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Airside performance of herringbone wavy fin-and-tube heat exchangers – data with larger diameter tube

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ABSTRACT

This study examines the airside performance of the wavy fin-and-tube heat exchangers having a larger diameter tube (D_c = 16.59 mm) with the tube row ranging from 1 to 16. It is found that the effect of tube row on the heat transfer performance is quite significant, and the heat transfer performance deteriorates with the rise of tube row. The performance drop is especially pronounced at the low Reynolds number region. Actually more than 85% drop of heat transfer performance is seen for $F_p \sim 1.7$ mm as the row number is increased from 1 to 16. Upon the influence of tube row on the frictional performance, an unexpected row dependence of the friction factor is encountered. The effect of fin pitch on the airside performance is comparatively small for $N = 1$ or $N = 2$. However, a notable drop of heat transfer performance is seen when the number of tube row is increased, and normally higher heat transfer and frictional performance is associated with that of the larger fin pitch.

1. Introduction

Finned tube heat exchangers are very extensively used in various industrial applications. They are quite compact, light weight, and characterized by a relatively low cost fabrication. Normally the dominant resistance is on the airside; therefore the exploitation of enhanced fin surfaces is very common to effectively improve the overall heat transfer performance. Among the enhanced fin patterns, the herringbone wavy fin surface shown in Fig. 1 is one of the most popular surfaces since it can lengthen the airflow inside the heat exchanger and cause better mixing of the airflow.

The first comprehensive study related to the herringbone wavy fin pattern was done by Beecher and Fagan [1]. They presented test results for twenty-one herringbone fin-and-tube heat exchangers having $N = 3$. Data were presented in terms of Nusselt number, Nu_a, based on the arithmetic mean temperature difference (AMTD) vs. Graetz number. However, the wavy fin geometry tested by Beecher and Fagan [1] was rather uncommon when compared to practical design. Their fins were electrically heated, and thermocouples were embedded in the plates to determine the plate surface temperature. The power to the several electric heaters was adjusted to maintain a constant temperature over the airflow length. This simulated a fin-and-tube heat exchanger having 100% fin efficiency and zero contact resistance between the tube and fin. Webb [2] recasts the investigation of Beecher and Fagan [1], and developed the correlation.

A series investigation of the herringbone wavy fin patterns based on commercially available samples was conducted by Wang et al. [3–7], Chokeman and Wongwises [8], Wongwises and Chokeman [9], Kim et al. [10]. Effects of fin spacing, the number of tube row, wave height, fin pattern, fin thickness, and edge corrugation were systematically examined. Wang et al. [11] developed a general correlation based on their database. Their proposed correlation had a mean deviation of 6.98% for the heat transfer performance and a mean deviation of 8.82% for the friction factors. However, the foregoing data and correlations were mainly based on P_t = 25.4 mm and P_l = 19.05 mm with a tube size around 10 mm. Extrapolations of this correlation to other ranges of P_t and P_l are not recommended. Note that this kind of configuration is often used in small air-conditioning system.

For commercial applications, large diameter tube about 16 mm is very popularly used in ventilator and fan-coil units. Unfortunately, very rare data were available for this kind of configuration. In this connection, it is the objective of this study to provide airside performance of this popular fin geometry and discuss the effect of fin pitch and tube row on the relevant performance.

2. Experimental apparatus and data reduction

As tabulated in Table 1, the sample coils are of wavy fin configuration (δ_f = 0.12 mm, D_c = 16.59 mm, X_f = 8.25 mm, P_d = 2.2 mm,

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Nomenclature

A_c	minimum free-flow area, (m^2)
A_o	total surface area, (m^2)
D_c	fin collar outside diameter, $D_o + 2\delta_f$, (m)
D_o	outer tube diameter, (m)
f	friction factor, dimensionless
F_p	fin pitch, (m)
G_c	mass flux of the air based on the minimum flow area, (kg m^{-2} s^{-1})
h_o	airside heat transfer coefficient, (W m^{-2} K^{-1})
H	fin spacing, (m)
j	$Nu/RePr^{1/3}$, the Colburn factor, dimensionless
K_c	abrupt contraction pressure-loss coefficient
K_e	abrupt expansion pressure-loss coefficient
L	depth of the heat exchanger, (m)
N	number of longitudinal tube rows, dimensionless
P_l	longitudinal tube pitch, (m)
P_d	wave height, (m)
P_t	transverse tube pitch, (m)
Pr	Prandtl number
Re_{Dc}	Reynolds number based on tube collar diameter, dimensionless
V_{fr}	frontal velocity, (m s^{-1})
V_{max}	maximum velocity, (m s^{-1})
X_f	projected fin length, (m)
θ	corrugation angle, degree
δ_f	fin thickness, (m)
ρ_m	mean mass density of fluid, (kg m^{-3})
ρ_1	inlet mass density of fluid, (kg m^{-3})
ρ_2	outlet mass density of fluid, (kg m^{-3})
σ	contraction ratio of cross-sectional area
μ	dynamic viscosity of fluid, (N s m^{-2})
ΔP	pressure drop, (Pa)

P_t = 38.1 mm, P_l = 33 mm) with fin pitch ranging from 1.64 to 3.75 mm and the number of tube row from 1 to 16 as shown in Table 1. Definitions of relate geometric parameters can be seen from Fig. 1. Detailed construction of the circuitry arrangement is identical to those by Seshimo and Fujii [12] and Wang and Chi [13], and is depicted in Fig. 2. The present test was conducted in an open wind tunnel, relevant details can be found from previous works [5,13]. Tests are performed for the test samples with frontal velocity ranging from 1 to 5 m s^{-1}.

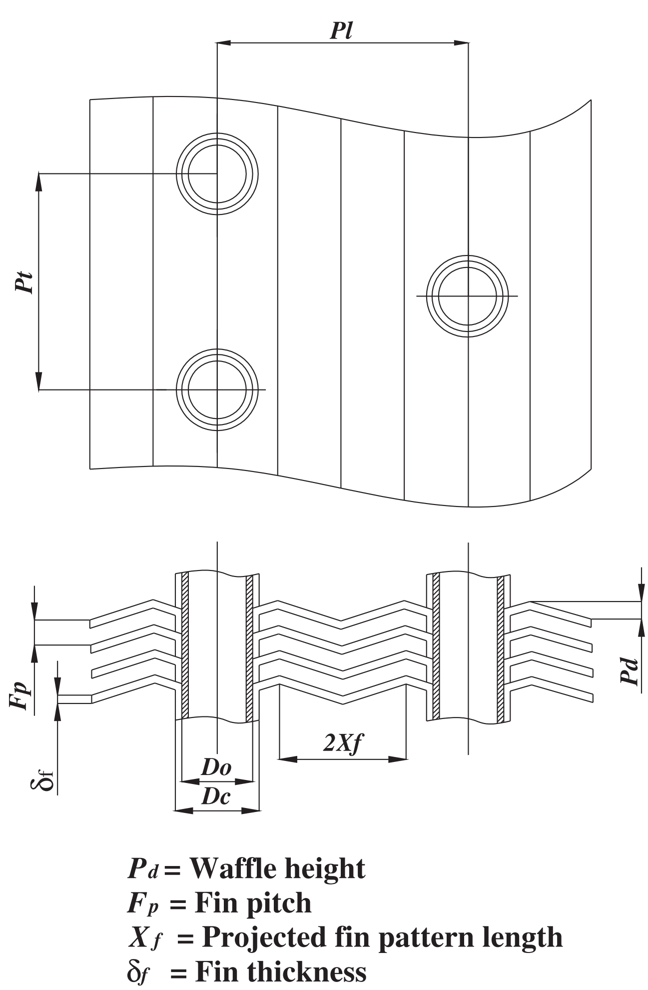

Fig. 1. Schematic of geometric parameters.

To obtain the heat transfer and pressure loss characteristics of the test coil from the experimental data, the ε-*NTU* method is applied to determine the *UA* product in the analysis. Detailed reduction can be seen from previous work (e.g. [13]). The airside heat transfer characteristics are presented in terms of the Colburn j factor:

$$j = \frac{h_o}{\rho V_{\max} Cp_a} Pr^{2/3}, \tag{1}$$

where $V_{max} = V_{fr}/\sigma$. The term, σ, is the ratio of the minimum flow area to frontal area. All the fluid properties are evaluated at the average values of the inlet and outlet temperatures under the steady state condition. The friction factors are calculated from the pressure drop equation proposed by Kays and London [14]. The relation for the nondimensional friction factor, f, in terms of pressure drop is shown below:

$$f = \frac{A_c}{A_o}\frac{\rho_m}{\rho_1}\left[\frac{2\Delta P \rho_1}{G_c^2} - (1+\sigma^2)\left(\frac{\rho_1}{\rho_2} - 1\right)\right], \tag{2}$$

where A_o and A_c stand for the total surface area and the flow cross-sectional area, respectively. Uncertainties in the reported experimental values of the Colburn j factor and friction factor f were estimated by the method suggested by Moffat [15]. The uncertainties ranged from 2.7% to 16.2% for the j factor, and 2.8% to 21.3% for f. The highest uncertainties were associated with lowest Reynolds number.

Table 1
Detailed geometric parameters of the wavy fin samples.

No.	F_p (mm)	N, Row	Tubes	Width (mm)	Height (mm)	Depth (mm)
1	3.19	1	10	600	381	33
2	1.82	1	10	600	381	33
3	3.55	2	10	600	381	66
4	1.76	2	10	600	381	66
5	3.37	4	10	600	381	132
6	1.91	4	10	600	381	132
7	3.75	8	10	600	381	264
8	1.73	8	10	600	381	264
9	3.61	12	10	600	381	396
10	1.79	12	10	600	381	396
11	3.57	16	10	600	381	528
12	1.64	16	10	600	381	528

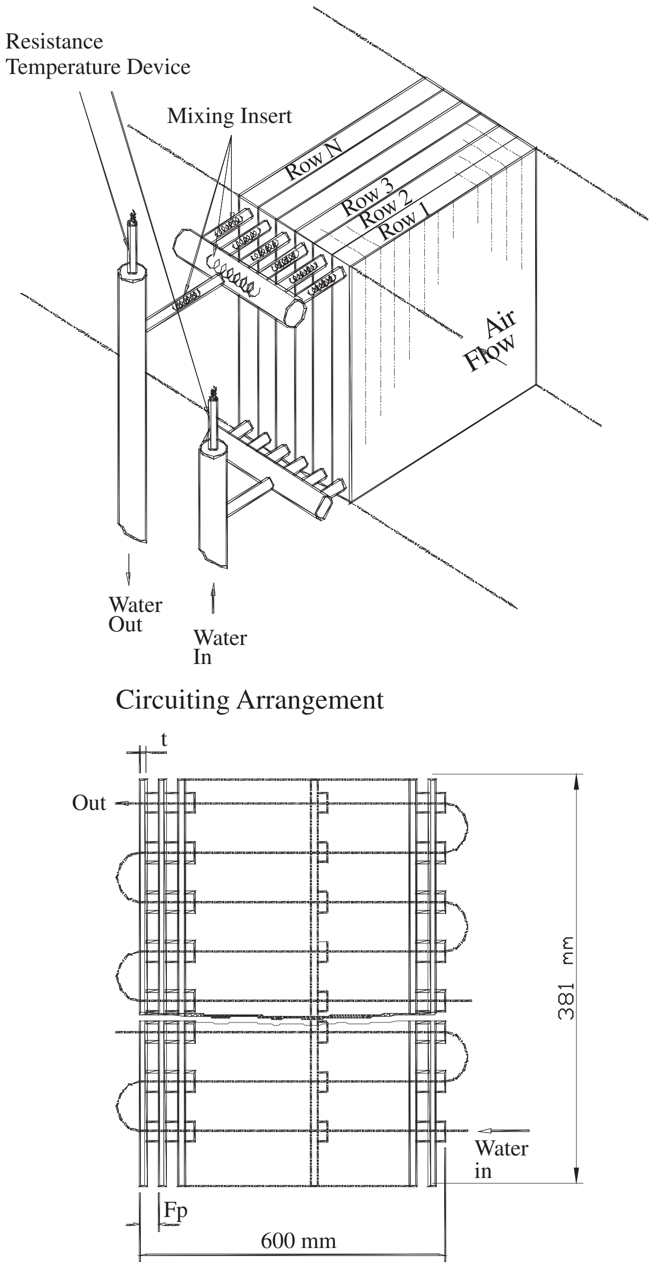

Fig. 2. Schematic of the construction and circuitry of test samples.

3. Results and discussion

The effect of fin pitch on the airside performance for different number of tube row is seen in Fig. 3. The corresponding number of tube row is 1, 2, 4, 8, 12, and 16, respectively. The fin pitches shown in Fig. 3(a) is roughly around 1.7 mm whereas it is approximately 3.3 mm in Fig. 3(b). For a shallow tube row like $N = 1$ or $N = 2$, the relative independence of heat transfer performance with tube row is seen. This phenomenon may be associated with development of boundary layer along the fin surface. On the other hand, a further rise of tube row casts a tremendous impact on heat transfer, yet the heat transfer performance suffered significantly with the rise of tube row. Notice that the heat transfer performance drop is especially pronounced at the low Reynolds number region. In fact, a loss of more than 85% heat transfer performance is observed for $F_p \sim 1.7$ mm as the tube row is increased from 1 to 16 at a Reynolds number of 2000. Meanwhile, the deterioration of heat transfer performance is reduced when the Reynolds number is increased. Analogous trend is observed for $F_p \sim 3.3$ mm but with a smaller deterioration of heat transfer performance against tube row. The results are in line with the numerical prediction of corrugated channel performed by Yang et al. [16]. Using a Lam–Bremhorst low Reynolds number turbulence model, Yang et al. [16] investigated the transitional Reynolds number subject to the influence of corrugation angles. They found that the transition position from laminar to turbulent had been moved from downstream cycle to upstream cycle when the fin spacing is increased. In fact, the transition Reynolds number is reduced considerably with the rise of H/L. Where H is the corrugated fin spacing and L is the axial length of a cycle. As a consequence, one can see that the heat transfer performance for larger fin spacing outperforms that of smaller fin spacing when the number of tube row is sufficiently large.

The effect of the number of tube row on the frictional performance is also shown in Fig. 3. It is interesting to note that the friction factors, although not so pronounced as that of heat transfer performance, depend on the number of tube row to some extent. This is quite unusual because the plain fin-and-tube heat exchanger (Wang and Chi [13] and recent larger diameter plain fin data Liu et al. [17]) as well as some highly interrupted fin-and-tube heat exchangers like louver, slit, and convex louver [5,18–19] did not reveal this kind of behaviour. In fact, normally the published data showed that the friction factor is independent of the number of tube row provided $N \geqslant 2$. The tube row dependence for friction factor occurs only for the present wavy fin configuration. In the open literatures, test data of the herringbone fin surface showing a dependence of the axial length or the number of tube row had been reported by the Mirth and Ramadhyani [20] and Wang et al. [21]. However, Mirth and Ramadhyani [20] did not provide any explanation while Wang et al. [21] suspects the condensate may alter the flow field that leads to this consequence. Lin et al. [22] carried out a flow visualization for condensate flow pattern, and they concluded from their observation that the friction dependence phenomenon is related to the non-uniform condensation caused by the wavy corrugation. However, it should be mentioned that the studies [20–22] were operated under dehumidifying conditions where condensate was normally seen provided the surface is below the dew point. However, the test results for the present wavy fin-and-tube heat exchangers were operated at fully dry condition where no condensate takes place. In this sense, there must be other explanation about this unique phenomenon.

The previous studies for wavy fin surface tested in completely dry condition by Wang et al. [3–7] revealed only very minor influence of tube row on the friction factor. The results by Wang et al. [3–7] are no surprise for their test samples were mainly based on $P_l = 19.05$ mm with N ranging from 1 to 6 whereas the present P_l is 33 mm having a maximum tube row of 16. The effective axial length of this study is much longer than the previous one. The noticeable drop of heat transfer performance with the rise of tube row (especially for $N > 4$) for $F_p \sim 1.7$ mm is mainly due to fully developed characteristics. The effect of fin pitch on the airside performance for different number of tube row is seen in Fig. 3. As seen, a notable drop of heat transfer performance is seen when the number of tube row is increased over 4. The performance drop is especially pronounced at the low Reynolds number region. The relative independence of heat transfer performance for $N = 2$ may be associated with development of boundary layer along the fin surface. As pointed out by Kim and Kim [23] who examined the influence of wider pitch on the fin-and-tube heat exchangers having plain fin geometry. Their data also suggests a tiny increase of j-factor with a rise of fin pitch. However, they also mentioned that the dependence of j-factor with fin pitch is reduced when boundary layer interaction could not occur. The results are in line with the

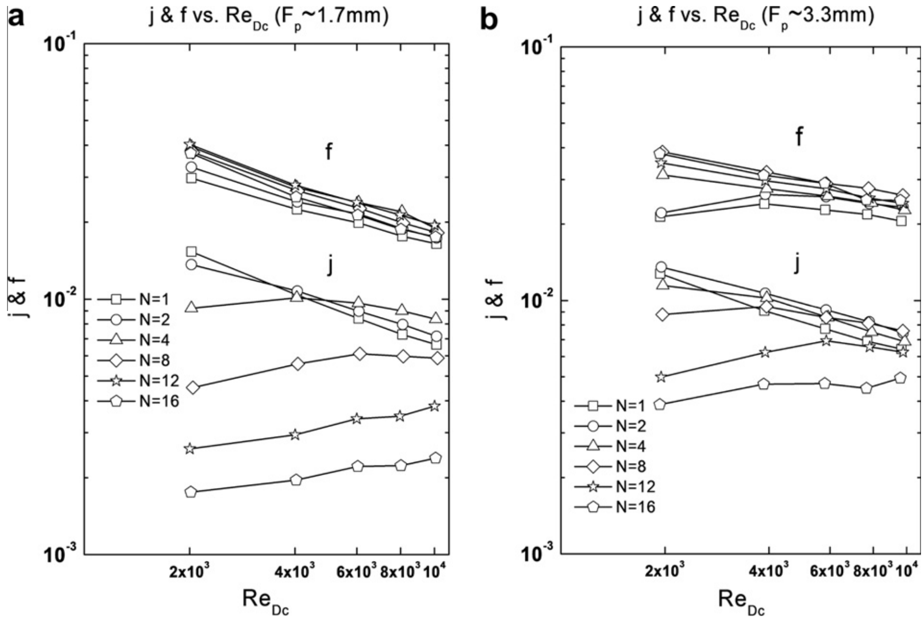

Fig. 3. Effect of the number of tube row on the airside performance for (a) $F_p \sim 1.7$ mm, (b) $F_p \sim 3.3$ mm.

present test results. For $N = 1–4$, the interactions of boundary layer is relatively small, giving rise to a negligible effect of fin pitch on the *j*-factor.

The row dependence for friction factor may originate from the longer axial length and the corrugation from which a significant change of flow pattern occurs. According to the flow visualization of corrugated channels by Ali and Ramadhyani [24], some noticeable unsteadiness/instability is manifested by wavering of the dye streak occurring at some corrugation, indicating part of the corrugation channel is steady (from entrance to the location where unsteadiness occurs) while the flow field of the rest of the channel is unstable. The point of starting wavering depends on the corrugation angle, *H*/*L*, and the Reynolds number. Apparently, the frictional performances are drastically different in these two regions (steady flow field and unsteady flow field) yet the effective total pressure drop is summed from these two regions. Initially, the pressure drop is mainly engendered from the steady contribution when the number of tube row is shallow, and the contribution of the unsteady portion is gradually increased with a further rise of tube row. Hence, the combination varies with different contribution of these two regimes, leading to a dependence of number of tube row on the frictional performance. Note that it is not necessary that the unsteady contribution surpasses steady contribution. This is because addition entrance contribution exists in the steady portion, thereby one can see the friction factor is first increased with the number of tube row and it peaks at some certain tube row; then decreases thereafter. In this sense, one can see that the dependence of friction factor with the tube row is more pronounced for shallow tube row. This is because the flow field for very deep row coil is well mixed and the difference caused by steadiness and unsteadiness becomes less profound.

Converse to the present wavy fin geometry, the highly interrupted surfaces like louver or slit had tremendously mingled with the air flows between adjacent fin channels. Therefore the effect of tube row on the frictional performance is inconceivable. The effect of fin pitch on the airside performance for the present wavy fin geometry is depicted in Fig. 4. The associated tube rows are 2, 4, 8, and 16, respectively. For a shallow tube row like 2 (Fig. 4(a)) or 4 (Fig. 4(b)), one can see a comparatively small influence of fin pitch either on heat transfer or on frictional performance. The results agree with the foregoing arguments about the steady/unsteady flow field caused by the corrugations and the axial length. With a rise of tube row, the unsteady contribution becomes more and more pronounced and the increase of fin spacing accentuates the unstable flow field. As a consequence, both heat transfer and frictional performance for the larger fin pitch is much higher than those of smaller fin pitch. The flow observation carried out by Ali and Ramadhani [24] had confirmed the present measurement. They found the formation of large, clearly defined, spanwise vortices in the transitional regime, and such vortices were not observed in the narrower channel. Moreover, the generated vortices may shed at the apex, leading to an appreciable heat transfer difference amid smaller and larger fin pitch.

4. Conclusions

This study presents the airside performance of the wavy fin-and-tube heat exchangers having a larger diameter tube ($D_c = 16.59$ mm). A total of twelve samples of heat exchangers subject to change of the number of tube row and fin pitch are made and tested. Tests are conducted in an open wind tunnel at controlled environment. Major conclusions of this study are summarized as follows:

(1) The effect of tube row on the heat transfer performance is quite significant. It is found that the heat transfer performance deteriorates with the rise of tube row. The performance drop is especially pronounced at the low Reynolds number region. In fact, more than 85% drop of heat transfer performance is seen for $F_p \sim 1.7$ mm as the row number is increased from 1 to 16.
(2) Upon the influence of tube row on the frictional performance, an unexpected row dependence of the friction factor is encountered.
(3) The effect of fin pitch on the airside performance is comparatively small for $N = 1$ or $N = 2$. However, a notable drop of heat transfer performance is seen when the number of tube row is increased, and normally higher heat transfer and frictional performance is associated with that of the larger fin pitch.

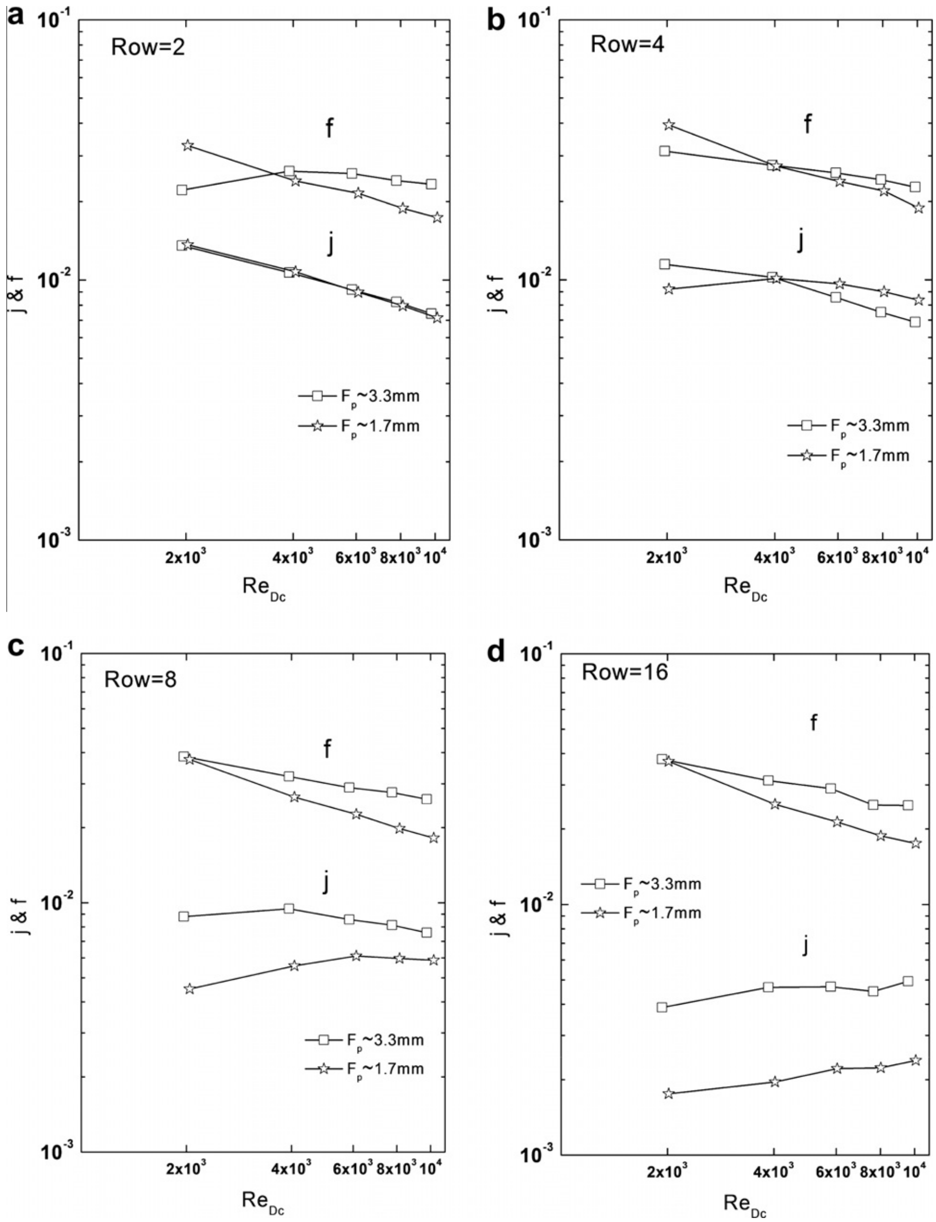

Fig. 4. Effect of the fin pitch on the airside performance for (a) $N = 2$, (b) $N = 4$, (c) $N = 8$, and (d) $N = 16$.

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