Performance of Plain Fin-and-tube Heat Exchangers-Data With Larger Diameter Tube Under Dehumidifying Conditions

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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). 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. It is found that te effect of fin pitch on the sensible j factor is, in general, diminished with the rise of tube row. 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. This is because the condensate adhered phenomena..

Keywords: Fin-and-tube heat exchanger, plain fin, dehumidification.

1. Introduction

Fin-and-tube heat exchangers are widely used in applications of air-conditioning and refrigeration systems. They can be applicable to condensers and evaporators. In evaporators, which typically use aluminum fins with the surface temperature generally being below the dew point temperature. As a result, simultaneous heat and mass transfer occurs along the fin surfaces.

Many studies have been published on the heat and mass transfer characteristics of fin-and-tube heat exchangers under dehumidifying conditions. For instance, McQuiston [1,2] presented experimental data for five plate fin-and-tube heat exchangers, and developed a well-known heat transfer and friction correlation for both dry and wet surfaces. Mirth and Ramadhyani [3,4] investigated the heat and mass characteristics of wavy fin heat exchangers. Their results showed that the Nusselt number was very sensitive to changes of the inlet dew point temperature, and that the Nusselt number decreased with an increase of dew point temperatures. Similar results were reported by Fu et al. [5] in dehumidifying heat exchangers having a louver fin configuration. They reported a pronounced decrease of the wet sensible heat transfer coefficients with the rise of the inlet relative humidity. Contrary to this, the experimental data of Seshimo et al. [6] indicated that the Nusselt number was relatively independent of the inlet conditions. Wang et al. [7] studied the effects of the fin pitch, the number of tube rows, and the inlet relative humidity on the heat transfer performance under dehumidification, they concluded that the sensible heat transfer performance is relatively independent of the inlet humidity. The differences in the existing literature are attributed to the different reduction methodologies. Pirompugd et al. [8,9] presented a new reduction method for the calculation of the heat and mass transfer characteristics for fin-and-tube heat exchangers under dehumidifying conditions. Their results showed that the heat and mass transfer characteristics were relatively independent of fin pitch and of relative humidity.

The foregoing studies are conducted for a nominal tube diameter of 9.52, 7.94 or 7 mm which are quite popular in typical small air-conditioning system. In typical applications like fan-coil or ventilator, use of larger diameter like 15.88 mm is also very common. Unfortunately, there is 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 and to examine the applicability of the existing correlation 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].

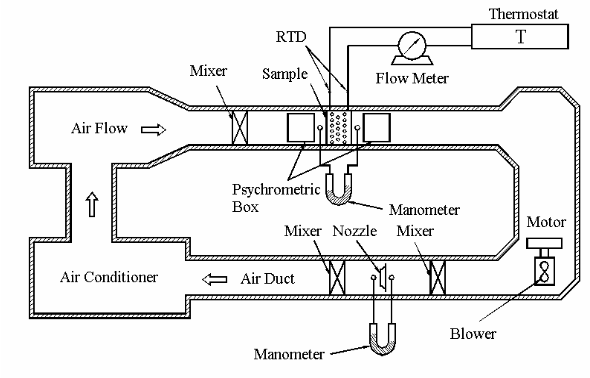

Figure 1. Schematic of the test apparatus

Table 1 Detailed geometric parameters of the test samples.

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

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 precalibrated RTDs. The water volumetric flow 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, $\left|\dot{Q}_r-\dot{Q}_a\right|/\dot{Q}_{avg}$, 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 ±6.5% for reduction of the sensible heat transfer coefficient whereas it is within ±8% 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;

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

where

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

heat transfer performance is in terms of the Colburn j factor, i.e,

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

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

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

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 the presence to more tube row gives rise to significantly mixing, thereby leading to a hardly discriminable difference of sensible j factor as the row number is increased to 8. In the meantime, the corresponding influence on friction factor shows slightly scattering. The variation is not so pronounced. However, one can still see a slight increase in friction factor for a tube row of eight. The results are not the same with those in dry condition. For heat exchangers under completely dry operation, Rich [17] concluded that the heat transfer coefficients were essentially independent of fin spacing for continuous plate fin geometry. The recent experimental data of Liu et al. [18] also support this finding. It is likely that the slight rise of friction factor with the fin pitch is associated with the condensate drainage situation. Notice that condensate drainage within fin-and-tube heat exchanger is a very complex phenomenon for it interacts with both fin and tube surfaces.

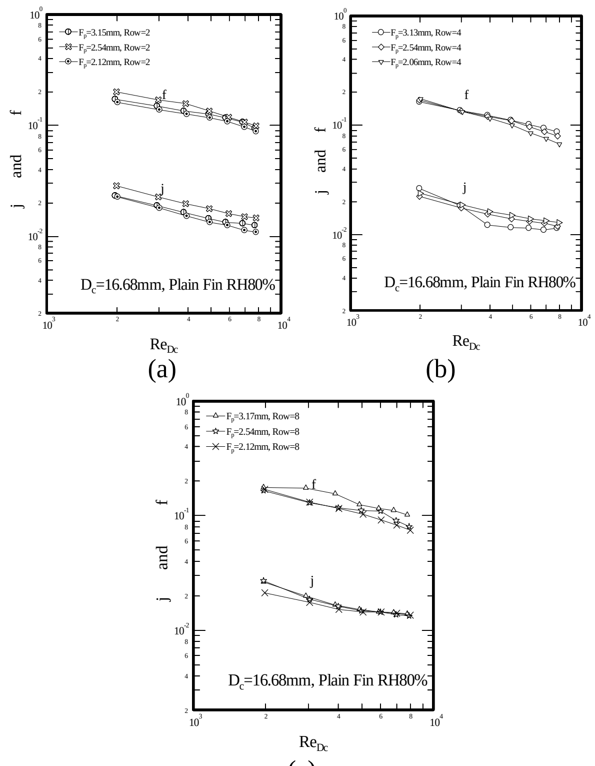

Figure 2. Effect of fin pitch on heat transfer and friction characteristics (a) N = 2; (b) N = 4; and (c) N =8.

Results related to the influence of tube row on the airside performance are shown in Fig. 3. 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. 3(a), for a Reynolds number being less 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. Apparently, this is associated with the condensate blow off. The condensate is easier to adhere between fin surface when the Reynolds number is low, resulting in less influence of tube row. In the meantime, the adhered condensate may be blowing 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 may suspend between fins whereas smaller condensate just rolls alongside the fin, leading to this inconsistency.

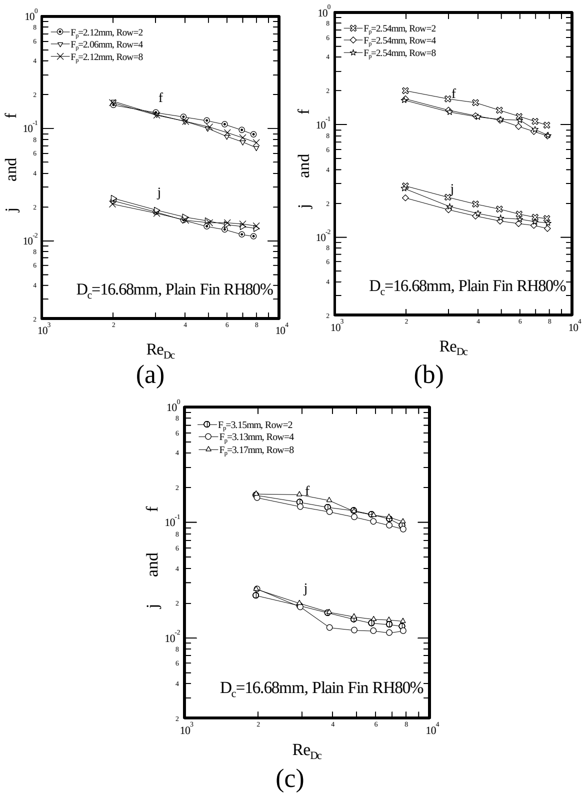

Figure 3 Effect of the number of tube row on the heat transfer and friction characteristics (a) F_p =2.1 mm, (b) F_p = 2.54 mm; and (c) F_p = 3.15 mm.

5. Conclusion

This study presents the airside performance of the fin-and-tube heat exchangers having plain fin geometry with a larger diameter tube (Dc = 15.88 mm). 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.

(2) 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 because the condensate adhered phenomena.

Acknowledgments

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