An experimental investigation on the airside performance of fin-and-tube heat exchangers having slit fins under wet condition†

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Abstract

In this study, the heat transfer and friction characteristics of the 5.3 mm O.D. slit-finned heat exchangers under wet condition have been experimentally investigated. Plain-finned heat exchangers having the same 5.3 mm O.D. tubes are also tested for comparison purpose. The effect of fin pitch on j and f factor is negligible. Slit fin samples yield higher j and f factors than plain fin samples. For one row configuration, the average f factor ratio between slit fin sample and plain fin sample is 2.18. The ratio increases to 2.41 for two row configuration, and to 2.65 for three row configuration. As for the j factor, the ratios are approximately the same (1.61, 1.70 and 1.71 for one, two and three row configuration). Both j and f factor increase as the number of tube row decreases. The same trend is observed for the plain fin samples. At high Reynolds numbers, the j/f ratios of the slit fin are approximately the same as those of the plain fin. At low Reynolds numbers, the j/f ratios of the slit fin are smaller than those of plain fin. Data are compared with existing correlations.

Keywords: Heat exchanger; Heat transfer coefficient; Pressure drop; Fin; Slit; Wet

1. Introduction

Fin-and-tube heat exchangers have been widely used for heat exchange between an air and a liquid for many years. In forced convective heat transfer between air and liquid, the controlling thermal resistance is on the airside. To improve the airside performance, efforts have been made in two ways. One way is an usage of high performance fins. These include wavy fin, louver fin, convex louver fin, slit fin, etc. The other way is an usage of small diameter tubes. Circular tubes of fin-and-tube heat exchangers induce profile drag and form low performance region behind tubes. Such losses are proportional to the tube diameter, and could be reduced by adopting small diameter tubes. Heat exchangers having 7.0 mm diameter tubes are widely used for residential application. Very recently, however, 5.0 mm diameter tubes are finding applications. Webb and Kim [1] and Wang [2] provide recent progress on this issue.

In the evaporators of residential air-conditioners, the surface temperature of the fins is generally below the dew point temperature of the processing air, and moisture is condensed on the fins. Literature shows that many studies have been carried out on the wet heat transfer and pressure drop characteristics of the plain or wavy fin-and-tube heat exchangers [3-8]. On the slit fin, however, relatively little attention has been provided, considering that slit fins are widely used in evaporators of air-conditioning equipments. Geometric dimensions related with the slit-finned heat exchanger are shown in Fig. 1.

Fig. 1. Dimensions of slit fin-and-tube heat exchanger.
Table 1. Previous studies on slit fins.

| Author name       | Slit formed | \(w_s\) [mm] | \(h_s\) [mm] | \(n_x\) | \(A/A_t\) | \(P_l\) [mm] | \(P_t\) [mm] | \(P_f\) [mm] | \(D_c\) [mm] | \(N\) |
|-------------------|------------|--------------|--------------|---------|-----------|-------------|-------------|-------------|-------------|------|
| Wang and Chang [9]| Both sides | 1.0          | N/A          | 9       | 0.112     | 1.4         | 20.4        | 12.7        | 7.5         | 2   |
| Wang et al. [10]  | One side   | 2.2          | 1.0          | 4       | 0.203     | 1.21–2.46   | 25.4        | 22.0        | 10.3        | 1–4 |
| Wang et al. [11]  | Both sides | N/A          | 0.75/0.5     | 7       | N/A       | 1.48–2.50   | 25.0        | 21.65       | 10.32       | 1–4 |
| Ma et al. [13]    | One side   | 1.0          | 0.8          | 11      | 0.156     | 1.20–1.60   | 20.0        | 17.32       | 7.52        | 1–2 |
| Wang et al. [12]  | Both sides | 1.0          | 1.0          | 9       | N/A       | 1.27–1.81   | 21.0        | 12.7        | 7.6         | 1–3 |
| Yun et al. [14]   | One side   | N/A          | 0.6          | 4       | 0.258     | 1.2~1.7     | 21.0        | 13.3        | 7.21        | 2, 3 |

At a low flow velocity, heat transfer coefficients of the slit fin sample were nearly the same as those of the plain fin sample. The reason was attributed to the water condensate, which may block the slit and reduce the potential advantage of the slit fin. Wang et al. [10] investigated the effect of fin pitch and number of tube row for the slit-finned heat exchangers with the geometry provided in Table 1. At a large fin pitch of 2.5 mm, the effect of number of tube row was very small for both \(j\) and \(f\) factors. At a small fin pitch of 1.2 mm, on the other hand, both \(j\) and \(f\) factors decreased as the number of tube row increased. They explained that, at a large fin pitch, condensate drainage is much easier than at a small fin pitch, and \(j\) and \(f\) factors behave analogous to those of a plain fin-and-tube heat exchanger (which shows a negligible row effect). At a small fin pitch, the effect of condensate retention is significant, and significant row effect is manifested. Wang et al. [11] extended the study to three more slit fin geometries. Both for \(j\) and \(f\) factor, the effect of number of tube row and fin pitch was not pronounced. They developed \(j\) and \(f\) correlations based on their own data and Wang et al. [10]. Wang et al. [12] investigated the effect of hydrophilic coating for the slit-finned heat exchangers having the same 5.3 mm O.D. tubes. More investigation, especially on the effect of tube row, fin pitch, etc. is necessary.

In this study, heat transfer and pressure drop characteristics of slit-finned heat exchanger having 5.3 mm O.D. tubes are experimentally investigated under wet condition. Plain-finned heat exchangers having the same 5.3 mm O.D. tubes are also tested for comparison purpose. Fins are hydrophilic coated to ease the condensate drainage.

2. Experiments

2.1 Heat exchanger samples

A total of eighteen samples tested in the present study consist of nine plain fin samples and nine slit fin samples. The geometric parameters are listed in Table 2, and detailed dimensions of the slit fin are illustrated in Fig. 2. The height and the width of the samples are 234 mm (12 steps) and 400 mm, respectively. The slit fin has 7 slits of 1.0 mm slit width and 0.7 mm slit height. All slits are formed normal to the flow direction. For all the samples, the transverse tube pitch (\(P_t\)) is 19.5 mm, the longitudinal tube pitch (\(P_l\)) is 11.2 mm, and the tube diameter (\(D_c\)) is 5.3 mm. The \(D_c\) was determined by measuring the tube outer diameter (including fin collar) from the samples. The fin pitch was varied from 1.1 mm to 1.3 mm, and the number of tube row was varied from one to three. A micro-fin tube having 60 micro-fins with 0.12 mm fin height and 25 degree helix angle was used in the tube-side. The tube was circuited to cross-counter configuration with single inlet and outlet.

2.2 Test apparatus and procedures

A schematic drawing of the apparatus is shown in Fig. 3. It
consists of a suction-type wind tunnel, water circulation and control units, and a data acquisition system. The apparatus is situated in a constant temperature and humidity chamber. The airside inlet condition of the heat exchanger is maintained by controlling the chamber temperature and humidity. The inlet and outlet dry and wet bulb temperatures are measured by the sampling method as suggested in ASHRAE Standard 41.1 [15]. A diffusion baffle is installed behind the test sample to mix the outlet air. The waterside inlet condition is maintained by regulating the flow rate and inlet temperature of the constant temperature bath situated outside of the chamber. Both the air and the water temperatures are measured by pre-calibrated RTDs (Pt-100°Ω sensors). Their accuracies are ±0.1°C. The water flow rate is measured by a mass flow meter, whose accuracy is ±0.1% of full scale. The airside pressure drop across the heat exchanger is measured using a differential pressure transducer.

The air flow rate is measured using a nozzle pressure difference according to ASHRAE Standard 41.2 [16]. The accuracy of the differential pressure transducer is ±1.0 Pa.

During the experiment, the water temperature was held at 6°C. The chamber temperature was maintained at 35°C with

Table 2. Geometric dimensions of the test samples.

| No. | Fin pattern | \( w_s \) [mm] | \( h_s \) [mm] | \( n_s \) | \( A_s/A_f \) | \( P_f \) [mm] | \( P_l \) [mm] | \( D_c \) [mm] | \( t_f \) [mm] | \( N \) |
|-----|-------------|----------------|----------------|--------|-------------|--------------|--------------|--------------|-------------|------|
| 1   | Slit        | 1.0            | 0.7            | 7      | 0.314       | 1.1          | 19.5         | 11.2         | 5.3         | 0.11 | 1   |
| 2   | Slit        | 1.0            | 0.7            | 7      | 0.314       | 1.1          | 19.5         | 11.2         | 5.3         | 0.11 | 2   |
| 3   | Slit        | 1.0            | 0.7            | 7      | 0.314       | 1.1          | 19.5         | 11.2         | 5.3         | 0.11 | 3   |
| 4   | Slit        | 1.0            | 0.7            | 7      | 0.314       | 1.2          | 19.5         | 11.2         | 5.3         | 0.11 | 1   |
| 5   | Slit        | 1.0            | 0.7            | 7      | 0.314       | 1.2          | 19.5         | 11.2         | 5.3         | 0.11 | 2   |
| 6   | Slit        | 1.0            | 0.7            | 7      | 0.314       | 1.2          | 19.5         | 11.2         | 5.3         | 0.11 | 3   |
| 7   | Slit        | 1.0            | 0.7            | 7      | 0.314       | 1.3          | 19.5         | 11.2         | 5.3         | 0.11 | 1   |
| 8   | Slit        | 1.0            | 0.7            | 7      | 0.314       | 1.3          | 19.5         | 11.2         | 5.3         | 0.11 | 2   |
| 9   | Slit        | 1.0            | 0.7            | 7      | 0.314       | 1.3          | 19.5         | 11.2         | 5.3         | 0.11 | 3   |
| 10  | Plain       | N/A            | N/A            | N/A    | N/A         | 1.1          | 19.5         | 11.2         | 5.3         | 0.11 | 1   |
| 11  | Plain       | N/A            | N/A            | N/A    | N/A         | 1.1          | 19.5         | 11.2         | 5.3         | 0.11 | 2   |
| 12  | Plain       | N/A            | N/A            | N/A    | N/A         | 1.1          | 19.5         | 11.2         | 5.3         | 0.11 | 3   |
| 13  | Plain       | N/A            | N/A            | N/A    | N/A         | 1.2          | 19.5         | 11.2         | 5.3         | 0.11 | 1   |
| 14  | Plain       | N/A            | N/A            | N/A    | N/A         | 1.2          | 19.5         | 11.2         | 5.3         | 0.11 | 2   |
| 15  | Plain       | N/A            | N/A            | N/A    | N/A         | 1.2          | 19.5         | 11.2         | 5.3         | 0.11 | 3   |
| 16  | Plain       | N/A            | N/A            | N/A    | N/A         | 1.3          | 19.5         | 11.2         | 5.3         | 0.11 | 1   |
| 17  | Plain       | N/A            | N/A            | N/A    | N/A         | 1.3          | 19.5         | 11.2         | 5.3         | 0.11 | 2   |
| 18  | Plain       | N/A            | N/A            | N/A    | N/A         | 1.3          | 19.5         | 11.2         | 5.3         | 0.11 | 3   |
60% relative humidity. Experiments were conducted varying the frontal air velocity from 0.5 m/s to 2.0 m/s. The energy balance between the airside and the tube-side was within ±2% for most of the test range. It increased to ±5% at the lowest velocity. All the data signals were collected and converted by a data acquisition system (a hybrid recorder). The data were then transmitted to a personal computer for further manipulation. An uncertainty analysis was conducted following ASHRAE Standard 41.5 [17], and the results are listed in Table 3. The major uncertainty on the friction factor was the uncertainty of the differential pressure measurement (±10%), and the major uncertainty on the heat transfer coefficient (or j factor) was that of the tube-side heat transfer coefficient (±10%). The uncertainties decreased as the airside Reynolds number increased.

2.3 Data reduction

The data reduction details are provided by Mirth and Ramadhyani [18], Pirompugd et al. [19] and Kim et al. [20], and short summary is provided here. For the cross-counter configuration of the present study, appropriate equations for the heat exchanger analysis are given by ESDU 98005 [21], and are summarized in Table 4. The \(UA\) value is obtained from the following equations.

\[
UA = C_{\text{max}} \cdot NTU
\]
\[R = \frac{C_{\text{max}}}{C_{\text{min}}} = \min(\dot{m}_w, \dot{m}_c, \frac{c_p}{b_j}) / \max(\dot{m}_w, \dot{m}_c, \frac{c_p}{b_j}) \cdot \text{(1)}
\]

For the one row configuration, a cross-flow \(\varepsilon - NTU\) equation of mixed-unmixed configuration [22] was used. The airside heat transfer coefficient under wet condition \(h_a\) is obtained from the following equations.

\[
b_{w,m} = \frac{1}{\eta_a h_a A_j} - \frac{b_t}{h_t A_j} - \frac{b_f}{k A_j} \cdot \text{(3)}
\]

\[
h_a = \frac{h_{w,m}}{b_{w,m}} \cdot \text{(4)}
\]

Here, \(b_w, b_t, b_{w,m}\) are the slope of saturated air enthalpy at the water, tube wall, and water film temperature respectively. For the tube-side heat transfer coefficients, the correlation [23] developed for the 7.0 mm micro-fin tube was used. The usage of the 7.0 mm tube correlation to the 5.0 mm tube may be justified because the correlation is of non-dimensional form. In addition, the small portion of the tube-side thermal resistance (less than 5% of the total resistance) further justifies the usage of the 7.0 mm tube correlation. The correlation is as follows.

\[
Nu_a = 0.00172 \cdot Re_a^{0.12} \cdot Pr_a^{0.31} \cdot \text{min}(3,000 \leq Re_a \leq 21,000) \cdot \text{(5)}
\]

\[
Nu_a = 0.0376 \cdot Re_a^{0.81} \cdot Pr_a^{0.3} \cdot \text{min}(21,000 \leq Re_a \leq 45,000) \cdot \text{(6)}
\]

As the characteristic length of the Nusselt or the Reynolds number in Eqs. (5) and (6), the maximum diameter to the fin root was adopted. The surface efficiency \(\eta_s\) for use in Eq. (3) is obtained from Eq. (7).

\[
\eta_s = 1 - \frac{A_f}{A_e} (1 - \eta) \cdot \text{(7)}
\]

The fin efficiency is given by Schmidt [24] as

\[
\eta = \frac{\tanh(mr,\varphi)}{mr,\varphi} \cdot \text{(8)}
\]

where

\[
m = \sqrt{2b_m} \cdot \sqrt{\frac{1}{f}} \cdot \text{(9)}
\]

\[
\varphi = \frac{R_w}{r_e} \left[ 1 + 0.35 \ln \left( \frac{R_w}{r_e} \right) \right] \cdot \text{(10)}
\]

\[
R_w = 1.28 \cdot \frac{P}{r_e} \left[ \sqrt{\frac{P}{2}} + \frac{P_t}{P} - 0.2 \right]^{0.5} \cdot \text{(1 row)} \cdot \text{(11)}
\]

\[
R_w = 1.27 \cdot \frac{P}{r_e} \left[ \sqrt{\frac{P}{2}} + \frac{P_t}{P} - 0.3 \right]^{0.5} \cdot \text{(over 2row)} \cdot \text{(12)}
\]

With Eqs. (7)-(12), an iterative procedure is needed to obtain the airside heat transfer coefficient \(h_a\). In the figures, heat transfer coefficients are presented as j factors.

\[
Re_{\text{in}} = \frac{\rho V_{\text{in}} D_c}{\mu_c} \cdot \text{(13)}
\]

\[
j = \frac{h_t}{\rho V_{\text{in}} c_p} \cdot \text{(14)}
\]

All the fluid properties are evaluated at an average air tem-
The core friction factor is calculated from the measured pressure drop.

\[ f = \frac{A_t}{A_s} \frac{2\Delta \rho_p}{\rho_{sv} (\rho_{sv} V_{sv})^2 - (1 + \sigma^2) [\frac{\rho_p}{\rho_{sw}} - 1]} \]  

(15)

In Eq. (15), the entrance and the exit loss coefficients are neglected following the suggestion by Wang et al. [25]. The friction factors obtained by including the loss coefficients into Eq. (15) agreed with those obtained from Eq. (15) within 1%.

### 3. Results and discussions

Fig. 4 shows the effect of fin pitch on the j and f factors of the present plain and slit fin geometries. The figure shows that effect of fin pitch on j and f factor is negligible. It has been reported by many investigators that the effect of fin pitch on the heat transfer coefficient and friction factor is not significant [5, 26, 27]. Considering the small variation of the fin pitch (from 1.1 to 1.3 mm), the foregoing argument should hold true for the present samples. Fig. 4 show that slit fin samples yield higher j and f factors than plain fin samples. For one row configuration, the ratio increases to 2.41 for two row configuration, and to 2.65 for three row configuration. As for the j factor, the ratios are approximately the same (1.87, 1.79 and 1.82 for one, two and three row configuration). In summary, wet f factor ratio is larger than dry f factor ratio. On the other hand, wet j factor ratio is smaller than dry j factor ratio. The reason may be attributed to the difference in condensate drainage pattern. For plain fin, condensate drain smoothly.

For slit fin, condensate may block the slits, deteriorates the heat transfer and increases the pressure drop.

The effect of tube row on wet j and f factor of slit fin samples along with those of plain fin samples are shown in Fig. 5. For both slit and plain fin samples, j factor increases as the number of tube row decreases. Careful inspection of slit fin f factor reveals that f factor also increases as the number of tube row increases.

This trend has been observed by many investigators for plain or slit fin geometry [10, 12, 13, 28, 29] and could be explained by the boundary layer effect. On the fin surface, the boundary layer gets thicker as the number of tube row increases. Then, both the heat transfer and pressure drop decrease as the number of tube row increases.

In Fig. 6, j/f ratios of slit fin samples are compared with those of plain fin samples for 1.2 mm fin pitch. At high Reynolds numbers, the j/f ratios of the slit fin are approximately the same as those of the plain fin. At low Reynolds numbers, on the other hand, the j/f ratios of the slit fin are smaller than those of plain fin. The efficiency index \((\Delta f)_{slit}/(\Delta f)_{plain}\) is the largest for three row sample, followed by two row and one row sample. This implies that slit fin geometry is more beneficial at high Reynolds numbers and at large number of row.

As mentioned in the introduction, two correlations [11, 13] are available to predict the wet j and f factors of slit fin-and-
tube heat exchangers. The correlations are listed in Table 5. The present data are compared with the correlation, and the results are shown in Fig. 7. RMS errors of the correlations are listed in Table 6. Fig. 7(a) along with Table 6 show that \( j \) factors are underpredicted by both Ma et al. [13] and Wang et al. [11] correlation with RMS error of 0.21 and 0.37 respectively. Fig. 7(b) along with Table 6 show both Ma et al. [13] and Wang et al. [11] correlation underpredict the \( f \) factors. The RMS errors are 0.16 and 0.54 respectively. In Fig. 8, plain fin data are also compared with Wang et al. [5] correlation.
Although McQuistion [4] provided wet surface j and f correlation, his correlations were not considered because of narrow applicable range of the correlations (limited to 9.6 mm O.D. tube, 4 row). Fig. 8 shows that j factors are overpredicted (RMSE = 0.29) and f factors are underpredicted (RMSE = 0.43). The poor predictions of j and f factors are anticipated because Wang et al. [5] correlations were developed based on his own data (limited to 10.23 mm tube O.D.).

4. Conclusions

In this study, heat transfer and friction characteristics of the 5.3 mm O.D. slit-finned heat exchangers under wet condition are experimentally investigated. Plain-finned heat exchangers having the same 5.3 mm O.D. tubes are also tested for comparison purpose. Focus was given to the effect of fin pitch as
well as the number of tube row. Listed below are major find-
(1) The effect of fin pitch on $j$ and $f$ factor is negligible.
Considering the small variation of the fin pitch (from 1.1 to
1.3 mm), the foregoing argument should hold true for the pre-
(2) Slit fin samples yield higher $j$ and $f$ factors than plain fin
samples. For one row configuration, the average $f$ factor ratio
between slit fin sample and plain fin sample is 2.18. The ratio
increases to 2.41 for two row configuration, and to 2.65 for
three row configuration. As for the $j$ factor, the ratios are ap-
proximately the same (1.61, 1.70 and 1.71 for one, two and
three row configuration).
(3) Both $j$ and $f$ factor increases as the number of tube row
degrates. The same trend is also observed for the plain fin
samples.
(4) At high Reynolds numbers, the $j/f$ ratios of the slit fin
are approximately the same as those of the plain fin. At low
Reynolds numbers, on the other hand, the $j/f$ ratios of the slit
fin are smaller than those of plain fin.
(5) Both $j$ and $f$ factors are reasonably predicted by Ma et al.
[13] correlation.

Nomenclature—

$A$ : Heat transfer area, $m^2$
$A_c$ : Minimum free flow area, $m^2$
$A_s$ : Slit area, $m^2$
$A_{t}$ : Heat transfer area at the mid-plane of the tube wall, $m^2$
$C$ : Heat capacity, $W/K$
$c_p$ : Specific heat, $J/kg \cdot K$
$D$ : Tube diameter including fin collar thickness, m
$D_h$ : Hydraulic diameter, m
$D_r$ : Maximum tubeside diameter (to the fin root), m
$f$ : Friction factor
$G$ : Mass flux, $kg/m^2s$
$h$ : Heat transfer coefficient, $W/m^2K$
$h_s$ : Height of the slit, m
$j$ : Colburn $j$ factor
$k_t$ : Thermal conductivity of the tube, $W/m \cdot K$
$L$ : Air flow length in the heat exchanger, m
$l_s$ : Length of the slit, m
$m$ : Mass flow rate, $kg/s$
$n_s$ : Number of slits
$N$ : Number of tube row
$NTU$ : Number of transfer units
$Nu$ : Nusselt number,
$P_{f}$ : Fin pitch, m
$P_t$ : Transverse tube pitch, m
$P_{l}$ : Longitudinal tube pitch, m
$Pr$ : Prandtl number
$R$ : Heat capacity ratio
$r_c$ : Tube radius including fin collar, m
$R_{eq}$ : Equivalent radius, m
$Re_{DB}$ : Reynolds number based on $D_h$
$Re_{wa}$ : Waterside Reynolds number
$Re_{ws}$ : Reynolds number based on $w_s$
$s$ : Fin spacing, m
$t$ : Tube wall thickness, m
$T$ : Temperature, K
$t_f$ : Fin thickness, m
$U$ : Overall heat transfer coefficient, $W/m^2K$
$V_{max}$ : Maximum airside velocity, $m/s$
$w_s$ : Width of the slit, m

Greek symbols

$\varepsilon$ : Effectiveness
$\Delta P$ : Pressure loss, Pa
$\Gamma$ : Condensate flow rate per unit width, $kg/m \cdot s$
$\eta$ : Fin efficiency
$\eta_s$ : Surface efficiency
$\phi_s$ : Ratio of slit area to total fin area
$\rho$ : Density, $kg/m^3$
$\mu$ : Dynamic viscosity, $kg/m \cdot s$
$\sigma$ : Contraction ratio of the cross-sectional area

Subscripts

$a$ : Air
$exp$ : Experimental
$i$ : Tubeside
$in$ : Inlet
$f$ : Fin
$m$ : Mean
$o$ : Airside
$p$ : Plain
$pred$ : Prediction
$out$ : Outlet
$s$ : Slit
$w$ : Water

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