An experimental investigation on the airside performance of fin-and-tube heat exchangers having corrugated louver fins - Part II; wet surface

Nae-Hyun KIM*
*Division of Mechanical System Engineering, Incheon National University
12-1 Songdo-Dong, Yeonsu-Gu, Incheon 406-772, Korea
E-mail: knh0001@inu.ac.kr

Received: 26 November 2019; Revised: 28 January 2020; Accepted: 25 February 2020

Abstract
In this study, two kinds of corrugated louver fin-and-tube heat exchangers – one having one corrugation per row and the other having two corrugations per row – were tested under wet condition, and the results were compared with those of the standard louver fin and the plain fin samples. The highest $j$ and $f$ factor were obtained for the double corrugated louver fin sample, followed by the single corrugated inclined louver fin sample, the standard louver fin sample and then the plain fin sample. This result is in contradiction with those obtained under dry condition, where the standard louver fin sample yielded the highest $j$ and $f$ factor. The high $j$ and $f$ factor of the corrugated louver fin sample may be due to the improved condensate drainage over the standard louver fin sample. Corrugated channels along with louvers of high louver angle of the corrugated louver fin samples appear to have resulted better condensate drainage. All the enhanced fin samples yielded larger heat transfer capacity than plain fin samples at the same pumping power. Furthermore, the largest heat transfer capacity per pumping power was obtained for the double corrugated louver fin sample, followed by the single corrugated louver fin sample and then the standard louver fin sample.

Keywords: Heat exchanger, Heat transfer coefficient, Pressure drop, Fin, Corrugated, Louver, Wet

Nomenclature

$A$ \hspace{1cm} \text{heat transfer area, m}^2

$A_c$ \hspace{1cm} \text{minimum free flow area, m}^2

$A_l$ \hspace{1cm} \text{louver area, m}^2

$A_s$ \hspace{1cm} \text{slit area, m}^2

$A_t$ \hspace{1cm} \text{heat transfer area at the mid-plane of the tube wall, m}^2

$b_f$ \hspace{1cm} \text{slope of the saturated air enthalpy curve at fin temperature, J/kgK}

$b_r$ \hspace{1cm} \text{slope of the saturated air enthalpy curve between tube wall and water temperature, J/kgK}

$b_t$ \hspace{1cm} \text{slope of the saturated air enthalpy curve at tube wall temperature, J/kgK}

$C$ \hspace{1cm} \text{heat capacity, W/K}

$c_p$ \hspace{1cm} \text{specific heat, J/kgK}

$D$ \hspace{1cm} \text{tube diameter including fin collar thickness, m}

$f$ \hspace{1cm} \text{friction factor}

$h$ \hspace{1cm} \text{heat transfer coefficient, W/m}^2\text{K}

$h_s$ \hspace{1cm} \text{height of the slit, m}

$i$ \hspace{1cm} \text{enthalpy, J/kg}

$j$ \hspace{1cm} \text{Colburn j factor}

$k$ \hspace{1cm} \text{thermal conductivity, W/mK}
1. Introduction

This article is a companion paper with Kim (2020), where the dry surface heat transfer and pressure drop characteristics of the corrugated louver fin-and-tube heat exchangers are discussed. In this article, wet surface characteristics of the same samples are discussed. Schematic drawing of the standard louver fin and two variants – convex and corrugated louver fin – is shown in Fig. 1. When the fin surface temperature is lower than the dew point temperature of the processing air, the moisture is condensed on the fins. In this case, total heat transfer consists of a sensible heat transfer by temperature difference and a latent heat transfer by moisture concentration difference, and the heat transfer performance is commonly presented as a wet surface heat transfer.
coefficient (or j factor), which is obtained through an appropriate analysis of heat and mass transfer (Kuehn et al., 1998). The pressure drop under wet condition generally increases compared with that under dry condition, and is commonly represented as a friction factor (or f factor).

The literature shows several studies on thermal performance of louver fin-and-tube heat exchangers under wet condition, which are summarized in Table 1. Wang and Chang (1998) investigated the effect of hydrophilic coating on wet surface heat transfer and pressure drop for louver fin samples having two different louver geometry—one with 3.0 mm louver pitch and 15.5° louver angle, the other 1.7 mm louver pitch and 25° louver angle. The hydrophilic coating did not affect the j factor. However, f factor decreased by 15 to 40% for the coated sample. Furthermore, at frontal air velocities lower than 0.7 m/s, j factors of louver fin sample were approximately the same as those of the plain fin sample. The reason was attributed to the blocking of inter-louver spacing by consensate, which mitigated the augmentation effect by louvers.

Wang et al. (2000) tested louver fin samples having two different geometry—one with 2.35 mm louver pitch and 27° louver angle, the other 2.0 mm louver pitch and 32° louver angle, and showed that j factor was relatively independent of fin pitch, whereas f factor increased with the decrease of fin pitch. They also proposed a correlation based on their data. Hong and Webb (1999) also investigated the effect of hydrophobic coating on louver fin samples. Similar conclusion to Wang and Chang (1998) that the effect was negligible on j factor, whereas f factor decreased significantly for the coated sample was reported. Ma et al. (2007) tested louver fin samples having 3.5 mm louver pitch and 15.7° louver angle. The effect of fin pitch on j and f factor was negligible. Furthermore, both j and f factor decreased with the increase of number of tube row. They also proposed j and f correlations based on their data. Kim (2015) tested nine louver fin samples having 1.6 mm louver pitch and 24° louver angle. Similar to the results of Wang et al. (2000), j factor was independent of fin pitch, whereas f factor increased as fin pitch decreased. Furthermore, both j and f factor decreased as number of tube row increased.

The literature survey reveals that no prior study exists for wet surface heat transfer of corrugated louver fin-and-tube heat exchangers. In this study, corrugated louver fin samples were tested under wet condition, and the results are compared with those of louver and plain fin samples (Kim, 2016). The dry surface heat transfer and pressure drop characteristics of the present corrugated louver fin samples are discussed in a companion paper (Kim, 2020).

Table 1 Previous studies on louver fin-and-tube heat exchangers

| Author name          | \( L_p \) (mm) | \( \Theta \) (deg) | \( n_l \) | \( A_l/A_f \) | \( P_l \) (mm) | \( P_f \) (mm) | \( P_l/P_f \) | \( D \) (mm) | \( N \) |
|----------------------|----------------|-------------------|---------|--------------|---------------|---------------|---------------|-------------|------|
| Wang and Chang (1998)| 3.0            | 15.5              | 11      | N/A          | 1.4           | 20.4          | 16.7          | 0.82        | 9.87 | 2    |
| Wang et al. (2000)   | 2.35           | 27.0              | 9       | N/A          | 1.21–2.49     | 25.4          | 19.05         | 0.75        | 9.0–9.5 | 1.2 |
|                      | 2.0            | 32.0              | 11      | N/A          | 1.21–2.49     | 25.4          | 22.0          | 0.87        | 7.0    | 2   |
| Hong and Webb (1999) | N/A            | N/A               | 3       | N/A          | 1.49          | 25.4          | 22.0          | 0.87        | 7.0    | 2   |
| Ma et al. (2007)     | 3.5            | 15.7              | 3       | 0.37         | 1.4–1.8       | 25.4          | 19.05         | 0.75        | 9.53   | 2.2 |
| Kim (2016)           | 1.6            | 24.0              | 6       | 0.335        | 1.3–1.5       | 21.0          | 12.7          | 0.6         | 7.3    | 1–3 |

Fig. 1 Geometric dimensions fin shapes of louver fin-and-tube heat exchangers
2. Experiments

2.1 Heat exchanger samples

Twelve samples - six single corrugated louver fin samples (1.5, 1.7 mm fin pitch, 1~3 row) and six double corrugated louver fin samples (1.5, 1.8 mm fin pitch, 1~3 row) - were tested in this study. The geometric parameters are listed in Table 2, and detailed dimensions of the 2 row louver fin samples are illustrated in Fig. 2. The height and the width of the sample were 252 mm (12 steps) and 400 mm respectively. The samples were hydrophilic coated using non-siliceous composite to facilitate the condensate drainage.

2.2 Test apparatus and procedures

The apparatus and the test procedure are described in detail in Kim (2016), and only a short summary is provided here. Tests were conducted in the apparatus shown in Fig. 3, which consisted of a suction-type wind tunnel and a water circulation unit. The apparatus was situated in a constant temperature and humidity chamber, where temperature and humidity were controlled using an air conditioning unit and a steam boiler. The air temperatures were measured by the sampling method as suggested in ASHRAE Standard 41.1 (1986). The water temperatures were measured by pre-calibrated RTDs (Pt-100Ω sensors), and the water flow rate was measured by a mass flow meter. The airside pressure drop was measured using a differential pressure transducer, and the air flow rate was measured using a nozzle pressure difference according to ASHRAE Standard 41.2 (1987).
Table 2  Geometric dimensions of the test samples

| No. | Fin pattern | $L_u$ (mm) | $\theta$ (deg) | $\beta$ (deg) | $n_i$ | $A_i/A_f$ | $P_f$ (mm) | $P_t$ (mm) | $P_f/P_t$ | $D$ (mm) | $N$ |
|-----|-------------|------------|----------------|---------------|-------|------------|------------|------------|------------|----------|-----|
| 1   | Single Corr. | 0.92       | 21.0           | 10.8          | 8     | 0.261      | 1.5        | 21.0       | 18.2       | 0.87     | 7.3  | 1  |
| 2   | Single Corr. | 0.92       | 21.0           | 10.8          | 8     | 0.261      | 1.5        | 21.0       | 18.2       | 0.87     | 7.3  | 2  |
| 3   | Single Corr. | 0.92       | 21.0           | 10.8          | 8     | 0.261      | 1.5        | 21.0       | 18.2       | 0.87     | 7.3  | 3  |
| 4   | Single Corr. | 0.92       | 21.0           | 10.8          | 8     | 0.261      | 1.7        | 21.0       | 18.2       | 0.87     | 7.3  | 1  |
| 5   | Single Corr. | 0.92       | 21.0           | 10.8          | 8     | 0.261      | 1.7        | 21.0       | 18.2       | 0.87     | 7.3  | 2  |
| 6   | Single Corr. | 0.92       | 21.0           | 10.8          | 8     | 0.261      | 1.7        | 21.0       | 18.2       | 0.87     | 7.3  | 3  |
| 7   | Double Corr. | 0.8        | 33.0           | 38.0          | 24    | 0.259      | 1.5        | 21.0       | 21.6      | 1.03     | 7.94 | 1  |
| 8   | Double Corr. | 0.8        | 33.0           | 38.0          | 24    | 0.259      | 1.5        | 21.0       | 21.6      | 1.03     | 7.94 | 2  |
| 9   | Double Corr. | 0.8        | 33.0           | 38.0          | 24    | 0.259      | 1.5        | 21.0       | 21.6      | 1.03     | 7.94 | 3  |
| 10  | Double Corr. | 0.8        | 33.0           | 38.0          | 24    | 0.259      | 1.8        | 21.0       | 21.6      | 1.03     | 7.94 | 1  |
| 11  | Double Corr. | 0.8        | 33.0           | 38.0          | 24    | 0.259      | 1.8        | 21.0       | 21.6      | 1.03     | 7.94 | 2  |
| 12  | Double Corr. | 0.8        | 33.0           | 38.0          | 24    | 0.259      | 1.8        | 21.0       | 21.6      | 1.03     | 7.94 | 3  |
| 13  | Std. Louver  | 1.4        | 24             | -             | 7     | 0.335      | 1.5        | 21.0       | 12.7      | 0.6      | 7.3  | 1  |
| 14  | Std. Louver  | 1.4        | 24             | -             | 7     | 0.335      | 1.5        | 21.0       | 12.7      | 0.6      | 7.3  | 2  |
| 15  | Std. Louver  | 1.4        | 24             | -             | 7     | 0.335      | 1.5        | 21.0       | 12.7      | 0.6      | 7.3  | 3  |
| 16  | Plain        | -          | -              | -              | -     | -          | 1.5        | 21.0       | 12.7      | 0.6      | 7.3  | 1  |
| 17  | Plain        | -          | -              | -              | -     | -          | 1.5        | 21.0       | 12.7      | 0.6      | 7.3  | 2  |
| 18  | Plain        | -          | -              | -              | -     | -          | 1.5        | 21.0       | 12.7      | 0.6      | 7.3  | 3  |

During the experiment, the water temperature was held at 6°C and the chamber temperature was maintained at 35°C with 60% relative humidity. Experiments were conducted varying the frontal air velocity from 0.5 m/s to 1.5 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. An uncertainty analysis was conducted following ASHRAE Standard 41.5 (1975), and the results are listed in Table 3. The major uncertainty on the friction factor was the uncertainty of the differential pressure transducer (±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 Pirompugd et al. (2009) and a 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 (1998). The $UA$ value is obtained from the following equations.

$$UA = C_{min} NTU$$

(1)

$$R = \min(m_{i,a}c_{pm}/b_r)/\max(m_{i,a}c_{pm}/b_r)$$

(2)

Here, $b_r$ is the slope of saturated air enthalpy between tube wall and water temperature.

$$b_r = \frac{\Delta T}{\Delta T_{wi}}$$

(3)
The airside heat transfer coefficient under wet condition \( (h_o) \) is obtained from the following equations.

\[
\frac{b_f}{\eta_o h_o A_o} = \frac{1}{UA} \left( \frac{b_y}{h_t A_t} - \frac{b_y T}{k A_t} \right) \tag{4}
\]

\[
h_o = \frac{h_o c_{pm}}{b_f} \tag{5}
\]

Here, \( b_y \) and \( b_t \) are the slope of saturated air enthalpy at the average fin and tube wall temperature.

\[
b_f = \frac{\Delta i_{x,f}}{\Delta T_{s,f}} \tag{6}
\]
For the tube-side heat transfer coefficients, Park et al. (1997) correlation, which was developed using the present 7.0 mm O.D. micro-fin tube, was used. The surface efficiency $\eta_s$ for use in Eq. (3) is obtained from Eq. (8).

$$\eta_s = 1 - \frac{A_f}{A_o} (1 - \eta)$$  \hspace{1cm} (8)

where the fin efficiency $\eta$ is given by Schmidt (1949). In the figures, heat transfer coefficients are presented as $j$ factors, and the flow velocities are presented as Reynolds numbers.

$$Re_D = \frac{\rho_a V_{\text{max}} D}{\mu_a}$$ \hspace{1cm} (9)

$$j = \frac{h_o}{\rho_a V_{\text{max}} c_{pa}} Pr_a^{2/3}$$ \hspace{1cm} (10)

The core friction factor ($f$) is calculated from the measured pressure drop.

$$f = \frac{A_o \rho_m}{A_o \rho_{in}} \left[ \frac{2 \Delta P_{\text{in}}}{(\rho_m V_{\text{max}})^2} - (1 + \sigma^2) \left( \frac{\rho_{in}}{\rho_{out}} - 1 \right) \right]$$ \hspace{1cm} (11)

3. Results and discussions

Figures 4 and 5 show the effect of fin pitch on $j$ and $f$ factor of single and double corrugated louver fin samples. The figures show that, for both samples, effect of fin pitch on $j$ and $f$ factor is negligible. As mentioned previously, existing investigations on standard louver fin samples show negligible effect of fin pitch on $j$ and $f$ factor [Ma et al. (2007), Kim (2016)]. In Figs. 6 and 7, the effect of number of tube row is shown for single and double corrugated louver fin samples. For both configurations, the effect of number of tube row on $j$ and $f$ factor is significant. Both $j$ and $f$ factor decrease as number of tube row increases. Again, these trends are commonly observed in fin-and-tube heat exchangers including louver fin samples [Ma et al. (2007), Kim (2016)]. In Figs 6 and 7, dry surface $j$ and $f$ factors of the same samples reported by Kim (2020) are also shown. Both $j$ and $f$ factor are larger under wet condition with $f$ factor at a greater amount. To be more specific, the ratios between wet $j$ factor and dry $j$ factor taken at ReD = 1000 were 1.47, 1.51, 1.49 for 1 row, 2 row, 3 row for single corrugated louver fin samples, and 1.61, 1.55, 1.83 for double corrugated louver fin samples. The ratios between wet $f$ factor and dry $f$ factor are 2.48, 2.19, 2.14 for single corrugated samples and 2.17, 1.82, 1.88 for double corrugated samples. In general, $j$ and $f$ factors under wet condition are known to be higher than those under dry condition due to the existence of condensate droplets or films on the fin surface (Kim and Webb, 2005). However, when inter-louver spacings are blocked by condensate as noted by Wang et al. (1998), $j$ and $f$ factor of the wet surface may be lower than those of the dry surface. Figures 6 and 7 show that this is not the case for the present corrugated louver fin samples.
Fig. 4 Effect of fin pitch on $j$ and $f$ factor of the single corrugated louver fin samples

(a) 1 row
(b) 2 row
(c) 3 row

Fig. 5 Effect of fin pitch on $j$ and $f$ factor of the double corrugated louver fin samples

(a) $P_f = 1.5$ mm  
(b) $P_f = 1.7$ mm

Fig. 6 Effect of number of tube row on $j$ and $f$ factor of the single corrugated louver fin samples

(a) $P_l = 1.5$ mm
(b) $P_l = 1.8$ mm

Fig. 7 Effect of number of tube row on $j$ and $f$ factor of the double corrugated louver fin samples
Fig. 8 Comparison of the $j$ and $f$ factors of the present samples with those of the standard louver and the plain fin samples [Kim (2016)] at $P_f = 1.5$ mm
In Fig. 8, \( j \) and \( f \) factors of the present samples are compared with those of the standard louver and the plain fin samples (Kim, 2016). Table 2 shows that the common fin pitch of all the samples is 1.5 mm, and the comparison is made at that fin pitch. Figure 8 shows that the highest \( j \) and \( f \) factor were obtained for the single corrugated louver fin, followed by the double corrugated louver fin, the standard louver fin and then the plain fin sample. This point is further elaborated in Fig. 9.

Figure 9 compares heat transfer enhancement factors (\( j/j_p \)) and pressure drop penalty factors (\( f/f_p \)) of the enhanced fin samples. The factors were obtained from Fig. 8 by dividing the \( j \) and the \( f \) factor of the enhanced fin samples by those of the plain fin sample at \( \text{Re}_D = 1000 \). Figure 9 shows that the trends are similar irrespective of number of tube row. The highest \( j \) factors were obtained for single corrugated louver fin sample, except for one row, where double corrugated inclined louver fin sample yielded slightly higher \( j \) factor. The lowest \( j \) factors were obtained for standard louver fin sample. As for the \( f \) factor, the highest values were obtained for the single corrugated samples, whereas double corrugated and standard samples yielded approximately the same values. To be specific, \( j/j_p \) of the single corrugated louver fin samples were 2.34, 1.89, 2.23 and \( f/f_p \) were 1.89, 2.06, 2.02 for one row, two row and three row configuration respectively. For the double corrugated louver fin samples, those were 2.43, 1.91, 2.15 for \( j/j_p \) and 1.39, 1.39, 2.46 for \( f/f_p \). For the standard louver fin samples, those were 2.15, 1.80, 1.73 for \( j/j_p \) and 1.45, 1.39, 1.52 for \( f/f_p \).

Also shown in Fig. 9 are \( j/j_p \) and \( f/f_p \) under dry conditions (Kim, 2020). The trends are quite different from those obtained under wet condition. The standard louver fin sample, which yielded the lowest \( j \) and \( f \) factors under wet condition, show the highest \( j \) and \( f \) factors under dry condition. According to Kim (2020), the larger louver to fin area (\( A_l/A_f = 0.335 \)) of the standard louver fin over those of single corrugated louver fin (\( A_l/A_f = 0.261 \)) and double corrugated louver fin (\( A_l/A_f = 0.259 \)) was responsible for the high \( j \) and \( f \) factor.

Then, why the trend was reversed under wet condition? Under wet condition, proper drainage of condensate along the fin surface is very important, especially for enhanced fins. Considering that the main heat transfer enhancement mechanism of the louver geometry is periodic disruption and renewal of boundary layer along the flow direction,
blocking of the inter-louver spacing by poorly-drained condensate will deteriorate the heat transfer performance. McLaughlin and Webb (2000a) identified two different types condensate formation on louvered surfaces – one formed between louvers (louver bridging), and the other formed between fins (fin bridging). It was observed louver bridging increased as the louver pitch decreased. Fin bridging increased as hydrophilicity of the fin surface decreased or fin pitch decreased. For the present samples, fins were hydrophilic coated, and fin bridging was not observed. As shown in Fig. 2, the louver angle of the inclined louver fin (33° for the single corrugated louver fin, 35° for the double corrugated louver fin) is much larger than that (24°) of the standard louver fin. With the increase of the louver angle (for a given louver length), the inter-louver spacing will increase, which will act advantageously to louver bridging. Furthermore, the increased turbulence intensity at the increased louver angle will help to remove the condensate from the gap. In addition, the wavy channels of the corrugated fin samples generally induce Goetler vortices on the concave surfaces (Goldstein and Sparrow, 1976), which may also help to remove the condensate. The difference in $j$ and $f$ factor between the single and the double corrugated louver fin samples may be explained by the difference in fin surface area. As discussed in Kim (2020), samples having larger fin surface area (double corrugated louver fin sample in the present case) yield smaller $j$ and $f$ factor.

Of course, the foregoing argument may depend on the amount of condensate on the fin surface. For the present experiment, the water temperature was held at 6°C and the chamber temperature was maintained at 35°C with 60% relative humidity. If both the air temperature and relative humidity were low and negligible condensate was available on the fin surface, $j$ and $f$ factors under wet condition would more or less follow those under dry condition. On the other hand, under high condensate loading with the louvers blocked by the condensate, totally different results may have been obtained. In Korea, the standard indoor air condition during cooling season is 24°C, 50% relative humidity, where much less condensate will be formed on the fin surface than the experimental condition. The present experimental condition (35°C with 60% relative humidity) was chosen to yield fully wet condition on all the samples.

A performance evaluation criterion proposed by Shah (1978) was used to compare the thermal performance of the enhanced fin samples. In Figure 10, $\eta_o h A_o V$ are drawn as a function of $P/V$, which shows that enhanced fin samples yield larger heat transfer capacity than the plain fin sample at the same pumping power. Furthermore, the largest heat transfer capacity per pumping power is obtained for the double corrugated louver fin sample, followed by the single corrugated and then the standard louver fin sample. Under dry condition (Kim, 2020), standard louver fin sample yielded the largest heat transfer capacity per unit volume. This shows that corrugated louver fins are advantageous over standard louver fin under wet condition.

The literature shows two correlations (Wang et al., 1998; Ma et al., 2007) to predict $j$ and $f$ factors of the standard louver fin-and-tube heat exchangers. The present corrugated louver fin sample data are compared with the predictions of the correlations, and the results are shown in Fig. 11. The figure shows that double corrugated louver fin sample data are predicted excellently. Especially, 92% of $j$ factors and 96% of $f$ factors of the double corrugated louver fin samples are predicted within ±20% by Ma et al. (2007) correlation. As for Wang et al. (1998) correlation, 83% of $j$ factors of the double corrugated louver fin samples are correlated within ±20%. However, predictions are poor for single corrugated louver fin samples. The $j$ factors are overpredicted and $f$ factors are underpredicted. At present, it is too early to draw any conclusion about the correlations due to the limited data and the consideration that the correlations were developed using standard louver fin data.
4. Conclusions

In this study, two kinds of corrugated louver fin-and-tube heat exchangers – one having one corrugation per row and the other having two corrugations per row – were tested under wet condition, and the results were compared with those of the standard louver fin and the plain fin samples. Listed below are major findings.

1) The highest $\eta$ and $f$ factor were obtained for the double corrugated louver fin sample, followed by the single corrugated louver fin sample, the standard louver fin sample and then the plain fin sample. This result is in contradiction with those obtained under dry condition, where the standard louver fin sample yielded the highest $\eta$ and $f$ factor.

2) The high $\eta$ and $f$ factor of the corrugated louver fin samples may be due to the improved condensate drainage over the standard louver fin sample. Corrugated channel may have induced intense flow mixing, and high louver angle of the corrugated louver fin increased the inter-louver spacing, which facilitated the condensate drainage.

3) At $Re_0 = 1000$, the heat transfer enhancement factors ($\eta/\eta_p$) of the single corrugated louver fin samples were 2.34, 1.89, 2.23 and the pressure drop penalty factors ($f/f_p$) were 1.89, 2.06, 2.02 for one row, two row and three row configuration respectively. For double corrugated louver fin sample, those were 2.43, 1.91, 2.15 for $\eta/\eta_p$ and 1.39, 2.46 for $f/f_p$. For the standard louver fin sample, those were 2.15, 1.80, 1.73 for $\eta/\eta_p$ and 1.45, 1.39, 1.52 for $f/f_p$.

4) All the enhanced fin samples yielded larger heat transfer capacity than that of plain fin sample at the same pumping power. Furthermore, the largest heat transfer capacity per pumping power was obtained for the double corrugated louver fin, followed by the single corrugated louver fin and the standard louver fin sample.

References

ASHRAE Standard 41.1, Standard Method for Temperature Measurement, ASHRAE (1986).
ASHRAE Standard 41.2, Standard Method for Laboratory Air-Flow Measurement, ASHRAE (1987).
ASHRAE Standard 41.5, Standard Measurement Guide, Engineering Analysis of Experimental Data, ASHRAE (1975).
ESDU 98005, Design and Performance Evaluation of Heat Exchangers: the Effectiveness and NTU method, Engineering and Sciences Data Unit 98005 with Amendment A, London ESDU International plc. (1998), pp. 122-129.
Goldstein, L. and Sparrow, E. M., Experiments on the transfer characteristics of a corrugated fin and tube heat exchanger configuration, Journal of Heat Transfer, Vol. 98 (1976), pp. 26-34.
Hong, K. and Webb, R. L., Performance of dehumidifying heat exchangers with and without wetting coatings, J. Heat Transfer, Vol. 121 (1999), pp. 1018-1026.
Kuehn, T. H., J. W. Ramsey and J. L. Threlkeld, Thermal Environmental Engineering, 3rd Ed. (1998), Pearson Pub.
Kim, N.-H., An experimental investigation on the airside performance of fin-and-tube heat exchangers having radial slit fins under wet condition, J. Thermal Sci. Tech., Vol. 11, No. 1 (2016), pp. 1-17.

Kim, N.-H., An experimental investigation on the airside performance of fin-and-tube heat exchangers having corrugated louver fins - Part I; dry surface, submitted to J. Thermal Sci. Tech. (2020)

Ma, X., Ding, G., Zhang, Y., and Wang, K., Airside heat transfer and friction characteristics for enhanced fin-and-tube heat exchanger with hydrophilic coating under wet conditions, Int. J. Refrig., Vol. 30 (2007), pp. 1153-1167.

Park, B.-B., You, S.-M., Yoon, B. and Yoo, K.-C., Experimental study of heat transfer and pressure drop characteristics for flow of water inside circular smooth and micro-fin tubes, Korean J. Air Cond. Refrig., Vol. 9, No. 4 (1997), pp. 454-461.

Pirompugd, W., Wang, C.-C. and Wongwises, S., A review on reduction method for heat and mass transfer characteristics of fin-and-tube heat exchangers under dehumidifying conditions, Int. J. Heat Mass Trans., Vol. 52, No. 9-10 (2009), pp. 2370-2378.

Schmidt, T. E., Heat transfer calculations for extended surfaces, J. of ASRE, Refrigeration Engineering, Vol. 4 (1949), pp. 351-357.

Shah, R. K., Compact heat exchanger surface selection methods, Proc. 5th Int. Heat Transfer Conf., Vol. 4 (1978) pp. 193-199.

Wang, C.-C. and Chang, C.-T., Heat and mass transfer for plate fin-and-tube heat exchangers, with and without hydrophilic coating, Int. J. Heat Mass Trans., Vol. 41 (1998), pp. 3109-3120.

Wang, C.-C., Lin, Y.-T. and Lee, C.-J., Heat and momentum transfer for compact fin-and-tube heat exchangers in wet conditions, Int. J. Heat Mass Trans., Vol. 43 (2000), pp. 3443-3452.

Webb, R. L. and Kim, N.-H., Principles of Enhanced Heat Transfer, 2nd Ed. (2005), Taylor and Francis Pub.