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On the heat and mass analogy of fin-and-tube heat exchanger

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Abstract

This study examines the heat and mass analogy of the fin-and-tube heat exchanger under dehumidifying process. A total of 36 fin-and-tube heat exchangers having plain fin geometry are experimentally examined. It is found that the ratio of $h_{c,o}/h_{d,o}C_{p,a}$ is in the range of 0.6–1.1 and is insensitive to change of fin spacing at low Reynolds number. However, it is noted that this ratio is not a constant throughout the test range. A slight drop of the ratio of $h_{c,o}/h_{d,o}C_{p,a}$ is seen with the decrease of fin spacing and with the rise of the Reynolds number. This is associated with the more pronounced influence during condensate removal. Moreover, during the dehumidifying process, the temperature gradient is directly responsible for establishing the concentration gradient, suggesting the heat transfer and mass transfer are not independent. Based on a simple analysis, one can easily find that the increasing rate of $\left(\frac{dc}{dT}\right)\big|_i$ slightly exceeds that of $\frac{\Delta c}{\Delta T}$. As a result, the ratio of $h_{c,o}/h_{d,o}C_{p,a}$ can be proved to be slightly decreased with the rise of the Reynolds number.

Keywords: Fin-and-tube heat exchanger; Heat and mass analogy; Dehumidifying

1. Introduction

Plate fin-and-tube heat exchangers are composed of a plurality of heat transfer tubes that are inserted into respective bores of fins, and are closely fitted and fixed to the latter by means of tube expansion or the like method. The fin-and-tube heat exchangers are employed in a wide variety of engineering applications like air-conditioning apparatus, process gas heater, and cooler. For typical applications of exploitation the fin-and-tube heat exchangers, the air-side resistance generally comprises over 90% of the total thermal resistance. Hence enhanced fin patterns such as wavy, louver, slit, and convex-louver are adopted for augmentation. However, plain fin geometry is still the mostly commonly used for its reliability when comparing to other enhanced fin patterns.

The fin-and-tube heat exchangers can be applicable to both condenser and evaporators. In the evaporators which typically operated at a surface temperature below the dew point temperature. Hence, simultaneous heat and mass transfer occurs along the fin surfaces. In general, the complexity of the moist air flow pattern across the fin-and-tube heat exchangers under dehumidifying conditions makes the theoretical simulations very difficult. Accordingly, it is necessary to resort to experimentation.

Many experimental studies have been carried out to study the heat and mass transfer characteristics of the fin-and-tube heat exchangers under dehumidifying conditions. For instance, McQuiston [1,2] presented experimental data for five plate fin-and-tube heat exchangers, and developed a well-known heat transfer and friction correlation applicable to dry and wet surfaces. Mirth and Ramadhyani [3,4] investigated the heat and mass characteristics of wavy fin heat exchangers. Their results showed that the Nusselt numbers were very sensitive to change of inlet dew point temperatures, and the Nusselt number decreases with an increase of dew point temperatures. Similar results were reported by Fu et al. [5] in dehumidifying heat exchangers having a louver fin configuration. They reported a pronounced decrease of the wet sensible heat transfer coefficients with increases of inlet relative humidity. On the contrary, the experimental data of Seshimo

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Nomenclature

c	concentration	T	temperature
D	diffusivity	c_∞	ambient concentration
D_i	inside diameter of tube	$C_{p,a}$	heat capacity
F_p	fin pitch	D_c	tube collar diameter
$h_{c,o}$	sensible heat transfer coefficient	k	thermal conductivity
$h_{d,o}$	mass transfer coefficient (based on W)	k_m	mass transfer coefficient (based on c)
N	number of tube row	L	characteristics length
P_l	longitudinal tube pitch	Nu	Nusselt number
Pr	Prandtl number	P_t	transverse tube pitch
Q_a	heat transfer rate at the air-side	Re_{D_c}	Reynolds number based on D_c
Q_w	heat transfer rate of the water side	t	fin thickness
Re_{D_i}	Reynolds number based on D_i	T_∞	ambient temperature
Sc	Schmidt number	W	humidity ratio
Sh	Sherwood number		

et al. [6] indicated that the Nusselt number was relatively independent of inlet conditions. Wang et al. [7] studied the effect of the fin pitch, the number of tube row, and inlet relative humidity on the heat transfer performance under dehumidification, and concluded that the sensible heat transfer performance is relatively independent of inlet humidity. The difference in the existing literatures is attributed to the different reduction methodology as shown by Pirompugd et al. [8].

Even though many efforts have been devoted to the study of the wet-coils, the available literature on the dehumidifying heat exchangers still offers limited information especially in association with mass transfer. This can be made clear from the reported data were mainly focused on the study of the sensible heat transfer characteristics, very few attention was paid to the mass transfer characteristics. Most of the investigators simply used heat-mass analogy to estimate the corresponding mass transfer coefficient from available heat transfer coefficient. Therefore, the objective of the present study is to examine the applicability of the heat and mass analogy based on experimental data.

2. Experimental apparatus and reduction method

The schematic diagram of the experimental air circuit assembly and related data reduction method had been addressed in some previous studies such as those by Wang et al. [7], Pirompugd et al. [8,9]. Readers who are interested in this may be referred to these studies. Details of the test samples can be seen from Table 1. Notice that no hydrophilic treatment was made to all the test heat exchangers.

Table 1
Geometric dimension of the sample plain fin-and-tube heat exchangers

Nos.	F_p (mm)	t (mm)	D_c (mm)	P_t (mm)	P_l (mm)	N
1	1.19	0.115	8.51	25.4	19.1	1
2	1.75	0.120	10.34	25.4	22.0	1
3	2.04	0.115	8.51	25.4	19.1	1
4	2.23	0.115	10.23	25.4	19.1	1
5	2.50	0.120	10.34	25.4	22.0	1
6	1.20	0.115	6.93	17.7	13.6	1
7	1.21	0.115	6.93	17.7	13.6	1
8	1.98	0.115	6.93	17.7	13.6	1
9	1.99	0.115	6.93	17.7	13.6	1
10	1.23	0.115	8.51	25.4	19.1	2
11	1.70	0.120	8.62	25.4	19.1	2
12	2.06	0.115	8.51	25.4	19.1	2
13	2.24	0.130	10.23	25.4	22.0	2
14	3.20	0.130	10.23	25.4	22.0	2
15	1.22	0.115	7.53	21.0	12.7	2
16	1.22	0.115	7.53	21.0	12.7	2
17	1.23	0.115	10.23	25.4	19.1	2
18	1.78	0.115	7.53	21.0	12.7	2
19	1.79	0.115	7.53	21.0	12.7	2
20	1.82	0.130	10.23	25.4	22.0	2
21	3.13	0.120	8.62	25.4	19.1	2
22	1.23	0.115	10.23	25.4	19.1	4
23	1.55	0.115	10.23	25.4	19.1	4
24	2.03	0.130	10.23	25.4	22.0	4
25	2.23	0.130	10.23	25.4	22.0	4
26	3.00	0.130	10.23	25.4	22.0	4
27	1.21	0.115	8.51	25.4	19.1	4
28	1.22	0.115	7.53	21.0	12.7	4
29	1.60	0.115	8.51	25.4	19.1	4
30	1.70	0.120	8.62	25.4	19.1	4
31	1.78	0.115	7.53	21.0	12.7	4
32	2.31	0.115	10.23	25.4	19.1	4
33	3.13	0.120	8.62	25.4	19.1	4
34	1.85	0.130	10.23	25.4	22.0	6
35	2.21	0.130	10.23	25.4	22.0	6
36	3.16	0.130	10.23	25.4	22.0	6

3. Results and discussion

The dehumidifying process involves heat and mass transfer simultaneously, if mass transfer data are unavailable, it is convenient to employ the analogy between heat and mass transfer. The existence of the heat and mass analogy is because the fact that conduction and diffusion in a liquid are governed by physical laws of identical mathematical form. Therefore, for air–water vapor mixture, the ratio of $h_{c,o}/h_{d,o}C_{p,a}$ is generally around unity, i.e.,

$$\frac{h_{c,o}}{h_{d,o}C_{p,a}} \approx 1 \tag{1}$$

The term in Eq. (1) approximately equals to unity for dilute mixtures like water vapor in air near the atmospheric pressure (temperature well-below corresponding boiling point). The validity of Eq. (1) relies heavily on the mass transfer rate. The experimental data of Hong and Webb [10] indicated that this value is between 0.7 and 1.1, Seshimo et al. [6] gave a value of 1.1. Eckels and Rabas [11] also reported a similar value of 1.1–1.2 for their test results of fin-and-tube heat exchangers having plain fin geometry. The aforementioned studies all showed the applicability of Eq. (1). In the present study, we notice that the values of $h_{c,o}/h_{d,o}C_{p,a}$ were generally between 0.6 and 1.1. Test results of this ratio for fully wet condition are shown in Fig. 1. As seen in the figure, the influence of tube row or geometry on this ratio is rather unobvious. However, we found that the ratios are quite different between the original Threlkeld method and the present modified method. The reduced results of this ratio are generally larger (0.3–1.4).

There are two major differences between the reduced results from the original Threlkeld method [12] and from the present tube-by-tube approach. Firstly, larger deviation of the original Threlkeled's methods is encountered. This is related to the considerable influence of inlet humidity of the original Threlkeld's method upon which counter-cross flow configuration is based. For the present reduction method, the ratio is insensitive to change of inlet humidity provided that the surface is fully wet. Secondly, reduction by the present method indicates that the ratio of $h_{c,o}/h_{d,o}C_{p,a}$ slightly decreases with the Reynolds number whereas the original Threlkeld method shows the opposite trend (slightly increase with the Reynolds number). Notice that with the rise of inlet flow inertia, the condensate can be easily removed to provide more room for further condensation, thereby giving rise to a higher mass transfer performance. The condensate removal becomes even pronounced with smaller fin spacing. As mentioned by Pirompugd et al. [8], the condensate retention becomes more and more severe when the fin spacing is reduced. In that regard, the removal of condensate subject to larger flow inertia helps to improve the mass transfer performance considerably provided that the condensate retention phenomenon is eliminated. Therefore, one can see that the ratio of $h_{c,o}/h_{d,o}C_{p,a}$ is slightly decreased with the fin spacing and with the rise of Reynolds number. Notice that the effect of fin spacing on the ratio of $h_{c,o}/h_{d,o}C_{p,a}$ is only effective when Reynolds number is sufficient high. One of the explanations of this departure is because the high air flow rate increases the vapor shear to wipe out the condensate. Conversely, the effect of fin spacing on the ratio of $h_{c,o}/h_{d,o}C_{p,a}$ is comparatively small at low Reynolds number when the condensate retention is comparatively severe.

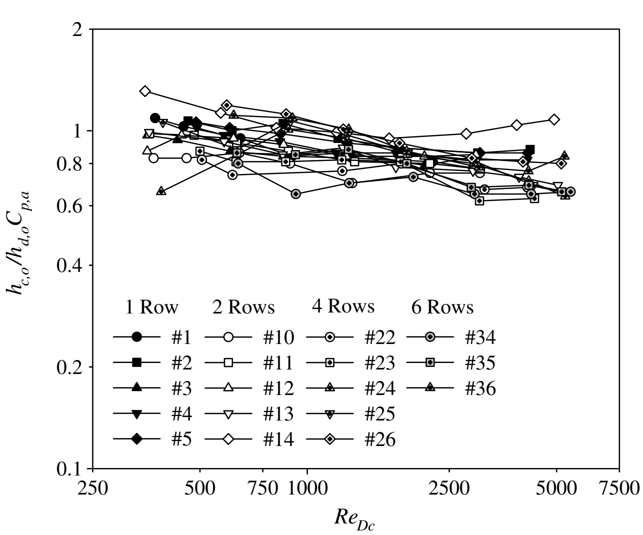

Fig. 1. Ratio of $h_{c,o}/h_{d,o}C_{p,a}$ vs. Reynold number.

In addition to the foregoing qualitative description about the slight decrease of the ratio of $h_{c,o}/h_{d,o}C_{p,a}$ vs. *Re*, a more rigorous physical explanation of this phenomenon is described in the following. The physical interpretation is originally proposed by Venkatesan and Fogler [13] who commented about the analogy in the wax deposition on a cold wall. The present study extends their viewpoint to interpret the test results of fin-and-tube heat exchanger under dehumidification.

As is well-known that the original heat-mass transfer analogy is valid only when the temperature and concentration fields are independent with each other. During the dehumidifying process, the temperature gradient is directly responsible for establishing the concentration gradient. Thus, these fields are not truly independent. This is especially true for turbulent flow conditions where the concentration boundary layer thickness is not independent of the temperature boundary layer thickness. As a result, the heat-mass transfer analogy described earlier is not expected to hold. For heat and mass transfer at the interface, the following equations hold:

$$h_{c,o}\Delta T = -k\left.\frac{dT}{dy}\right|_i \tag{2}$$

$$k_m\Delta c = -D\left.\frac{dc}{dy}\right|_i \tag{3}$$

In the foregoing equations, ΔT and Δc are the temperature difference and concentration difference between the ambient and the interface, respectively. The subscript i denotes at the interface. Adopting the definitions of Nusselt number and Sherwood number, i.e. $Nu = hL/k$ and $Sh = k_mL/D$ (L represents certain characteristics length), one can arrive the following equations from Eqs. (2) and (3):

$$Nu = \frac{L}{\Delta T}\left(-\frac{\mathrm{d}T}{\mathrm{d}y}\bigg|_i\right) \tag{4}$$

$$Sh = \frac{L}{\Delta c}\left(-\frac{\mathrm{d}c}{\mathrm{d}y}\bigg|_i\right) \tag{5}$$

Dividing Eq. (4) with Eq. (5) yields:

$$\frac{Nu}{Sh} = \left(\frac{-\frac{\mathrm{d}T}{\mathrm{d}y}\Big|_i}{-\frac{\mathrm{d}c}{\mathrm{d}y}\Big|_i}\right)\frac{\Delta c}{\Delta T} \tag{6}$$

For the case of independent heat and mass transfer, the temperature gradient and concentration gradient do not interact with each other, the above equation is often correlated in the famous form as:

$$\frac{Nu}{Sh} = \left(\frac{Pr}{Sc}\right)^{1/3} \tag{7}$$

For dehumidification process, the interface concentration is regarded as saturated and is related to the interface temperature. Hence $c_i = c_i(T)$, Eq. (6) can be rewritten as:

$$\frac{Nu}{Sh} = \left(\frac{\mathrm{d}T}{\mathrm{d}c}\right)\bigg|_i \frac{\Delta c}{\Delta T} = \frac{\frac{\Delta c}{\Delta T}}{\left(\frac{\mathrm{d}c}{\mathrm{d}T}\right)\big|_i} \tag{8}$$

Based on the derived Eq. (8), let's consider a general case of dehumidification. A schematic diagram of concentration vs. temperature is shown in Fig. 2. Note that the humidity ratio W is usually used as the driving potential of mass transfer for dehumidification instead of concentration. The shape of the relation curve of the humidity vs. temperature is "concave upwards" as shown in Fig. 2. In that regard, the interfacial temperature rises when the inlet frontal velocity is increased. This is applicable even when the tube side temperature is fixed (evaporation inside tube) due to increased heat transfer rate with the rise of frontal velocity. In that regard, one can see the value of $\left(\frac{\mathrm{d}c}{\mathrm{d}T}\right)\big|_i$ increases when the frontal velocity is increased. In the meantime, the value of $\frac{\Delta c}{\Delta T}$ is also increased. However, one can easily find out that the increasing rate of $\left(\frac{\mathrm{d}c}{\mathrm{d}T}\right)\big|_i$ slightly exceeds that of $\frac{\Delta c}{\Delta T}$. This suggests the difference between denominator and numerator of Eq. (8) is being increased when the frontal velocity is increased, thereby showing a slight decrease of the $h_{c,o}/h_{d,o}C_{p,a}$ ratio with the rise of frontal velocity.

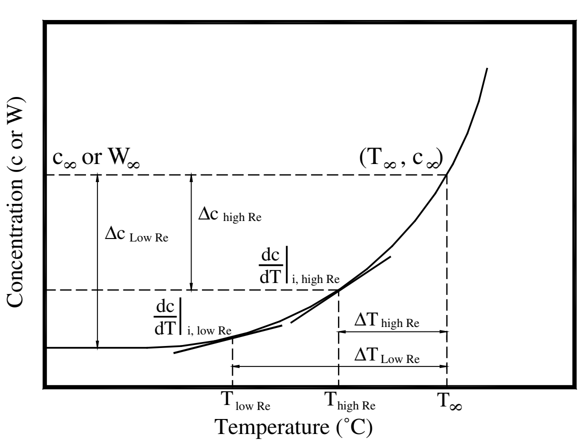

Fig. 2. Relation between concentration and temperature.

4. Conclusion

The dehumidifying process involves heat and mass transfer simultaneously, if mass transfer data are unavailable, it is convenient to employ the analogy between heat and mass transfer. For air–water vapor mixture, the ratio of $h_{c,o}/h_{d,o}C_{p,a}$ is generally around unity. Applicability of the heat and mass analogy in association with the fin-and-tube heat exchangers is examined in the present study. A total of 36 fin-and-tube heat exchangers having plain fin geometry is tested in a controlled environment. It is found that the ratio of $h_{c,o}/h_{d,o}C_{p,a}$ is in the range of 0.6–1.1 and is insensitive to change of fin spacing at low Reynolds number. However, the ratio is not a constant throughout the test range. A slight drop of the ratio of $h_{c,o}/h_{d,o}C_{p,a}$ is seen with the decrease of fin spacing and with the rise of the Reynolds number. This is associated with the more pronounced influence caused by condensate removal. Moreover, during the dehumidifying process, the temperature gradient is directly responsible for establishing the concentration gradient. The heat transfer and mass transfer are not independent. Based on a simple analysis, one can easily find that the increasing rate of $\left(\frac{\mathrm{d}c}{\mathrm{d}T}\right)\big|_i$ slightly exceeds that of $\frac{\Delta c}{\Delta T}$. As a result, the ratio of $h_{c,o}/h_{d,o}C_{p,a}$ can be proved to be slightly decreased with the rise of Reynolds number.

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