Heat Transfer Characteristics of Individual Row of Fin and Tube Heat Exchangers

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ABSTRACT: Heat transfer performances of individual row of two-row fin and tube heat exchangers are experimentally investigated. Tested are four heat exchangers which are geometrically identical with the exception of fin shape, slit or louver, and that the fins between the first row and the second row are connected or separated. The tube diameter and fin spacing of the heat exchangers examined are 7 mm and 1.4 mm, respectively. All thermal fluid measurements are made using a psychrometric calorimeter. In order to evaluate air-side heat transfer coefficients of individual rows, tube-side water flow rates of individual rows are independently controlled such that the water-side temperature drops in each row remain at 5℃. Frontal air velocity varies in the range from 0.7 m/s to 2.5 m/s. Heat transfer coefficients are presented in terms of Colburn j-factor. The results show that the heat transfer coefficient of the upstream row is larger than that for the downstream row at low Reynolds numbers.

Nomenclature

A : heat transfer area [m^2]
A_{free} : minimum flow area of air [m^2]
C_P : constant pressure specific heat [J/kg℃]
d_h : hydraulic diameter [m]
$G_{\max}$: mass flux of air flowing through the minimum flow area [kg/m^{2}s]
h : heat transfer coefficient [W/m^2℃]
j : Colburn j-factor
k : thermal conductivity [W/m℃]
L : streamwise length of a heat exchanger [m]
$\dot{m}$: mass flow rate [kg/s]
Pr : Prandtl number
Q : heat transfer rate [kcal/hr]
Re : Reynolds number
t : thickness [m]
T : temperature [℃]
U : overall heat transfer coefficient [W/m^2℃]

Greek symbols

Δd_o : increase in diameter after tube expansion [m]
η : overall surface efficiency
η_f : fin efficiency
μ : viscosity [kg/ms]
ρ : density [kg/m^3]

Superscripts

a : air
c : contact between fin collar and tube
f : fin
i : inlet

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o : outlet
w : water

1. Introduction

Fin and tube heat exchangers are widely used in modern air-conditioners as well as in process plants. Improvements of heat transfer performance of heat exchangers have been of importance in view of energy economy and environmental protection. During last several decades, attention has been paid to enhancement of air-side heat transfer performance because the air-side thermal resistance accounts for about 70 percent of the total thermal resistance. The air-side heat transfer coefficient is influenced by many parameters such as fin shape, row spacing, tube diameter, fin pitch and number of rows. Among them, the effect of number of rows is closely related to the heat exchanger downsizing. The number of rows of evaporators and condensers in domestic air-conditioners has been reduced to three or two from four or more. This reduction was possible due to the fact that most of heat transfer occurs via the first row while manufacturing cost of each row is the same. Accurate prediction of heat transfer performance of each row is necessary in order to design heat exchangers with reduced rows.

The effects of the aforementioned parameters on air-side heat transfer have been investigated by many researchers. Due to various and intensive investigations, there have been significant improvements in air-side heat transfer performance. On the other hand, relatively less attention has been paid to variation of air-side heat transfer coefficient of individual row. As the air-side heat transfer enhancement technology saturates, fin-tube heat exchanger designers get interested in how to make the contribution of rear row(s) become larger. In this regard, variation of air-side thermal resistance in each row and the effect of conduction through fins are investigated in the present work.

Information on row-by-row variation in heat transfer coefficient as well as the effect of the number of tube rows on the average air-side heat transfer coefficient in fin-tube heat exchangers are not abundant. McAdams[1] presented a table showing that the row-to-row variation in heat transfer coefficient for staggered banks of unfinned tubes. This table shows that the heat transfer coefficient increases from the minimum for the first row and asymptotically approaches a maximum value as the number of row increases. Rich[2] made measurements of air-side heat transfer coefficients for the individual rows of a 4-row plate fin and tube heat exchanger. He reported that the heat transfer coefficient of the downstream rows is higher than that for the upstream rows at high Reynolds numbers while the heat transfer coefficient of the upstream rows is higher at low Reynolds numbers. In the present work, four heat exchanger specimens are tested, which are geometrically identical with the exception of fin shape and whether the 1st and 2nd rows are connected or separated. Heat transfer performance of individual rows is examined in view of conduction through fins.

2. Experimental apparatus and methods

Heat transfer rate and coefficient of individual rows in four heat exchanger specimens are measured using a psychrometric calorimeter. This calorimeter is designed based on air-enthalpy method described in ASHRAE handbook[3] and its schematic is shown in Fig. 1. The water bath located outside the air conditioned chambers provides the heat exchangers with controlled water. The temperature of water at various locations are measured with RTDs with the accuracy of ±0.1℃. All signals out of sensors and code testers are collected by a console and processed for reduced values. Figure 2

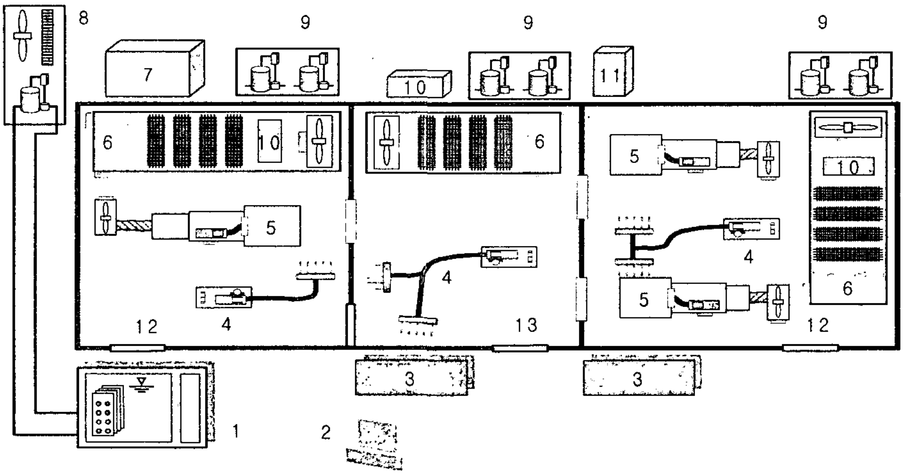


Fig. 1 Schematic diagram of multi-calorimeter.

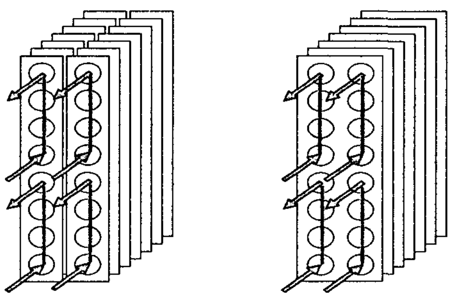


Fig. 2 Schematic of heat exchanger configuration.

Table 1 Specifications of heat exchanger samples (Unit: m)

Notation	L1	L2	S1	S2
Fin type	Louver	Louver	Slit	Slit
Fin pitch	0.0014	0.0014	0.0014	0.0014
Step pitch	0.021	0.021	0.021	0.021
Row pitch	0.0125	0.0125	0.0125	0.0125
Tube diam.	0.00735	0.00735	0.00735	0.00735
No of rows	2	2	2	2
Shape	Separated	Connected	Separated	Connected

Table 2 Experimental condition

Air inlet		Water inlet	
Temp.	Frontal velocity	Temp.	Flow rate
35℃ (DB) 24℃ (WB)	0.7~2.5 m/s (6 steps)	60℃	Controlled for $\Delta T=5$℃

shows a schematic of heat exchanger specimens used in this work. Flow paths for the 1st row and 2nd row are independently constructed in order to measure heat transfer coefficients of individual rows. Water flow rate through each row is independently regulated by control valves. Two types of fin, louver or slit, are tested in the present work.

Two specimens are tested for each fin type for the purpose of conduction effect examination. Fins for the 1st row and the 2nd row of the first specimen are separated while they are connected in the 2nd specimen. All heat exchangers tested are made of copper tube and aluminum fin and they are geometrically identical with the exception of fin shape. Physical data of heat exchanger samples are summarized in Table 1. The specimens are leak-tightly installed in front of a code tester and U-bends of specimens are well insulated during measurements. Experiments have been carried out under heating condition. Test conditions are shown in Table 2. Temperature and relative humidity (RH) of air in front of heat exchanger are 35℃ and 40%, respectively. Frontal air velocity varies from 0.7 to 2.5 m/s by 6 steps. Tube side water inlet temperature is 60℃. The water flow rate for each row is regulated such that temperature decrease in both rows takes place at the fixed value of 5℃. This way of doing experiments is somewhat different from Rich's method.[2] In his experiments, the tube side water flow rate was fixed such that the variation of the temperature drop between water inlet and outlet was allowed. Rich's measurements of temperature decrease in tube side water for downstream rows were as small as 0.1℃. Temperature decrease of 0.1℃ is close to modern RTD's measurement error bound so that the uncertainty of heat transfer calculation in downstream rows must be large. If water flow rate in individual row is regulated such that temperature decreases in every rows become 5℃, the uncertainty of heat transfer measurements will be negligible even though much more efforts are needed in experiments. Measurements of parameters are made when air temperature, water exit temperature and flow rates reach their steady states. Values of air flow rate, dry & wet bulb temperatures of air at both of inlet and outlet, water flow rate and inlet & outlet temperatures of water are read. The difference of heat transfer rates evaluated by air flow and by water flow is less than 3% in this experiment.

3. Data reduction

Heat transfer rate of a heat exchanger can be evaluated by an air temperature change or a water temperature change as follows:

$$Q_a = \dot{m}_a c_{p,a}(T_{a,i} - T_{a,o}) \tag{1}$$

$$Q_w = \dot{m}_w c_{p,w}(T_{w,i} - T_{w,o}) \tag{2}$$

The total heat transfer rate between the fluids, in the present work, is determined as an arithmetic mean of those two values,

$$Q = \frac{Q_a + Q_w}{2} \tag{3}$$

Since the inlet and outlet temperatures of air and water are measured, a log-mean-temperature-difference (ΔT_{LM}) can be calculated. Then, the average overall heat transfer coefficient (U) can be determined from the relationship of $Q = UA\Delta T_{LM}$. In general, thermal resistance of a fin-tube heat exchanger consists of convective resistance, conductive resistance and contact resistance. The very thin tube and fin of present heat exchangers are made of copper and aluminum, respectively. Since the conductive thermal resistance of these materials accounts for around 1% of total thermal resistance, it can be negligible. The total thermal resistance then can be expressed as follows:

$$\frac{1}{UA} = \frac{1}{h_c A_c} + \frac{1}{h_w A_w} + \frac{1}{\eta h_a A_a} \tag{4}$$

In order to evaluate the air-side heat transfer coefficient from the above equation, several parameters need to be determined. Contact resistance ($1/h_c A_c$) is determined by a correlation suggested by Sawai et al.[4] as follows:

$$\frac{h_c}{t_f} = 1.38\times10^{11}\Delta d_o + 1.62\times10^{7} \tag{5}$$

Tube side convective heat transfer coefficient can be evaluated by Gnielinski's correlation[5] as,

$$\mathrm{Nu}_w = \frac{(f_w/8)(\mathrm{Re}_w - 1000)\mathrm{Pr}_w}{1 + 12.7\sqrt{f_w/8}(\mathrm{Pr}_w^{2/3} - 1)}$$
$$f_w = (1.82 \ln \mathrm{Re}_w - 1.64)^{-2} \tag{6}$$

Overall surface efficiency has a relationship with fin efficiency as follows:

$$\eta = 1 - \frac{A_f}{A_a}(1 - \eta_f) \tag{7}$$

where fin efficiency can be calculated by Schmidt's correlation,[6]

$$\eta_f = \frac{\tanh\left(\frac{\beta d_c \phi}{2}\right)}{\left(\frac{\beta d_c \phi}{2}\right)} \tag{8}$$

Reynolds number of air is defined based on the maximum mass flux and hydraulic diameter determined at minimum free flow area,

$$\mathrm{Re}_a = \frac{G_{\max} d_h}{\mu_a} \tag{9}$$

where hydraulic diameter at the minimum free flow area is defined as follows:

$$d_h = \frac{4A_{free}L}{A_a} \tag{10}$$

Air heat transfer coefficient can be presented in terms of Colburn j-factor defined as,

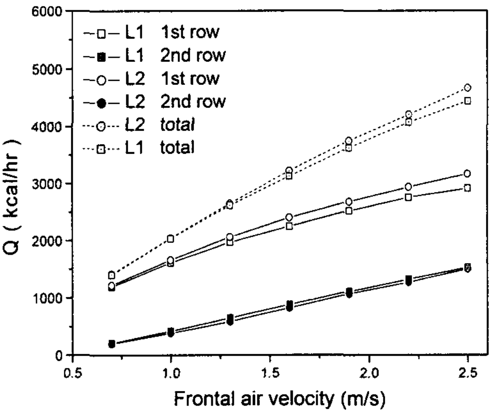
$$j = \frac{h_a}{G_{\max} C_{p,a}} \mathrm{Pr}_a^{2/3} \tag{11}$$

Based on error propagation analysis, uncertainties of water heat transfer rate, air heat transfer rate and air heat transfer coefficient are estimated to be less than 3%, 5% and 8%, respectively. The uncertainty of contact thermal resistance and tube-side convective resistance are assumed to be 5% in this calculation.

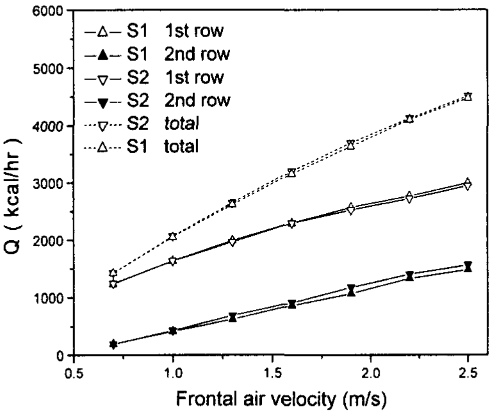
4. Results and Discussion

In order to evaluate heat transfer rate and heat transfer coefficient of the 2nd row, the conditions of air entering the 2nd row must be determined. However, it is not easy to measure these values because the spacing between fins or rows is very small. The conditions of air leaving the 1st row are assumed to be the same as those of air entering the 2nd row in the present analysis. For this purpose, it is assumed that the air leaving the 1st row gains the same amount of energy as the energy that the tube side water of the 1st row loses.

Heat transfer rates for louver fin and for slit fin heat exchangers versus frontal air velocity are presented in Fig. 3. This figure shows a monotonous increase in heat transfer rate with respect to the frontal air velocity. It also show that heat transfer rate in heat exchangers whose 1st and 2nd rows are connected (L2, S2) are larger than that for heat exchangers which are separated (L1, S1). There will be no conduction



(a) Louver fin



(b) Slit fin

Fig. 3 Heat transfer rate versus frontal air velocity.

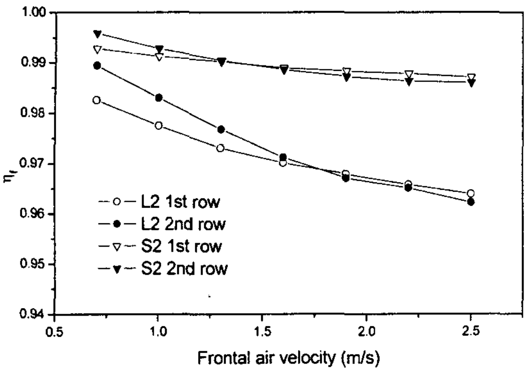


Fig. 4 Fin efficiency versus frontal air velocity.

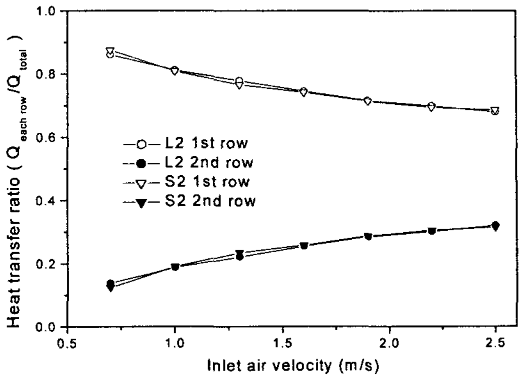
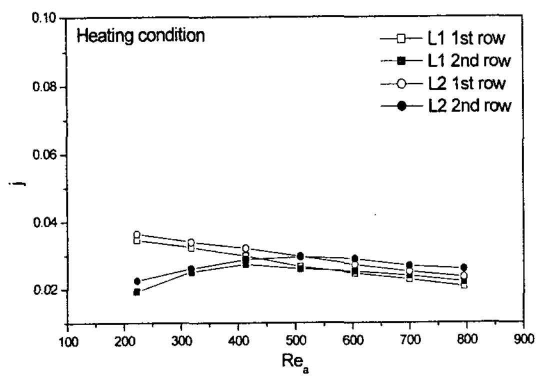


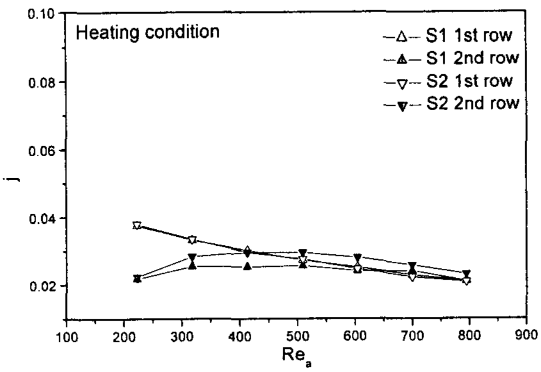
Fig. 5 Contribution of individual row to total heat transfer.

between rows in separated heat exchangers (L1, S1). However, conductive heat transfer between 1st and 2nd rows through fins is expected in connected heat exchangers (L2, S2) because the temperature distribution over streamwise length is not uniform. The temperature difference between fins and air, which is a driving force for convective heat transfer at downstream row (2nd row), is smaller than that at upstream row (1st row). Therefore, the heat energy is transferred to the 2nd row by means of conduction from the 1st row where temperature difference or driving force is large enough. This situation implies that more heat energy than expected to be transferred by the fins of the 2nd row is transferred due to the fins of the 1st row. In other words, the 2nd row utilizes the 1st row fins as well in heat transfer. This argument is supported by the fin efficiency of the 1st and 2nd rows as plotted in Fig. 4. This plot shows that the fin efficiency of the 2nd row is larger than that of the 1st row at low frontal air velocity. As the frontal air velocity increases, the discrepancy reduces or becomes opposite.

There still remains necessity of numerical simulation in order for more detailed information on what happens in multi-row fin-tube heat exchangers. Figure 5 shows the ratio of heat transfer rate of each row to the total heat transfer rate. Air-side heat transfer coefficient in terms of Colburn j-factor versus Reynolds



(a) Louver fin



(b) Slit fin

Fig. 6 Heat transfer coefficient of each row.

number is plotted in Fig. 6. The heat transfer coefficient of the 2nd row is close to or larger than that of the 1st row at high Reynolds number. In general, boundary layer develops around the leading edge region of the 1st row and it plays a role of convective thermal resistance. Also, turbulence is enhanced in the 2nd row and it will promote heat transfer by enhancing fluid mixing. This is why the heat transfer coefficient of the 2nd row is expected to be larger than that of the 1st row. In the low Reynolds number region, However, heat transfer coefficient of the 2nd row appears to be smaller than that of the 1st row. This is controversy to the aforementioned expectation. It is speculated that poor heat transfer zone developed in the rear of tubes of the 1st row caused this result. That is, a vortex system develops in back of a tube. The air in this vortex system does not tend to flow across the wake boundary, but circulates inside the wake. Since net air flow rate in this zone is smaller than the other region, temperature of air is close to tube or fin. This vortex system expands downstream to reach the front of the next row tube and make convective heat transfer in the 2nd row ineffective when Reynolds number is low. Saboya & Sparrow[7] showed the effect of a vortex system on heat transfer by means of naphthalene sublimation method. Flow pattern of a vortex system developed in rear of a tube and its behaviour depending on Reynolds number is well described in Tasi et al.'s numerical analysis.[8] As the Reynolds number increases, turbulence develops further and poor heat transfer zone of a vortex system shrinks so that heat transfer coefficient of the 2nd row increases. This trend of heat transfer coefficient variation is similar to Rich's[2] experimental results.

5. Conclusions

An experimental study was conducted to investigate the heat transfer characteristics of individual rows of fin and tube heat exchangers. Row-by-row variations in heat transfer rate and heat transfer coefficients are investigated for geometrically identical four heat exchangers with the exception of fin shape, slit or louver, and that the fins associated between the first row and the second row are connected or separated. The important conclusions made in this study are summarized as follows:

(1) Heat transfer rate of connected heat exchanger is shown to be larger than that for separated heat exchanger due to the conduction effect through the streamwise length of the fin when connected.

(2) Heat transfer coefficient of the 2nd row is close to or larger than that for the 1st row at high Reynolds numbers, but is smaller at low Reynolds numbers.

(3) Heat transfer performance of multi-row heat exchangers can be enhanced by avoiding row-to-row separation, increasing frontal air velocity and/or arranging heat transfer tubes in a staggered way.

Acknowledgement

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