EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER CHARACTERISTICS OF LOUVER FIN-AND-TUBE HEAT EXCHANGER WITH 5 MM DIAMETER TUBES UNDER WET CONDITIONS

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ABSTRACT

The heat transfer characteristics of fin-and-tube heat exchangers with 5 mm diameter tubes under wet conditions were investigated by experiments on 11 louver fin-and-tube heat exchangers. The effects of fin pitch, fin size and inlet relative humidity on air-side performance were analyzed. The results indicated that the effect of inlet relative humidity on the heat transfer rate is small, and heat transfer rate decreases with the increase of fin pitch, which is more obvious than that of fin-and-tube heat exchanger with 7 mm or larger diameter tubes. The water bridge was observed at the bottom of fin in experiments; however, it did not occur on the fin with 7 mm or large diameter tubes. A correlation for *j* factor was developed to predict the heat transfer performance of fin-and-tube heat exchanger with 5 mm diameter tubes. The mean deviations of the proposed *j* correlation are within 6.5%.

1. INTRODUCTION

The fin-and-tube heat exchangers with enhanced fin pattern like louver and wave are quite common in room air conditioners. There are several investigations of air-side performance of enhanced fin-and-tube heat exchanger. Mirth and Ramadhyani (1993 and 1994) tested wavy fin-and-tube heat exchanger, and found that Nusselt numbers were very sensitive to the change of inlet air dew point temperature in the dehumidifying process. To obtain the correlation for predicting performance, Wang (2000) reported experimental data and developed correlation which is suitable for predicting performance of fin-and-tube heat exchanger with 10.33 mm diameter tubes under wet conditions. Because tube diameter is gradually reduced to 7 mm, Ma (2009) tested the performance of enhanced fin with 7 mm diameter tubes, and presented correlation of heat, mass transfer and pressure drop for fin-and-tube with 7 mm and 10.33 mm diameter tubes.

The above researches are focused on fin-and-tube heat exchanger with 7 mm or larger diameter tubes. Recently, for the lower material cost and less refrigerant charge, fin-and-tube heat exchangers with smaller diameter tubes (diameter is smaller than or equal to 5 mm) gradually replace those with 7 mm or larger diameter tubes. When tube diameter decreases from 7 mm to 5 mm, the tube cross-sectional area can be reduced by 49%, so the refrigerant charge can be decreased accordingly and the explosion risk of air conditioners using flammable refrigerants (e.g. R290) can be obviously decreased. Moreover, with the decrease of tube diameter, the heat exchange capacity can be enhanced if a constant heat exchanger size is required.

For the fin of heat exchanger, the permitted fin pitch depends on the tube diameter, and the fin size is related to the balance of fin side heat transfer resistance and tube side heat transfer resistance, so the fin configuration for smaller diameter tube is different from that for larger diameter tube, leading to different performance of the heat exchanger. However, no experimental data on the performance of fin-and-tube heat exchangers with smaller diameter tubes are published, and the existent correlations developed based on the experimental data of heat exchangers with 7 mm and larger diameter tubes may not be directly extended to those with small diameter tubes. Thus, it is necessary to carry out experimental research on performance of fin-and-tube heat exchanger with smaller diameter tubes, and develop performance prediction correlation.

In this study, the air-side heat transfer characteristic of fin-and-tube heat exchanger with 5 mm diameter tubes were investigated experimentally. The effects of fin configuration (including fin size, fin pitch, fin pattern, etc) on the heat transfer characteristic were analyzed. Based on the experimental data, new correlations of heat transfer for fin-and-tube heat exchanger with 5 mm diameter tubes were developed, and the correlations can agree with the experiment data well.

2. EXPERIMENTAL APPARATUS

The experimental apparatus is schematically illustrated in Fig. 1, which includes an air flow loop, a water flow loop, a data acquisition system and the test heat exchangers.

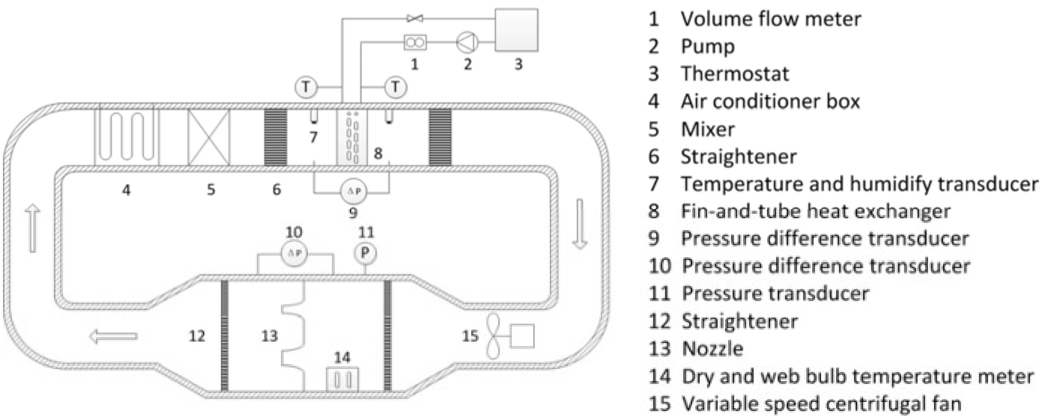

Figure 1. Schematic of experimental system

The air flow loop is a close type wind tunnel. A variable speed centrifugal fan (0.75 kW) is used to circulate the air passing through the nozzle chamber, the air conditioner box, the mixing device, the straightener, and the test heat exchanger orderly. The air flow rate is measured by multiple nozzles based on the ASHRAE 41.2 standard (1987). A differential pressure transmitter (GE Druck, model LPM 9000) with ±5.0 Pa precision is used to measure the air pressure difference across nozzle. A pressure transmitter (GE Druck, model PTX 1400) with ±1.0 kPa precision and a dry bulb and wet bulb temperature transducer (CHINO, model R220-30) with ±0.3^{o}C precision are used to measure the inlet air conditions of nozzles. The air conditioner box is used to control the temperature and humidity of air at test section inlet, which are allowed ±0.2^{o}C and ±3% fluctuation range. The test section is insulated with a 20 mm thick thermal insulation material. A differential pressure transmitter (GE Druck, model LPM 9481) with ±0.2 Pa precision is used to measure the pressure difference across the heat exchangers. The dry bulb temperature and relative humidity of air at inlet and outlet of test section are measured by two temperature and humidity transducers (VAISALA, model HMP 233) with ±0.1^{o}C and ±1.4% precision. Six K-type thermocouples with ±0.1^{o}C precision welded on the tube surface are used to measure the fin base temperature.

The water flow loop consists of thermostat (ADVANTEC, model TBH 127AA), centrifugal pump and magnetic flow meter (TOKYO KEISO, model MGM 1010K) with ±0.15 L/min precision. The heat transfer fluid on the tube side is cold water. The purpose of this loop is to provide the cool capacity of the test heat exchangers. After the water reaches the required temperature, it is pumped out of the thermostat, delivered to the heat exchanger and then returned to the thermostat. The water temperature differences between inlet and outlet of heat exchangers are measured by two K-type thermocouples with a calibrated accuracy of ±0.1°C. All signals are registered by a data acquisition system and finally averaged over the elapsed time.

11 fin-and-tube heat exchangers consist of aluminum fins and copper tubes. The fin configurations are shown in Table 1. The hydrophilic coating on fin surface is organic resin, and its contact angle is 34° which is measured by contact angle measurement. Condensation phenomena on the fin surface are recorded by a video camera located at the outlet of test section. Total 14 test conditions are listed in Table 2.

Table 1. Geometric dimension of the tested fin-and-tube heat exchangers

No.	δ (mm)	D_c (mm)	P_t (mm)	P_l (mm)	F_p (mm)	Row number	Fin pattern and details
1	0.095	5.2	19	11	1.2	1	3 louvers
2	0.095	5.2	19	11	1.4	1	3 louvers
3	0.095	5.2	19	13.6	1.1	1	3 louvers
4	0.095	5.2	19	13.6	1.2	1	3 louvers
5	0.095	5.2	19	13.6	1.4	1	3 louvers
6	0.095	5.2	18	13.8	1.2	1	3 louvers
7	0.095	5.2	18	13.8	1.3	1	3 louvers
8	0.095	5.2	18	13.8	1.4	1	3 louvers
9	0.095	5.2	18	14.7	1.2	2	3 louvers
10	0.095	5.2	18	14.7	1.3	2	3 louvers
11	0.095	5.2	18	14.7	1.4	2	3 louvers

Table 2. The test conditions

No.	$T_{a,in}$ (°C)	$T_{w,in}$ (°C)	V (m/s)	RH_{in} (%)
1	27	10	0.5	40
2	27	10	1	40
3	27	10	1.5	40
4	27	10	0.5	50
5	27	10	0.8	50
6	27	10	1	50
7	27	10	1.2	50
8	27	10	1.5	50
9	27	10	0.5	65
10	27	10	1	65
11	27	10	1.5	65
12	27	10	0.5	80
13	27	10	1	80
14	27	10	1.5	80

3. DATA RECUDTION

The reduction process is based on the Threlkeld (1970) method which is an enthalpy-based reduction method. The detailed reduction procedures are described as follows:

Heat transfer rate (Q_{ave}) on the airside can be calculated as:

$$Q_a = m_a\left(i_{a,in} - i_{a,out}\right) \tag{1}$$

$$Q_w = m_w c_{p,w}\left(T_{w,out} - T_{w,in}\right) \tag{2}$$

$$Q_{ave} = \frac{Q_A + Q_w}{2} \tag{3}$$

In the experiments, only those data that satisfy the ASHRAE 33-78 (2000) requirements (the energy balance conditions, $|Q_w - Q_a|/Q_{ave} \leq 0.05$) are considered in the final analysis.

Heat transfer coefficient in tube can be calculated as:

$$h_i = \left(\frac{k_i}{D_i}\right)\frac{\left(\mathrm{Re}_{Di} - 1000\right)\mathrm{Pr}\left(f_i/2\right)}{1 + 12.7\sqrt{f_i/2}\left(\mathrm{Pr}^{2/3} - 1\right)} \tag{4}$$

where

$$f_i = \left[1.58\ln\left(\mathrm{Re}_{Di}\right) - 3.28\right]^{-2} \tag{5}$$

$i_{a,in}$ and $i_{a,out}$ can be calculated as:

$$i_{a,m} = i_{a,in} + \frac{i_{a,in} - i_{a,out}}{\ln\left(\dfrac{i_{a,in} - i_{r,out}}{i_{a,out} - i_{r,in}}\right)} - \frac{\left(i_{a,in} - i_{a,out}\right)\left(i_{a,in} - i_{r,out}\right)}{\left(i_{a,in} - i_{r,out}\right) - \left(i_{a,out} - i_{r,in}\right)} \tag{6}$$

$$i_{r,m} = i_{r,out} + \frac{i_{r,out} - i_{r,in}}{\ln\left(\dfrac{i_{a,in} - i_{r,out}}{i_{a,out} - i_{r,in}}\right)} - \frac{\left(i_{r,out} - i_{r,in}\right)\left(i_{a,in} - i_{r,out}\right)}{\left(i_{a,in} - i_{r,out}\right) - \left(i_{a,out} - i_{r,in}\right)} \tag{7}$$

$T_{p,i,m}$ and $T_{p,o,m}$ are presented as:

$$T_{p,i,m} = \frac{Q_{ave}}{h_i A_{p,i}} + T_{r,m} \tag{8}$$

$$T_{p,o,m} = \frac{Q_{ave} \cdot x_p}{k_p A_{p,m}} + T_{p,i,m} \tag{9}$$

$$A_{p,m} = \frac{A_{p,o} - A_{p,i}}{\ln\dfrac{A_{p,o}}{A_{p,i}}} \tag{10}$$

To calculate fin efficiency, R and h_s should be assumed firstly. R represents the ratio of sensible heat transfer characteristics to mass transfer performance, and h_s represents sensible heat transfer coefficient on the airside.

For full wet conditions, fin efficiency can be calculated by the following equation:

$$\eta_{f,h,full,wet} = \frac{2r_i M^*\left(T_{fb} - T_a^*\right)}{M_{fb}^2\left(r_0^2 - r_i^2\right)\left(T_a - T_{fb}\right)} \cdot \left[\frac{K_1\left(M^* r_o\right)I_1\left(M^* r_i\right) - I_1\left(M^* r_o\right)K_1\left(M^* r_i\right)}{K_1\left(M^* r_0\right)I_0\left(M^* r_i\right) + K_0\left(M^* r_i\right)I_1\left(M^* r_0\right)}\right] \tag{11}$$

$$M_{fb}^2 = \frac{2h_s}{k_f \delta_f}\left(1 + \beta \frac{W_a - W_{fb}}{T_a - T_{fb}}\right) \tag{12}$$

For partially wet condition, ξ refers to the boundary line between dry region and wet region, and can be calculated by Eqs. (13) and (14):

$$\frac{d^2 T_f}{dr^2} + \frac{1}{r}\cdot\frac{dT_f}{dr} + M^{*2}\left(T_a^* - T_f\right) = 0 \qquad \text{for wet region} \tag{13}$$

$$\frac{d^2 T_f}{dr^2} + \frac{1}{r}\cdot\frac{dT_f}{dr} + M^{*2}\left(T_a - T_f\right) = 0 \qquad \text{for dry region} \tag{14}$$

Then, partially wet fin efficiency can be calculated by following equation:

$$\eta_{f,h,partically,wet} = \frac{2r_i M^*}{M_{fb}^2\left(r_0^2 - r_i^2\right)\left(T_a - T_{fb}\right)} \cdot \left\{ \begin{array}{l} \left[\dfrac{I_1\left(M^* r_i\right)K_0\left(M^*\xi\right) + I_0\left(M^*\xi\right)K_1\left(M^* r_i\right)}{I_0\left(M^* r_i\right)K_0\left(M^*\xi\right) - I_0\left(M^*\xi\right)K_0\left(M^* r_i\right)}\right]\left(T_{fb} - T_a^*\right) \\ -\left[\dfrac{I_0\left(M^* r_i\right)K_1\left(M^* r_i\right) + I_1\left(M^* r_i\right)K_0\left(M^* r_i\right)}{I_0\left(M^* r_i\right)K_0\left(M^*\xi\right) - I_0\left(M^*\xi\right)K_0\left(M^* r_i\right)}\right]\left(T_{dew} - T_a^*\right) \end{array} \right\} \tag{15}$$

The enthalpy-temperature ratio of inner tube wall and cooling water (b'_r) can be calculated as:

$$b'_r = \frac{i_{s,p,i,m} - i_{r,m}}{T_{p,i,m} - T_{r,m}} \tag{16}$$

The enthalpy-temperature ratio of tube wall (b'_p) can be calculated as:

$$b'_p = \frac{i_{s,p,p,m} - i_{s,p,i,m}}{T_{p,o,m} - T_{p,i,m}} \tag{17}$$

The total heat transfer coefficient of heat exchanger is presented as:

$$U_0 = \frac{Q_{ave}}{F \Delta i_m A_0} \tag{18}$$

The sensible heat transfer coefficient of heat exchanger can be calculated as:

$$h_i = \left(\frac{k_i}{D_i}\right)\frac{\left(\mathrm{Re}_{Di} - 1000\right)\mathrm{Pr}\left(f_i/2\right)}{1 + 12.7\sqrt{f_i/2}\left(\mathrm{Pr}^{2/3} - 1\right)} \tag{19}$$

where $b'_{w,p}$ is air enthalpy-temperature ratio when temperature is T_{pom}, and $b'_{w,m}$ is air enthalpy-temperature ratio when temperature is $T_{w,m}$;

For full wet condition, R can be calculated as:

$$\frac{\left(i_{a,in}-i_{a,out}\right)}{\left(W_{a,in}-W_{a,out}\right)}=R\frac{\left(i_{a,m}-i_{s,p,o,m}\right)+\left(\varepsilon-1\right)\left(i_{a,m}-i_{s,w,m}\right)}{\left(W_{a,m}-W_{s,p,o,m}\right)+\left(\varepsilon-1\right)\left(W_{a,m}-W_{s,w,m}\right)} \tag{20}$$

$$\varepsilon=\frac{A_0}{A_{p,o}} \tag{21}$$

For partially wet condition, R can be calculated as:

$$\frac{\left(i_{a,in}-i_{a,out}\right)}{\left(W_{a,in}-W_{a,out}\right)}=R\frac{b'_{w,m}}{b'_{w,p}}\frac{\left(A_{p,o}+\eta_{f,wet}A_f\right)\left(i_{a,m}-i_{s,p,o,m}\right)}{A_{p,o}\left(W_{a,m}-W_{s,p,o,m}\right)+A_{f,wet}\left(W_{a,m}-W_{s,w,m}\right)} \tag{22}$$

j factor is presented as:

$$j=\frac{h_s}{G_c c_{p,a}}\mathrm{Pr}^{2/3} \tag{23}$$

4. RESULTS AND DISCUSSION

Fig. 2 shows the effect of inlet relative humidity on Colburn j factors which indicates the airside heat exchanger performance of fin-and-tube heat exchanger with 5 mm diameter tubes. As shown in Fig. 2, the Colburn j factors increase slightly with increase of inlet relative humidity. In Fig. 3, some louvers on the fin surface are blocked by the flow of water film, which eliminates the heat transfer enhancement of water film flow caused by higher relative humidity. Thus, the Colburn j factors are insensitive to inlet relative humidity.

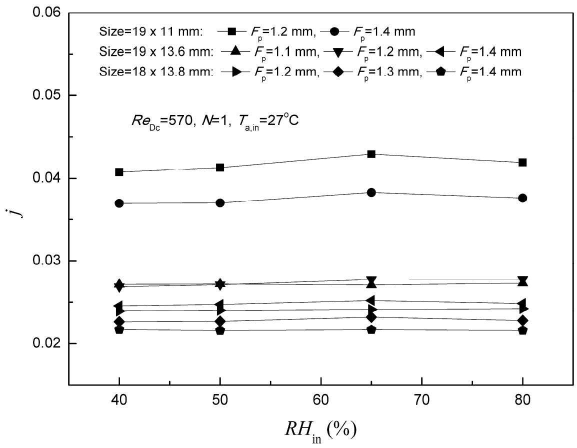

Figure 2. Variations of Colburn j factor as function of inlet relative humidity (RH)

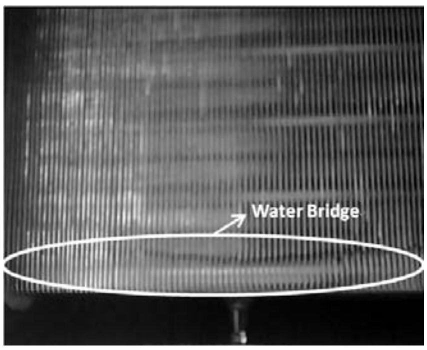

Figure 3. The location of water bridge on fin for 5 mm diameter tubes

Fig. 4 depicts the effect of fin pitch on air-side heat transfer performance of heat exchanger with 5 mm diameter tubes. The Colburn j factors decrease with the increases of fin pitch. Partially wet conditions will happen as the Reynolds number increases. Thus, the effect of fin pitch on Colburn j factors is more obvious at high Reynolds number. These results are similar to those reported by Ma et al. (2007 and 2009).

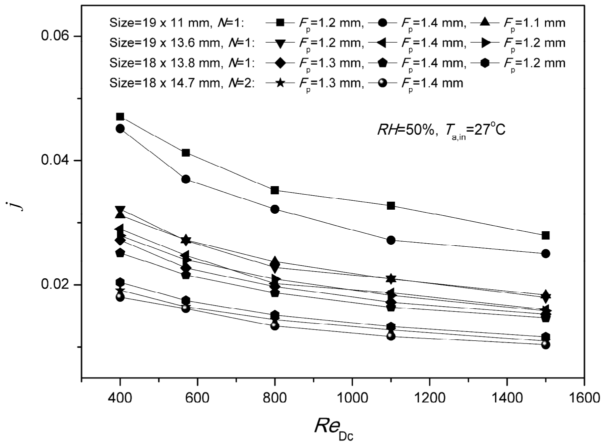

Figure 4. Variations of Colburn j factor as function of air Reynolds number

Moreover, the water bridge occurs at the bottom of fin with hydrophilic coating as Fig. 3 shown. But the water bridge did not occur in fin-and-tube heat exchanger with 7 mm or 10.33 mm diameter tubes in previous studies (Ma *et al*., 2007 and 2009). The occurrence of water bridge may due to the smaller fin size and fin pitch. When fin size decreases, the maximum coverage area of film decreases accordingly which leads to the height of condensate film increases. Then, the smaller fin pitch allows two higher condensate films at adjacent fin surface to combine to a water bridge. However, because of the small coverage area and the bottom location of water bridge, the effects of relative humidity and fin pitch on heat transfer performance are similar to previous research (Ma *et al*., 2007 and 2009) in which no water bridge occurring on fin surface.

5. CORRELATION

The multiple linear regression technique in a practical range of experimental data ($350 < Re_{Dc} < 4500$) is carried out, and the suitable correlation of j is given as follows:

$$j = 0.0899\,\mathrm{Re}_{Dc}^{-0.4228}\left(\frac{P_t}{P_l}\right)^{1.9513}\left(\frac{F_p}{D_c}\right)^{-0.5677} N^{-0.2712} \tag{24}$$

Range for applicability for Eq. (24) is given as follows: D_c=5.2 mm, P_t=18-19 mm, P_l=11-14.7 mm, F_p=1.1-1.4 mm, N=1-2, Re_{Dc}=350-4500.

The proposed heat transfer j factor correlation, Eq. (24), can describe 85.7% of the test data within the deviation of ±15%. The proposed correlation of j has a mean deviation of 6.5%.

6. CONCLUSION

- Colburn j factors decrease with the increase of the fin pitch. This phenomenon is more obvious than that of fin-and-tube heat exchanger with 7 mm or larger diameter tubes;
- Colburn j factors are relatively insensitive to the change of inlet relative humidity;

- The water bridge occurs at the bottom of fin surface for 5 mm diameter tubes, while it was not observed on fin surface for 7 mm or large diameter tubes in previous;
- Colburn *j* factor correlation is proposed to describe heat transfer performance of fin-and-tube heat exchanger with 5 mm diameter tubes. The mean deviations of the proposed j correlation are 6.5%.

7. NOMENCLATURE

A	heat transfer area	(m^2)
D	diameter	(m)
F_p	fin pitch	(m)
G_c	mass flus	(kg/m^2s)
h	heat transfer coefficient	(W/m^2K)
i	ehthalpy	($kJ\ kg^{-1}$)
I_0	Bessel function solution of the second kind, order 0	
I_1	Bessel function solution of the second kind, order 1	
j	Colburn heat transfer factor	
K_0	Bessel function solution of the first kind, order 0	
K_1	Bessel function solution of the first kind, order 1	
P_t	transverse tube pitch	(m)
P_l	longitudinal tube pitch	(m)
δ	fin thickness	(m)

Subscripts

a	air
f	fin
fb	fin base
ft	fin tip
i	inner
o	outer
r	refrigerant
s	saturate
w	water

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