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# Air-side performance of herringbone wavy fin-and-tube heat exchangers under dehumidifying condition – Data with larger diameter tube

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ABSTRACT

This study examines the airside performance of the herringbone wavy fin-and-tube heat exchangers in dehumidifying condition having a larger diameter tube ($D_c$ = 16.59 mm) with the tube row ranging from 2 to 12. Test results are compared to that of dry conditions and plain fin geometry. Upon the influence of surface condition (dry or wet) on the heat transfer performance, the heat transfer performance in dehumidifying condition normally exceeds that in dry condition, and is more pronounced with the rise of tube row or reduction of fin pitch. By contrast, it is found that the heat transfer coefficient for plain fin geometry in dehumidifying condition is slightly lower than that in dry condition. The pressure drops in wet condition is much higher than that in dry condition. However, the difference in pressure drop amid dry and dehumidifying condition for wavy fin configuration is less profound as that of plain fin geometry.


## 1. Introduction

The fin-and-tube heat exchangers are mostly used in air conditioning, refrigeration, power production and many other thermal processing applications. The popularity of this kind of heat exchangers arises from low cost, highly operational reliability, and easier installation/maintenance. In practical operation the major resistance lies on the airside; hence exploitation of enhanced fin surfaces like slit or louver is very common especially for residential or small business applications. On the other hand, industrial or larger commercial applications often avoid using the foregoing highly interrupted surfaces due to the concerns of blockage after long term operation. From this perspective, continuous fin patterns having plain or wavy configurations are still the most favorable selections as far as high capacity applications are concerned. For deploying the fin-and-tube heat exchangers pertaining to high capacity exploitation, larger diameter tube, larger longitudinal tube pitch, and larger transverse tube pitch is often incorporated into this kind of applications and the specific geometry (nominal $D_o$ = 15.875 mm, $P_t$ = 38.1 mm, $P_l$ = 33.1 mm) is regarded as the most commonly used. Liu et al. [1,2] had recently reported air side performance for plain fin-and-tube heat exchanger associated with this geometry for both dry and wet conditions. For the wavy fin geometry, most of the reported data are applicable for smaller diameter tube and tube pitch (Beecher and Fagan [3], Yan and Sheen [4], Wang et al. [5–9], Chokeman and Wongwises [10] and Wongwises and Chokeman [11]). Notice that typical applications like fan-coil or ventilator, exploitation of larger diameter like 15.88 mm is also very common. One of the major differences amid large diameter tube and smaller diameter tube is that the portion of pressure drop by tube is much higher. And it also influences the condensate drainage when the tube size is increased, leading to a change of heat transfer coefficient. Unfortunately, there is only very limited performance data of the fin-and-tube heat exchanger with larger diameter tube wavy fin-and-tube heat exchange under fully dry condition (Wang et al. [12,13]) in the open literature and there is virtually no data available in dehumidifying conditions associated with large diameter tube configuration. Hence, the objective of the present study is to provide relevant performance data to the database, and discuss some unusual characteristics of the present test samples.

## 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 [14]. A differential pressure transducer is used to measure the

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**Nomenclature**

| | |
|---|---|
| $A_f$ | fin surface area, ($m^2$) |
| $A_{p,i}$ | inner tube wall surface, ($m^2$) |
| $A_{p,m}$ | mean tube wall surface, ($m^2$) |
| $A_{p,o}$ | outer tube wall surface, ($m^2$) |
| $A_o$ | total surface area, ($m^2$) |
| $b'_p$ | slope of a straight line between the outside and inside tube wall temperatures, ($J\ kg^{-1}\ K^{-1}$) |
| $b'_r$ | slope of the air saturation curved at the mean coolant temperature, ($J\ kg^{-1}\ K^{-1}$) |
| $b'_{w,m}$ | slope of the air saturation curve at the mean water film temperature of the external surface, ($J\ kg^{-1}\ K^{-1}$) |
| $b'_{w,p}$ | slope of the air saturation curve at the mean water film temperature of the primary surface, ($J\ kg^{-1}\ K^{-1}$) |
| $C_{p,a}$ | moist air specific heat at constant pressure, ($J\ kg^{-1}\ K^{-1}$) |
| $D_c$ | fin collar outside diameter, $D_o + 2\delta_f$, (m) |
| $D_o$ | outer tube diameter, (m) |
| $F_P$ | fin pitch, (m) |
| $h$ | sensible heat transfer coefficient, ($W\ m^{-2}\ K^{-1}$) |
| $h_i$ | tube side heat transfer coefficient, ($W\ m^{-2}\ K^{-1}$) |
| $h_{o,w}$ | total heat transfer coefficient for wet external fin, ($W\ m^{-2}\ K^{-1}$) |
| $H$ | fin spacing, (m) |
| $k_p$ | thermal conductivity of tube, ($W\ m^{-1}\ K^{-1}$) |
| $k_w$ | thermal conductivity of water, ($W\ m^{-1}\ K^{-1}$) |
| $L$ | 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) |
| $V_f$ | frontal velocity, ($m\ s^{-1}$) |
| $X_f$ | projected fin length, (m) |
| $x_p$ | tube wall thickness, (m) |
| $y_w$ | thickness of condensate water film, (m) |
| $\theta$ | corrugation angle, degree |
| $\eta_{f,wet}$ | wet fin efficiency |
| $\delta_f$ | fin thickness, (m) |
| $\Delta P$ | pressure drop, (Pa) |

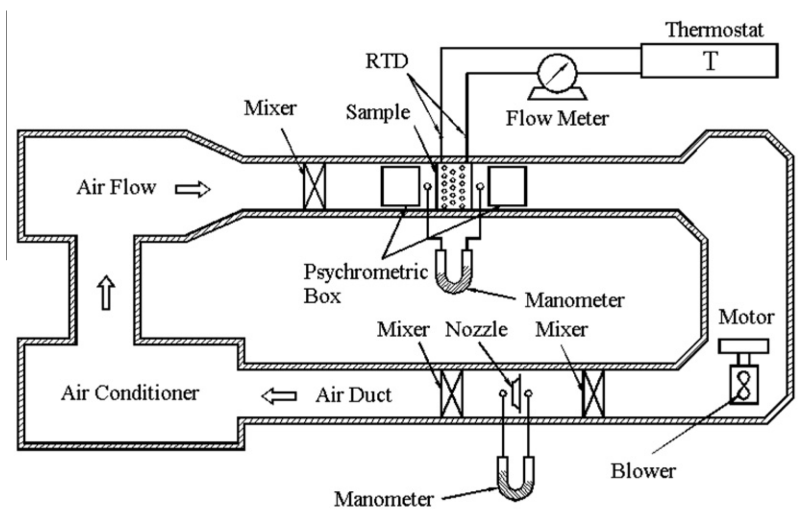



**Fig. 1.** Schematic of the test apparatus.

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 [15]. Further details of the experimental apparatus can be found from some previous studies [2,16]. Detailed geometry used for the present wavy fin-and-tube heat exchangers is tabulated in Table 1 with $P_t$ = 38.1 mm and $P_l$ = 33 mm. And schematic showing the present wavy fin-and-tube heat exchangers are shown in Fig. 2. The test conditions of the inlet-air are as follow:

Dry-bulb temperature of the air: 27 ± 0.1 °C.
Inlet relative humidity for the incoming air: 85%.
Inlet-air velocity: 1–5 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 [17]. The maximum uncertainty occurred at the smallest frontal velocity and is less than ±4.8% for the sensible heat transfer coefficient whereas it is within ±6.2% for the frictional reduction.

**Table 1**
Detailed geometric parameters of the wavy fin samples.

| No. | $F_p$ (mm) | N, Row | Tubes | Width (mm) | Height (mm) | Depth (mm) |
|---|---|---|---|---|---|---|
| 1 | 3.55 | 2 | 10 | 600 | 381 | 66 |
| 2 | 1.76 | 2 | 10 | 600 | 381 | 66 |
| 3 | 3.37 | 4 | 10 | 600 | 381 | 132 |
| 4 | 1.91 | 4 | 10 | 600 | 381 | 132 |
| 5 | 3.75 | 8 | 10 | 600 | 381 | 264 |
| 6 | 1.73 | 8 | 10 | 600 | 381 | 264 |
| 7 | 3.61 | 12 | 10 | 600 | 381 | 396 |
| 8 | 1.79 | 12 | 10 | 600 | 381 | 396 |

## 3. Data reduction

Basically, the present reduction method is analogous to Threlkeld's approach [18]. Detailed description of the reduction process

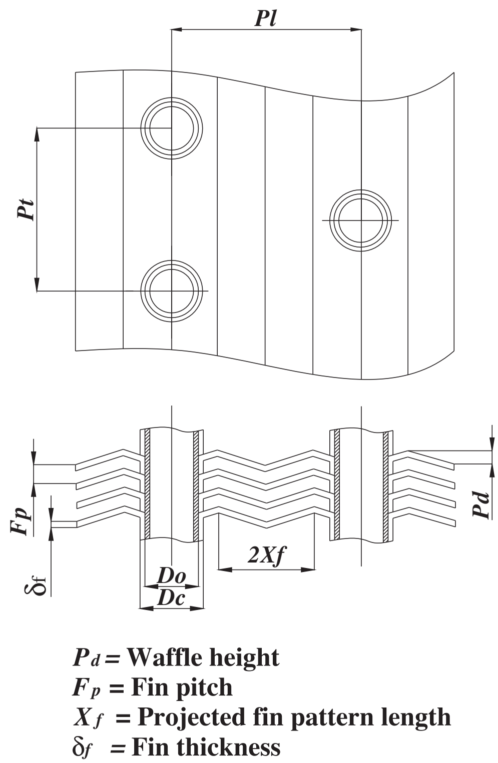



**Fig. 2.** Schematic of geometric parameters.

can be found from Wang et al. [16]. Notice that the Threlkeld method is an enthalpy-based reduction method. A very 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 [19]) as follows;

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

where

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

$y_w$ in Eq. (2) is the thickness of the water film. A constant of 0.005 inch was proposed by Myers [19]. In practice, $\frac{y_w}{k_w}$ accounts only 0.5–5% comparing to $\frac{C_{p,a}}{b'_{w,m} h}$ and is often neglected by previous investigators. Hence the sensible heat transfer coefficient under dehumidifying condition is given as

$$h = \frac{C_{p,a} h_{o,w}}{b'_{w,m}} \tag{3}$$

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 advance. A detailed evaluation of these four terms can be found from Wang et al. [16].

## 4. Results and discussion

Test results in terms of heat transfer coefficients and pressure drops for the test samples having $N$ = 2, 4, and 8, and 12 are shown in Fig. 3. For comparison purpose, test results for fully dry conditions from prior study [13] are also depicted in the figure. As expected, both heat transfer coefficients and pressure drops increase with the rise of frontal velocity. For dehumidifying condition with $N$ = 2, the heat transfer coefficients are relatively independent of fin pitch (Fig. 3(a)). With the rise of tube row, one can see a detectable rise of heat transfer coefficient in wet condition with a smaller fin pitch ($N$ = 4), but the tendency become reversed with a further rise of the number of tube row ($N$ = 8, 12). Notice that the heat transfer coefficient in fully dry condition is comparable with that in wet condition for $N$ = 2. However, the heat transfer coefficient in fully wet condition normally exceeds that of fully dry condition when $N > 2$, and the difference becomes more pronounced when the fin pitch is reduced. Note that the number of corrugation is proportional to the increase of the number of tube row, implicating an increased influence of corrugation when the tube row is increased. In fact, it is expected that considerable variations of heat transfer coefficient alongside the corrugation may occur, yet a low heat transfer region may appear at the valley and at the suction side of the wavy channel. This is applicable for a typical wavy fin channel in fully dry condition. The results were reported by a numerical visualization performed by McNab et al. [20]. They reported large regions airflow separation of recirculating flow across the apex when the bulk of the airflow passes across the center of the wavy channel as shown in Fig. 4(a). Analogous results were also reported by Hwang et al. [21] who performed an experimental and numerical study concerning the flow and heat transfer process of wavy duct. The effect of corrugation is intensified when the fin pitch is reduced. A schematic of the flow pattern showing both streamwise and spanwise direction is shown in Fig. 4(a) and (b). As seen, the streamwise flows are disturbed at the turning corner showing recirculation flow and separation/reattachment. The results suggest a low heat transfer region at the valley and at the suction side of the wavy channel. Therefore, test results of dry condition for deep row coil ($N$ = 8 and 12) shows substantial decrease of heat transfer performance when the fin pitch is reduced.

On the other hand, for a wavy channel at dehumidifying operating condition, water condensate appears as long as the surface temperature is below the corresponding dew point temperature. Lin et al. [22] conducted a visual observation of wavy fin geometry under dehumidifying conditions and a schematic of their observation is shown in Fig. 4(c). It appears that the condensate prevails on the valley and the suction side of the wavy fin geometry, indicating sufficient water vapor flow condenses nearby these regions. In this regard, the condensing water vapour may bring about the air flow toward the valley and suction side of the surface, thereby alleviating the influence of the recirculation/separation flow at these regions. As a consequence, better heat transfer performance under dehumidifying conditions emerges from the lifting flow circulation/separation as compared to that of dry condition. Consequently the deteriorating influence of fin pitch on the heat transfer coefficient for deep row coil ($N$ = 8 and 12) under wet condition is comparatively small related to that in dry condition.

For comparison purpose, test results for plain fin geometry from previous studies [1,2] with the same diameter tube, longitudinal tube pitch, and transverse tube pitch subject to dry and wet conditions is plotted in Fig. 5. It is interesting to know that the sensible heat transfer coefficient for the plain fin geometry under wet condition is either comparable or generally lower than that of dry condition. Test results for plain fin having a smaller diameter, longitudinal tube pitch, transverse pitch ($P_t$ = 25.4 mm, $P_l$ = 22 mm, $D_c$ = 10.3 mm, [16,23]) also revealed the same results as those of larger one. This is because, unlike those of wavy fin channel, the plain fin geometry does not manage to have pronounced recirculations alongside the fin channel. Therefore the presence of condensate does not contribute to eliminate this phenomenon. On the other hand, the presence of condensate may form additional barrier to heat transfer; thereby deteriorating the performance of plain fin surface in wet condition.

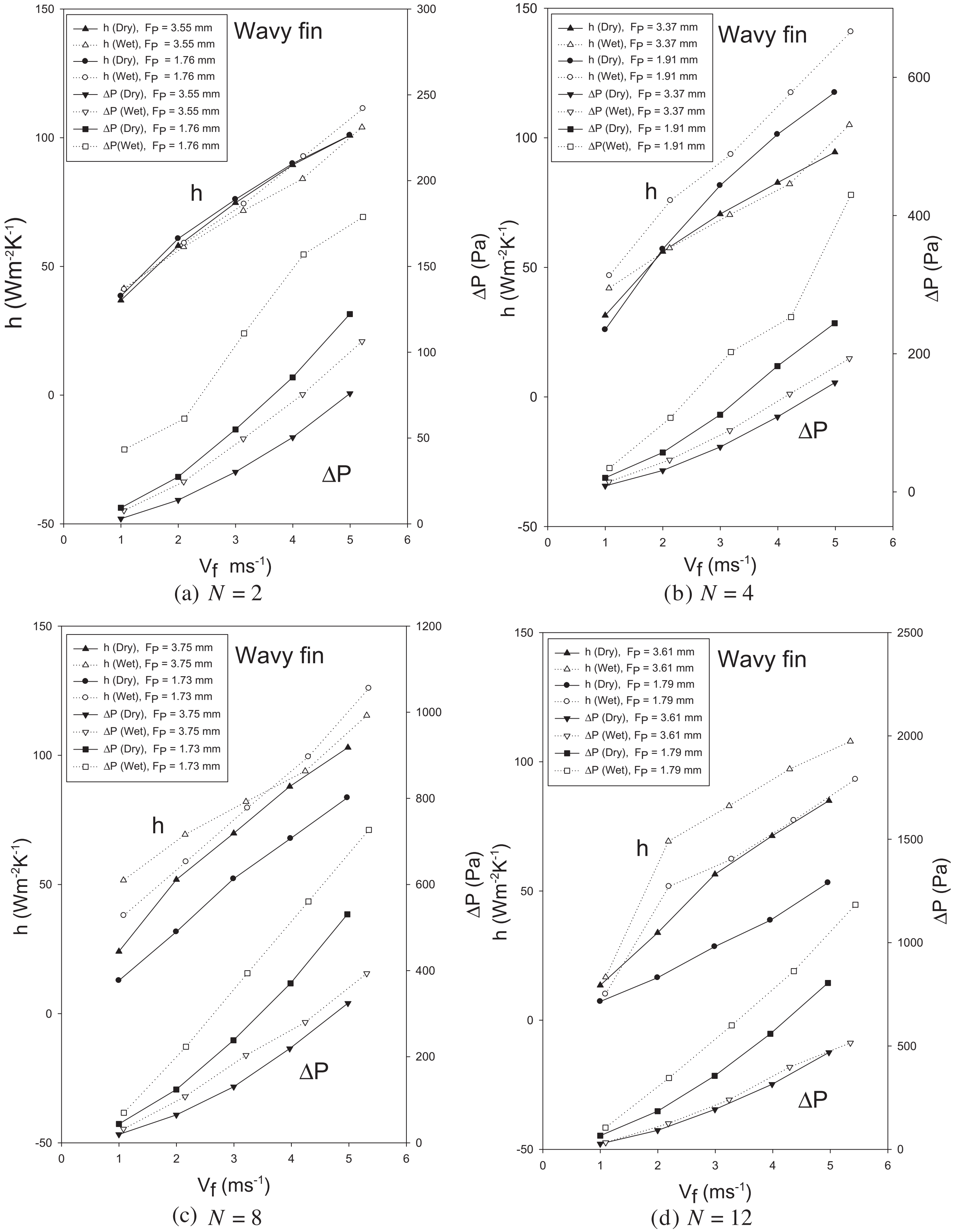
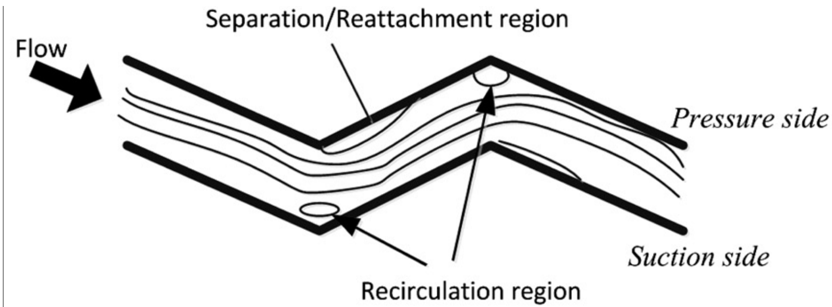
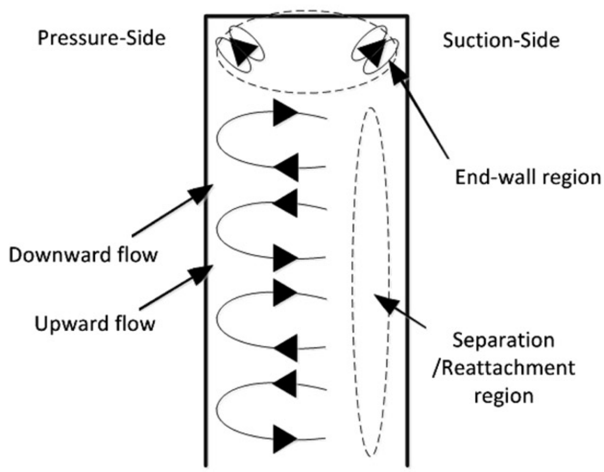


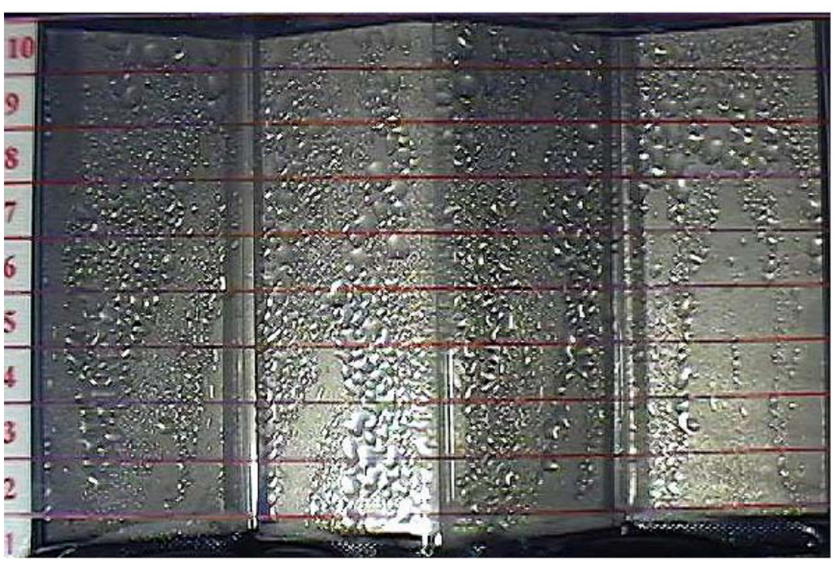

**Fig. 3.** Heat transfer coefficients and the pressure drops for the present wavy fin-and-tube heat exchangers in both dry and wet conditions.




(a) fully dry condition, side view (Hwang et al. [21]).




(b) fully dry condition, top view (Hwang et al. [21]).




(c) fully wet condition, top view (Lin et al. [22]).

**Fig. 4.** Schematic of airflow pattern in wavy fin channel subject to dry/wet surface condition.

Aside from the foregoing discussions, it is also observed that the relative increase of pressure drops for the present herringbone wavy fin-and-tube heat exchanger, when compared to that of plain fin geometry, is comparatively small. As shown in Fig. 5, for the same inlet humidity of 85%, normally the pressure drops for plain fin geometry in wet conditions is about 80–90% higher than that of dry condition. However, the associated increase for wavy fin geometry under wet condition is only around 50% or even lower. In practical applications, the wavy fin patterns can be either in the form of herringbone or smooth wavy fin patterns. Sparrow and Hossfeld [24] had examined the effect of rounding protruding edges of a wavy channel, and they reported a significant decrease in pressure drops by rounding the protruding edges. A recent study by Islamoglu [25] also unveiled a similar result that smoothing the

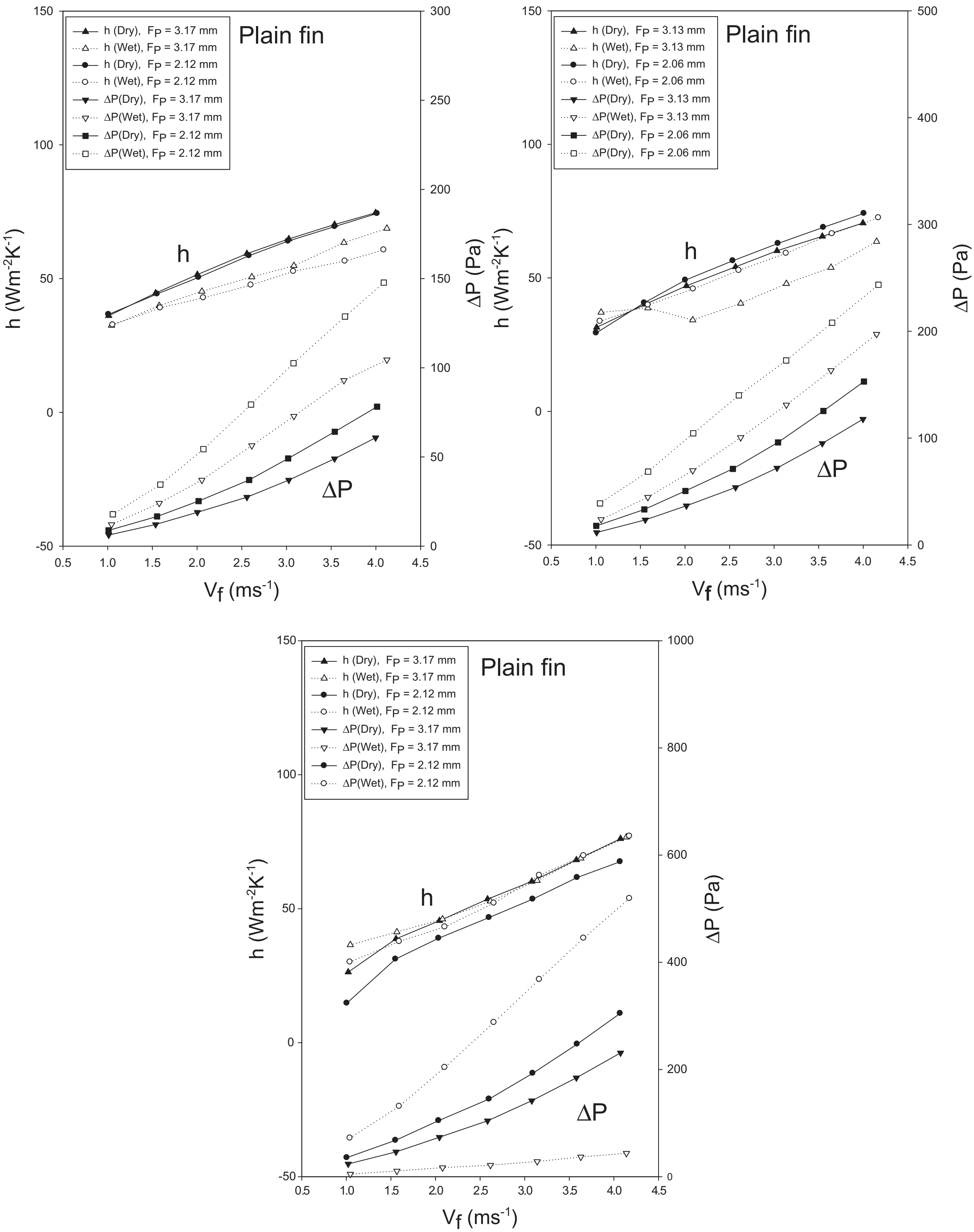



**Fig. 5.** Heat transfer coefficients and the pressure drops for the plain fin-and-tube heat exchangers in both dry and wet conditions.

apex of the wavy fin leads to a considerable reduction in pressure drops. For the present herringbone wavy fin-and-tube heat exchanger under dehumidifying condition, as clearly seen in Fig. 4(c), the presence of condensate plays a role in smoothing the valley and apex of the wavy fin geometry, resulting in an effective reduction of corrugation angle. Hence, although the presence of condensate may roughen the surface and increase the pressure drops, its presence also contributes to smooth apex/valley, offsetting the increased pressure drops by condensate. As a result, the dehumidifying wavy fin geometry shows only moderate increase of pressure drops relative to that in fully dry condition as compared to that of plain fin geometry.

## 5. Conclusions

This study presents the airside performance of the herringbone wavy fin-and-tube heat exchangers having a larger diameter tube ($D_c$ = 16.59 mm) pertaining to dehumidifying condition, and the test results are compared with that of dry conditions. A total of eight 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 controlled environment. Major conclusions of this study are summarized as follows:

(1) For dehumidifying condition with $N = 2$, the heat transfer coefficients are relatively independent of fin pitch. However, a detectable rise of heat transfer coefficient in wet condition with a smaller fin pitch ($N = 4$) is seen, but the tendency is reversed with a further rise of the number of tube row ($N = 8$, 12).
(2) Upon the influence of surface condition (dry or wet) on the heat transfer performance, the heat transfer performance in wet condition normally exceeds that in dry condition, and it becomes more pronounced with the rise of tube row or reduction of fin pitch. By contrast, it is found that the heat transfer coefficient for plain fin geometry in wet condition is slightly lower than that in dry condition.
(3) The pressure drops in wet condition is much higher than that in dry condition. However, the difference in pressure drop amid dry/wet condition is less profound for the wavy fin configuration as compared to that in plain fin geometry.

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## References

[1] Y.C. Liu, R. Hu, B.C. Yang, I.Y. Chen, C.C. Wang, Sensible airside performance of fin-and-tube heat exchangers – Data with larger diameter tube, ASHRAE Trans. 114 (1) (2008) 379–386.
[2] Y.C. Liu, S. Wongwises, W.J. Chang, C.C. Wang, Airside performance of fin-and-tube heat exchangers in dehumidifying conditions – Data with larger diameter, Int. J. Heat Mass Transfer 53 (2010) 1603–1608.
[3] D.T. Beecher, T.J. Fagan, Effects of fin pattern on the air-side heat transfer coefficient in plate finned-tube heat exchangers, ASHRAE Trans. 93 (2) (1987) 1961–1984.
[4] R.L. Webb, Air-side heat transfer correlations for flat and wavy plate fin-and-tube geometries, ASHRAE Trans. 96 (2) (1990) 445–449.
[5] C.C. Wang, W.L. Fu, C.T. Chang, Heat transfer and friction characteristics of typical wavy fin-and-tube heat exchangers, Exp. Thermal Fluid Sci. 14 (2) (1997) 174–186.
[6] C.C. Wang, Y.M. Tsi, D.C. Lu, A comprehensive study of convex-louver and wavy fin-and-tube heat exchangers, AIAA J. Thermophys. Heat Transfer 12 (3) (1998) 423–430.
[7] C.C. Wang, J.Y. Jang, N.F. Chiou, Effect of waffle height on the air-side performance of wavy fin-and-tube heat exchangers, Heat Transfer Eng. 20 (3) (1999) 45–56.
[8] C.C. Wang, J.Y. Jang, N.F. Chiou, A heat transfer and friction correlation for wavy fin-and-tube heat exchangers, Int. J. Heat Mass Transfer 42 (10) (1999) 1919–1924.
[9] C.C. Wang, Y.M. Hwang, Y.T. Lin, Empirical correlations for heat transfer and flow friction characteristics of herringbone wavy fin-and-tube heat exchangers, Int. J. Refrigeration 25 (2002) 653–660.
[10] Y. Chokeman, S. Wongwises, Effect of fin pattern on the air-side performance of herringbone wavy fin-and-tube heat exchangers, Int. J. Heat Mass Transfer 41 (2005) 642–650.
[11] S. Wongwises, Y. Chokeman, Effect of fin pitch and number of tube rows on the air side performance of herringbone wavy fin and tube heat exchangers, Energy Convers. Manage. 46 (2005) 2216–2231.
[12] C.C. Wang, Y.T. Lin, C.J. Lee, Y.J. Chang, An investigation of wavy fin-and-tube heat exchangers; A contribution to databank, Exp. Heat Transfer 12 (1999) 73–89.
[13] C.C. Wang, J.S. Liaw, B.C. Yang, Air-side performance of herringbone wavy fin-and-tube heat exchangers – Data with larger diameter tube, Int. J. Heat Mass Transfer 54 (2011) 1024–1029.
[14] ASHRAE Standard 41.2-1987, Standard Methods for Laboratory Air-flow Measurement, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, 1987.
[15] ASHRAE Standard 41.1-1986, Standard Method for Temperature Measurement, American Society of Heating, Refrigerating and Air-Conditioning Engineers, Inc., Atlanta, 1986.
[16] C.C. Wang, Y.J. Hsieh, Y.T. Lin, Performance of plate finned tube heat exchangers under dehumidifying conditions, J. Heat Transfer 119 (1997) 109–117.
[17] R.J. Moffat, Describing the uncertainties in experimental results, Exp. Thermal Fluid Sci. 1 (1988) 3–17.
[18] J.L. Threlkeld, Thermal Environmental Engineering, Prentice-Hall, Inc., New-York, 1970.
[19] 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.
[20] C.A. McNab, K.N. Atkinson, M.R. Heikal, Numerical modeling of heat transfer and fluid flow over herringbone corrugated fins, Heat transfer 6 (1998) 119–124.
[21] S.D. Hwang, I.H. Jang, H.H. Cho, Experimental study on flow and local heat/mass transfer characteristics inside corrugated duct, Int. J. Heat Fluid Flow 27 (2006) 21–32.
[22] Y.T. Lin, Y.M. Hwang, C.C. Wang, Performance of the herringbone wavy fin under dehumidifying conditions, Int. J. Heat Mass Transfer 45 (2002) 5035–5044.
[23] C.C. Wang, Y.C. Hsieh, Y.J. Chang, Y.T. Lin, Sensible heat and friction characteristics of plate fin-and-tube heat exchangers having plane fins, Int. J. Refrigeration 19 (4) (1996) 223–230.
[24] E.M. Sparrow, L.M. Hossfeld, Effect of rounding protruding edges on heat transfer and pressure drop in a duct, Int. J. Heat Mass Transfer 27 (1984) 1715–1723.
[25] Y. Islamoglu, Effect of rounding of protruding edge on convection heat transfer in a converging–diverging channel, Int. Commun. J. Heat Mass Transfer 35 (2008) 643–647.