactions amid flow development and bypass flow. The influence of tube row on the airside performance is rather small for both heat transfer and frictional characteristics at a fin pitch of 2.1 mm and when the Reynolds number is less than 4000. A slight deviation of this effect is encountered when fin pitch is increased to 2.54 mm or 3.1 mm due to condensate adhered phenomena.

1. Introduction

Fin-and-tube heat exchangers are widely-used heat transfer devices in applications like refrigeration and air conditioning systems. Its easier manufacturing, simpler construction, lower cost, and relatively easy in maintenance makes it one of the most commonly used heat exchangers. They can be applicable to both heating and cooling. Once the cooling process takes place below dew point temperature, condensate forms on the surface and result in a complex heat and mass transfer interactions. The heat transfer characteristics of fin-and-tube heat exchangers under this dehumidifying conditions had been studied by many researchers (e.g. McQuiston [1,2], Beecheer and Fagan [3], Yan and Sheen [4], Mirth and Ramadhyani [5,6], Wang et al. [7], Pirompugd et al. [8,9]). The published literatures offer considerable test results of plain fin data in wet condition.

However, the foregoing tests were conducted for typical heat exchangers of small air-conditioners where nominal tube diameters of 9.52, 7.94 or 7 mm were generally employed. In typical applications like fan-coil or ventilator, exploitation of larger diameter like 15.88 mm is also very common. Unfortunately, there is very limited performance data of the fin-and-tube heat exchanger with larger diameter tube in the open literature and is virtually no data available in dehumidifying conditions. Hence, the objective of the present study is to provide relevant performance data to the database.

2. Experimental setup

The schematic diagram of the experimental air circuit assembly is shown in Fig. 1. It consists of a closed-loop wind tunnel in which air is circulated by a variable speed centrifugal fan (7.46 kW, 10 HP). The air duct is made of galvanized sheet steel and has an 850 mm × 550 mm cross-section. The dry-bulb and wet-bulb temperatures of the inlet-air are controlled by an air-ventilator that can provide a cooling capacity of up to 21.12 kW (6RT). The air flow-rate measurement station is an outlet chamber set up with multiple nozzles. This setup is based on the ASHRAE 41.2 standard [10]. A differential pressure transducer is used to measure the pressure difference across the nozzles. The air temperatures at the inlet and exit zones across the sample heat exchangers are measured by two psychrometric boxes based on the ASHRAE 41.1 standard [11].

The working medium for the tube side is cold water. A thermostatically controlled reservoir provides cold water at selected temperatures. The temperature differences on the water side are measured by two pre-calibrated RTDs. The water volumetric flow

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Nomenclature

A_c	minimum flow area, m^2
A_f	fin surface area, m^2
$A_{p,i}$	inside surface area of tubes, m^2
A_o	total surface area, m^2
b'_p	slope of the air saturation curved between the outside and inside tube wall temperature, J kg^{-1} K^{-1}
b'_r	slope of the air saturation curved between the mean water temperature and the inside wall temperature, J kg^{-1} K^{-1}
$b'_{w,f}$	slope of the air saturation curved at the mean water film temperature of fin surface, J kg^{-1} K^{-1}
$b'_{w,p}$	slope of the air saturation curved at the mean water film temperature of tube surface, J kg^{-1} K^{-1}
$C_{p,a}$	moist air specific heat at constant pressure, J kg^{-1} K^{-1}
D_c	collar diameter, m
D_h	hydraulic diameter, m
f	Fanning friction factor
F_p	fin pitch, m
G_c	maximum mass flux at minimum flow area, kg m^{-2} s^{-1}
$h_{c,o}$	sensible heat transfer coefficient, W m^{-2} K^{-1}
h_i	inside heat transfer coefficient, W m^{-2} K^{-1}
j	Colburn j factor
k_p	thermal conductivity of tube, W m^{-1} K^{-1}
L	streamwise duct length, m
N	number of tube row
P_l	longitudinal tube pitch, m
P_t	transverse tube pitch, m
Pr	Prandtl number of air
ΔP	pressure drop, Pa
$\dot{Q}_a$	air side heat transfer rate, W
$\dot{Q}_{avg}$	average heat transfer rate, W
$\dot{Q}_r$	water side heat transfer rate, W
Re_{Dc}	air side Reynolds number based on the collar diameter
Re_{Dh}	Reynolds number based on hydraulic diameter
RH_{in}	inlet relative humidity
$U_{o,w}$	wet surface overall heat transfer coefficient, based on enthalpy difference, kg m^{-2} s^{-1}
x_p	wall thickness, m
x^+	inverse Graetz number
σ	contraction ratio
$\eta_{f,wet}$	fully wet fin efficiency
ρ	air density, kg m^{-3}
δ_f	fin thickness, m

rate is measured by a magnetic flow meter with a ±0.001 L/s precision. All the temperature measuring probes are resistance temperature devices (Pt100), with a calibrated accuracy of 0.05 °C. In the experiments, only the data that satisfy the ASHRAE 33-78 [12] requirements (namely, the energy balance condition, $|\dot{Q}_r - \dot{Q}_a|/\dot{Q}_{avg}$, is less than 0.05, where $\dot{Q}_r$ is the water-side heat transfer rate for and $\dot{Q}_a$ air-side heat transfer rate) are considered in the final analysis. Detailed geometry used for the present plain fin-and-tube heat exchangers is tabulated in Table 1. The test fin-and-tube heat exchangers are tension wrapped having a "L" type fin collar. The test conditions of the inlet-air are as follow:

Dry-bulb temperature of the air	27 ± 0.5 °C
Inlet relative humidity for the incoming air	50% and 80%
Inlet-air velocity	From 1 to 4 m/s
Inlet-water temperature	7 ± 0.5 °C
Water velocity inside the tube	1.5–1.7 m/s

The test conditions approximate those encountered with typical fan-coils and evaporators of air-conditioning applications. Uncertainties reported in the present investigation, following the single-sample analysis proposed by Moffat [13]. The maximum uncertainty occurred at the smallest frontal velocity and is less than ±4.7% for reduction of the sensible heat transfer coefficient whereas it is within ±6% for the frictional reduction.

3. Data reduction

Basically, the present reduction method is analogous to Threlkeld's approach [14]. Details of the reduction process can be found from the previous studies by Wang et al. [7]. Notice that the Threlkeld method is an enthalpy-based reduction method. A brief description of the reduction of heat and mass transfer is given as follows:

The overall heat transfer coefficient is related to the individual heat transfer resistance (Myers, [15]) as follows;

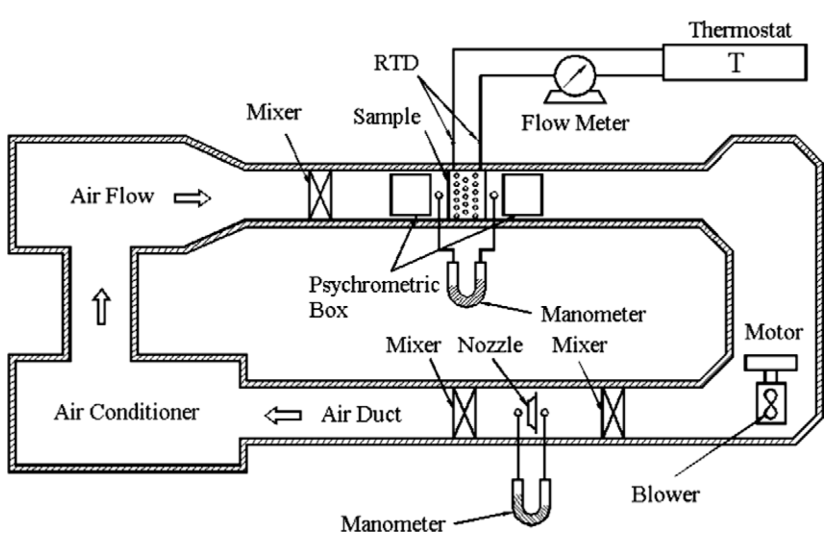

Fig. 1. Schematic of the test apparatus.

Table 1
Detailed geometric parameters of the test samples.

No.	Fin type	F_p (mm)	δ_f (mm)	D_c (mm)	P_t (mm)	P_l (mm)	N, row
1	Plain	2.12	0.12	16.68	38.1	33	2
2	Plain	2.54	0.12	16.68	38.1	33	2
3	Plain	3.17	0.12	16.68	38.1	33	2
4	Plain	2.06	0.12	16.68	38.1	33	4
5	Plain	2.54	0.12	16.68	38.1	33	4
6	Plain	3.13	0.12	16.68	38.1	33	4
7	Plain	2.12	0.12	16.68	38.1	33	8
8	Plain	2.54	0.12	16.68	38.1	33	8
9	Plain	3.17	0.12	16.68	38.1	33	8

$$\frac{1}{U_{o,w}} = \frac{b'_r A_o}{h_i A_{p,i}} + \frac{b'_p x_p A_o}{k_p A_{p,m}} + \frac{1}{h_{o,w}\left(\frac{A_{p,o}}{b'_{w,p} A_o} + \frac{A_f \eta_{f,wet}}{b'_{w,m} A_o}\right)} \tag{1}$$

where

$$h_{o,w} = \frac{1}{\frac{C_{p,a}}{b'_{w,m} h_{c,o}}} \tag{2}$$

The tube-side heat transfer coefficient, h_i, is evaluated from the Gnielinski correlation. The four quantities ($b'_{w,m}$, $b'_{w,p}$, b'_p, and b'_r) in Eq. (1) involving enthalpy–temperature ratios must be evaluated in

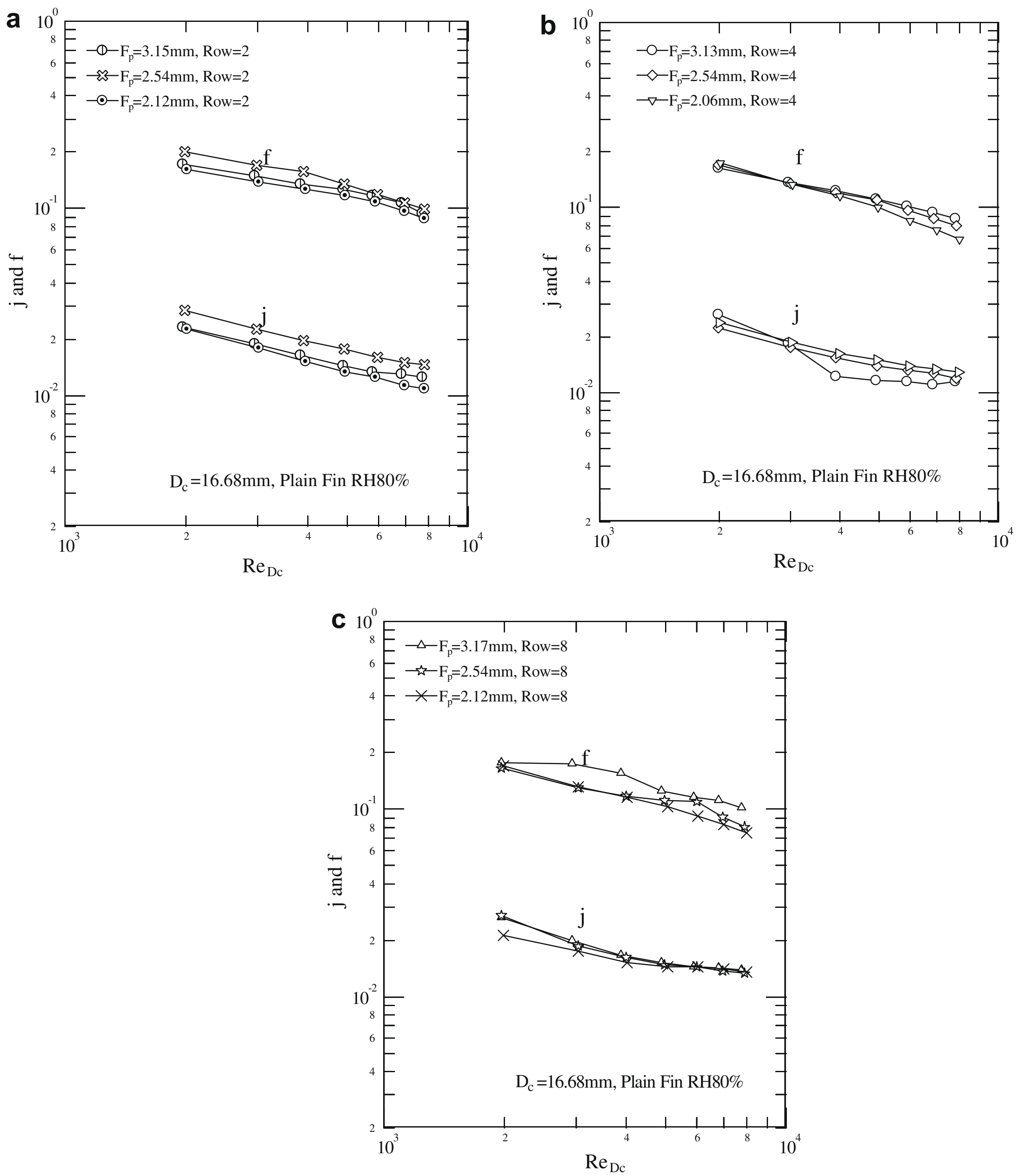

Fig. 2. Effect of fin pitch on heat transfer and friction characteristics (a) $N = 2$; (b) $N = 4$ and (c) $N = 8$ (RH = 80%).

advance. A detailed evaluation of these four terms can be found from Wang et al. [7]. The heat transfer performance is in terms of the Colburn j factor, i.e,

$$j = \frac{h_{c,o}}{G_c C_{p,a}} \mathrm{Pr}^{2/3} \tag{3}$$

The reduction of the friction factor of the heat exchanger is evaluated from the pressure drop equation proposed by Kays and London [16] as

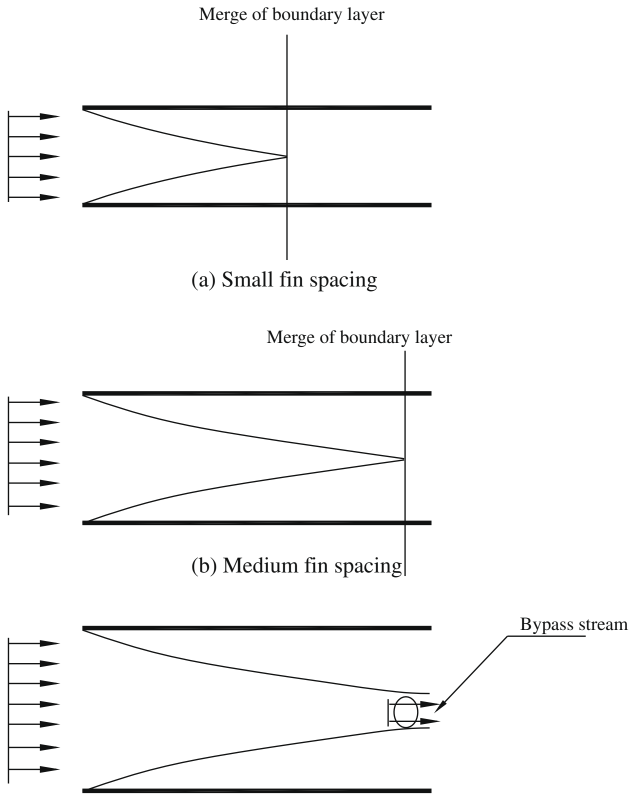

Fig. 3. Schematic of flow development alongside the fin channel: (a) small fin spacing, (b) medium fin spacing and (c) large fin spacing.

$$f = \frac{A_c}{A_o}\frac{\rho_i}{\rho_m}\left[\frac{2\rho_i \Delta P}{G_c^2} - (1+\sigma^2)\left(\frac{\rho_i}{\rho_o} - 1\right)\right] \tag{4}$$

Related explanation and calculation of the terminology can be seen from Wang et al. [7].

4. Results and discussion

A typical result concerning the effect of fin pitch on the airside performance for RH = 80% is schematically shown in Fig. 2. The corresponding tube rows are 2, 4, and 8, respectively. As expected, the friction factors and the sensible j factors decrease with increase of the Reynolds number. The effect of fin pitch on the sensible j factor is, in general, diminished with the rise of tube row. This is because more tube rows provide significantly mixing, thereby leading to a hardly detectable difference of j factor as the row number is increased to 8. However, there is a unique feature of the j factor for $N = 2$. The j factor shows appreciable increase when the fin pitch is increased from 2.12 mm to 2.54 mm, and a further rise to $F_p = 3.15$ mm yields a detectable drop of heat transfer performance. In fact it falls back to that of $F_p = 2.12$ mm. The special phenomenon is actually related to the developing characteristics of thermal and flow field. For further illustration of this phenomenon, one can examine the corresponding reciprocal of the inverse Graetz number x^+, which is defined as

$$x^+ = \frac{L/D_h}{\mathrm{Re}_{D_h}\mathrm{Pr}} \tag{5}$$

where L is the streamwise duct length and Pr is the Prandtl number. The flow may be considered to be fully developed when $x^+ > 0.1$ [17]. In general, the heat transfer performance within the heat exchanger is quite complex for it related to the interactions amid tubes and fins. For a shallow row number like $N = 2$, the effect of tube row is comparatively small, hence one can check the associated influence of development of flow field within channels. A close examination of the present test samples of $N = 2$ using Eq. (5) indicates that the contribution of development and fully developed region are quite corresponding. In this regard, one can realize the whole picture about the heat transfer performance subject to change of fin spacing as schematically shown in Fig. 3. For a smaller fin pitch as shown in Fig. 3(a), the flow develops along the

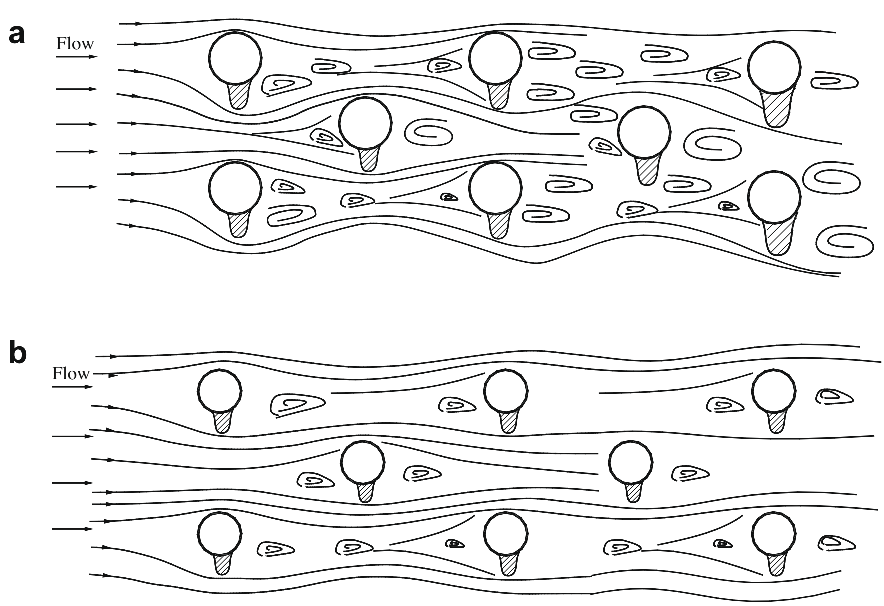

Fig. 4. Schematic of secondary flow subject to vapor shear (a) large fin spacing and (b) small fin spacing.

channel and merge accordingly somewhere alongside the channel, resulting in a comparatively low heat transfer performance. A further increase of fin spacing as seen in Fig. 3(b) will delay the conglomeration of boundary layer and increase the development length which gives rise to an increase of heat transfer performance. In the meantime, there will be no merge of boundary layer with a further increase of fin spacing as seen in Fig. 3(c). The nascent sign would suggest that the heat transfer performance will continue to rise since there is no boundary layer conglomeration. However, as clearly seen in Fig. 3(c), a bypass flow stream at the center region will offset the heat transfer gain from the development. In this sense, the heat transfer performance reveals a fallback when the fin spacing is sufficient large. However, it must be emphasized that this is applicable for shallow tube row where mixing caused by tubes is not so intensive.

In the meantime, the corresponding influence of fin pitch on the friction factor shows a slight scattering despite the variation is not so prominent. However, one can still see a marginal increase of friction factor for $N = 8$. The results are not in line with those in dry condition. For heat exchangers under completely dry operation, Rich [18] concluded that the friction factors were essentially independent of the number of tube row. The recent experimental data having larger diameter tube by Liu et al. [19] also support this finding. It is likely that the slight rise of friction factor with the fin pitch is associated with the condensate drainage. For a better understanding about the influence of condensate, Fig. 4(a) presents a cartoon to demonstrate how the friction factor may be slightly increased at a larger row number and a wider fin pitch. With a larger fin pitch, the effective vapor shear inside the fin spacing is comparatively small, the condensate is therefore easier to accumulate and

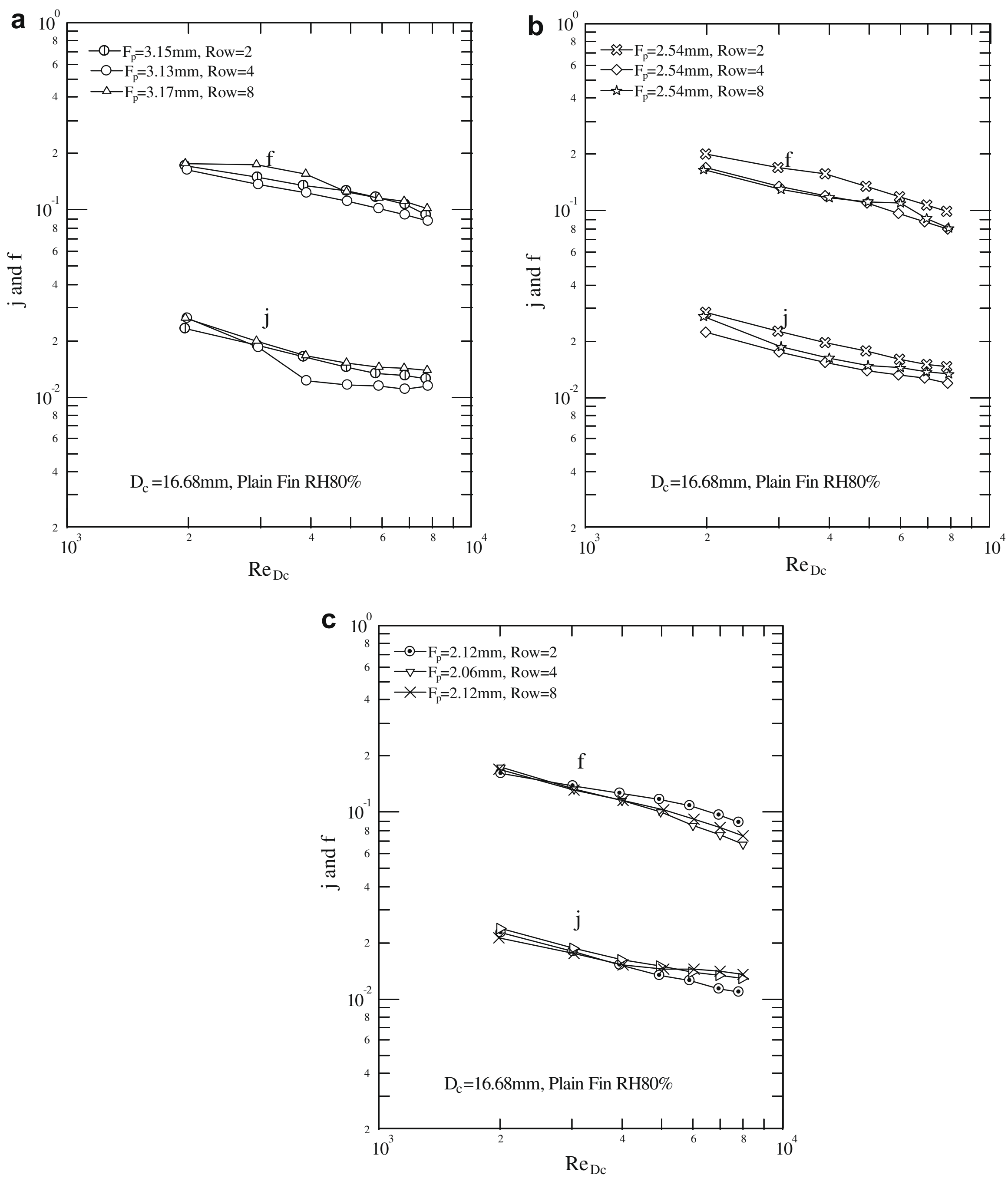

Fig. 5. Effect of the number of tube row on the heat transfer and friction characteristics: (a) $F_p = 2.1$ mm, (b) $F_p = 2.54$ mm and (c) $F_p = 3.15$ mm.

hang upon the surface. As a result, it is prone to having a higher frictional characteristic, yet the phenomenon may become more pronounced as the number of tube row is increased. On the other hand, the effect of vapor shear is reinforced at a smaller fin pitch and the condensate is easier to be removed from the surface. As a consequence, lower frictional performance is shown in Fig. 4(b) as the effect of secondary flow is reduced. Notice that condensate drainage within fin-and-tube heat exchanger is a very complex phenomenon for it interacts with both fin and tube surfaces.

Results regarding to the influence of the number of tube row on the airside performance are shown in Fig. 5. In general, the influence of tube row becomes less conceived when the fin pitch is reduced to 2.1 mm. As can be seen from Fig. 5(a), for a Reynolds number less than 4000, there is hardly any effect of the number of tube row on both heat transfer and frictional performance. By contrast, the sensible heat transfer j factors decrease with the rise of tube row when the Reynolds number is increased further. This is associated with the condensate blow off. The condensate is easier to adhere to the fin surface when the Reynolds number is low, resulting in a less influence of tube row. In the meantime, the adhered condensate may be blown off the fin surfaces when vapor shear is increased. Conversely, this phenomenon is not so significantly seen when the fin spacing is increased. This is because large condensate is prone to suspending between fins whereas smaller condensate just rolls alongside the fin, leading to this inconsistency.

5. Conclusion

This study presents the airside performance of the fin-and-tube heat exchangers having plain fin geometry with a larger diameter tube (D_c = 15.88 mm) under dehumidifying conditions. A total of nine samples of heat exchangers subject to change of the number of tube row and fin pitch are made and tested. Tests are conducted in a wind tunnel at controlled environment. Major conclusions of this study are summarized as follows:

(1) The effect of fin pitch on the sensible j factor is, in general, diminished with the rise of tube row. However, there is a distinct feature of the heat transfer performance occurring at a shallow row number (N = 2). The heat transfer performance is first increased when the fin pitch is increased from 2.12 mm to 2.54 mm, followed by a conceivable falloff if the fin pitch is increased to 3.15 mm. This unique characteristic is associated with the interaction between flow field development and bypass flow.
(2) The effect of fin pitch on the friction factor is somehow slightly scattering. There is a slight increase in friction factor for a tube row of eight. This is especially observable when the fin pitch is large. It is found that this phenomenon is related to condensate retention.
(3) The influence of tube row on the airside performance is rather small for both heat transfer and frictional characteristics. However, there is a slight deviation of this effect when fin pitch is increased to 2.54 mm or 3.1 mm. This is due to the condensate blown off phenomenon.

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References

[1] F.C. McQuiston, Heat mass and momentum transfer data for five plate-fin tube transfer surface, ASHRAE Trans. Part 1 84 (1978) 266–293.
[2] F.C. McQuiston, Correlation of heat, mass and momentum transport coefficients for plate-fin-tube heat transfer surfaces with staggered tubes, ASHRAE Trans. Part 1 84 (1978) 294–309.
[3] D.T. Beecher, T.J. Fagan, Effects of fin pattern on the air-side heat transfer coefficient in plate finned-tube heat exchanger, ASHRAE Trans. 93 (2) (1987) 1961–1984.
[4] W.M. Yan, P.J. Sheen, Heat transfer and friction characteristics of fin-and-tube heat exchangers, Int. J. Heat Mass Transfer 43 (2000) 1651–1659.
[5] D.R. Mirth, S. Ramadhyani, Prediction of cooling-coils performance under condensing conditions, Int. J. Heat Fluid Flow 14 (4) (1993) 391–400.
[6] D.R. Mirth, S. Ramadhyani, Correlations for predicting the air-side Nusselt numbers and friction factors in chilled-water cooling coils, Exp. Heat Transfer 7 (1994) 143–162.
[7] C.C. Wang, Y.C. Hsieh, Y.T. Lin, Performance of plate finned tube heat exchangers under dehumidifying conditions, J. Heat Transfer 119 (1997) 109–117.
[8] W. Pirompugd, S. Wongwises, C.C. Wang, A tube-by-tube reduction method for simultaneous heat and mass transfer characteristics for plain fin-and-tube heat exchangers in dehumidifying conditions, Heat Mass Transfer 41 (8) (2005) 756–765.
[9] W. Pirompugd, C.C. Wang, S. Wongwises, A fully wet and fully dry tiny circular fin method for heat and mass transfer characteristics for plain fin-and-tube heat exchangers under dehumidifying conditions, J. Heat Transfer 129 (9) (2007) 1256–1267.
[10] ASHRAE Standard 41.2-1987, Standard methods for laboratory air-flow measurement, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta GA, 1987.
[11] ASHRAE Standard 41.1-1986, Standard method for temperature measurement, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 1986.
[12] ASHRAE Standard 33-78, Method of testing forced circulation air cooling and air heating coils, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, GA, 1978.
[13] R.J. Moffat, Describing the uncertainties in experimental results, Exp. Thermal Fluid Sci. 1 (1987) 3–17.
[14] J.L. Threlkeld, Thermal Environmental Engineering, Prentice-Hall, Inc., New-York, 1970.
[15] R.J. Myers, The Effect of Dehumidification on the Air-Side Heat Transfer Coefficient for a Finned-Tube Coil, M.S. Thesis, University of Minnesota, Minneapolis, 1967.
[16] W.M. Kays, A.L. London, Compact Heat Exchanger, third ed., McGraw-Hill, New York, 1984. A.L.
[17] J.E. Sergent, A. Krum, Thermal Management Handbook for Electronic Assemblies, McGraw-Hill, New York, 1998.
[18] D.G. Rich, The effect of the number of tube rows on heat transfer performance of smooth plate fin-and-tube heat exchanger, ASHRAE Trans. 81 (1) (1975) 307–317.
[19] Y.C. Liu, R. Hu, R.B.C. Yang, I.Y. Chen, I.Y.C.C. Wang, Sensible airside performance of fin-and-tube heat exchangers – data with larger diameter tube, ASHRAE Trans. 114 (1) (2008) 379–386.