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Heat Transfer and Pressure Drop Characteristics of Plain Finned Heat Exchangers Having 5.0 mm Tubes

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Abstract

In this study, pressure drop and heat transfer characteristics of plain finned heat exchangers having 5.0 mm diameter (fin collar 5.3 mm) tubes were investigated. Six samples having different fin pitches (1.1 to 1.3 mm) and tube rows (1 and 2 row) were tested. The fin pitch had a negligible effect on j and f factors. Both j and f factors decreased as the number of tube row increased, although the difference was not significant for the f factor. When compared with the j and f factors of the samples having 7.3 mm diameter tubes, the present j and f factors yielded lower values. However, the j/f ratio was larger at low Reynolds numbers. Possible reasoning is provided from the flow pattern consideration. Comparison with existing correlations were made.

Key words: Dry surface; Heat transfer coefficient; Plain fin; Heat exchanger; Pressure drop

Nomenclature

A : heat transfer area [m^2]
c_p : specific heat [kJ/kgK]
D : tube diameter including fin collar[m]
D_r : tube diameter to the fin root [m]
f : friction factor [-]
G : mass flux [kg/m^{2}s]
h : heat transfer coefficient [W/m^{2}K]
j : Colburn j factor $\left(= \dfrac{h}{\rho V_{\max} c_p} Pr_a^{2/3}\right)$
k : thermal conductivity [W/mK]
Q : flow rate [kg/s]
N : number of row [-]
Nu_w : tube-sdie Nusselt number $\left(= \dfrac{h_i D_r}{k_w}\right)$
NTU : number of transfer unit [-]
P_f : fin pitch [m]
P_l : longitudinal tube pitch [m]
P_t : transverse tube pitch [m]
Pr : Prandtl number [-]
r_c : tube radius including fin collar [m]
R_{eq} : equivalent radius [m]
Re_D : Reynolds number based on tube diameter $\left(= \dfrac{V_{\max} D}{\nu}\right)$
Re_w : tube-side Reynolds number $\left(= \dfrac{G D_r}{\mu}\right)$
T : temperature [K]
t : tube wall thickness [m]
U : overall heat transfer coefficient [W/m^{2}K]
V_{max} : velocity based on the minimum flow area of the frontal surface [m/s]

Greek symbols

ΔP : pressure loss [Pa]
ν : kinematic viscosity [m^2/s]
η : fin efficiency [-]
η_0 : surface efficiency [-]
μ : dynamic viscosity [kg/ms]
ρ : density [kg/m^3]
σ : contraction coefficient [-]

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Lower characters

a	: air
c	: fin collar, cross-section
D	: tube diameter
f	: fin
i	: tube inside
m	: mean
max	: maximum
o	: tube outside
r	: refrigerant
t	: tube wall
w	: water

1. Introduction

Fin-and-tube heat exchangers are made of flat fins attached to circular tubes. The circular tubes 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. Although high performance fins such as louver or slit fins are widely used for household air-conditioners, plain fins are often preferred for dehumidification and frosting applications.

Many studies have been performed for the heat transfer and pressure drop characteristics of plain finned heat exchangers. Rich[1] tested 12.7 mm O.D., four-row samples having 114 to 811 fins/m, and reported that the effect of fin pitch on the heat transfer coefficient and friction factor was negligible. Later, he[2] investigated the effect of the tube row on the heat transfer coefficient and pressure drop for the samples having 1 to 6 rows at fixed fin pitch of 551 fins/m. At Reynolds numbers lower than 3000, the heat transfer coefficient increased as the tube row decreased. At higher Reynolds numbers, however, the trend was reversed. The reason may be attributed to the competing effect of the boundary layer and tube-generated turbulence[3,4]. At low Reynolds numbers, the boudary layer on the fin surface controls the heat transfer, and the heat transfer coefficient decreases as the tube row increases. At high Reynolds numbers, however, the tube-generated turbulence overcomes the boundary layer effect, and the heat transfer coefficient becomes larger for samples having larger number of tube rows. Following Rich[1,2], many investigations were conducted for different tube diameters and fin pitches. Those include McQuiston[5](D = 9.66 mm), Seshimo and Fujii[6](D = 9.52 mm), Kayansayan[7](D = 9.52, 12.3, 16.3 mm), Wang et al.[8](D = 10.23 mm), Wang and Chi[9](D = 7.23, 8.3, 10.0 mm) and Min et al.[10](D = 7.3 mm). Their findings were generally in line with those of Rich[1,2]. The heat transfer coefficient was almost independent of the fin pitch, and the friction factor was almost independent of the tube row.

Numerical attemps were also made to resolve the flow and heat transfer characteristics of the plain finned tube heat exchangers. Torikoshi et al.[11] conducted unsteady three dimensional computation for one row samples having different fin pitch. With the increase of fin pitch, the flow downstream of the samples was further disturbed. However, the heat transfer from the fin surface was almost independent of the fin pitch. Torikoshi and Xi[12] numerically investigated the effect of tube diameter, and reported that the pressure drop increased as the tube diameter increased. However, the heat transfer coefficient was almost independent of the fin pitch. Onishi et al.[13], Tsai et al.[14] considered the conjugate heat transfer from the fin surface.

The literature shows that no experimental data are available for fin-and-tube heat exchangers having tube diameter smaller than 7.0 mm. In this study, experiments are conducted for six fin-and-tube heat exchanger samples having 5.0 mm diameter tubes(5.3 mm after tube expansion). Data are compared with existing data of samples having 7.0 mm diameter tubes.

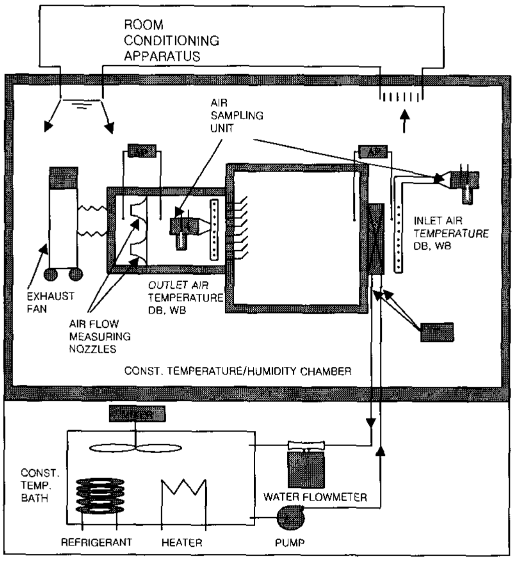

Fig. 1. Schematic drawing of the experimental setup.

2. Experiment

2.1 Heat exchanger samples

Six heat exchangers tested in the present study consist of three different fin pitch (1.1, 1.2, 1.3 mm) and two different tube row (1, 2 row). The height and the width of the samples were 234 mm and 400 mm respectively. For all the samples, the tube outer diameter after expansion was 5.3 mm, the transverse tube pitch (P_t) was 19.5 mm and the longitudinal tube pitch (P_l) was 11.2 mm. Micro-fin tubes having 60 microfins with 0.12 mm fin height and 25 degree helix angle were used at the tube-side. The tube-side was circuited to cross-counter configuration with single inlet and outlet.

2.2 Experimental apparatus and test procedures

A schematic drawing of the apparatus is shown in Fig. 1. 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 K. The water flow rate is measured by a positive displacement type flow meter, whose accuracy is ± 0.0015 liter/s. 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 transducers is 1 Pa.

During the experiment, the water temperature was held at 50℃. The chamber temperature was maintained at 21℃ with 50% 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 matched within ± 3%. The discrepancy increased as the air velocity decreased. An uncertainty analysis was conducted following ASHRAE Standard 41.5[17]. The uncertainty on the heat transfer coefficient was ± 12 %, and that of friction factor was ± 10%. 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 Reynolds number increased.

2.3 Data reduction

For the cross-counter configuration of the present study, appropriate equations for the heat exchanger analysis are given by Taborek[18]. The UA value is obtained from the following equations.

$$UA = \left\{ \left(\dot{m} c_p \right)_a \mathrm{NTU}_2 \right\} \tag{1}$$

$$\mathrm{NTU}_2 = -2 \ln (1 - K) \tag{2}$$

For two row configuration, the K is obtained from the following equations.

$$\frac{K}{2} + \left(1 - \frac{K}{2} \right) \exp(2KR) = \frac{1}{1 - RP} \tag{3}$$

$$R = \frac{T_{r,in} - T_{r,out}}{T_{a,in} - T_{a,out}} \tag{4}$$

$$P = \frac{T_{a,out} - T_{a,in}}{T_{r,in} - T_{air,out}} \tag{5}$$

For the one row configuration, a cross-flow ε - NTU equation was used. The airside heat transfer coefficient is obtained from the following equation.

$$\frac{1}{\eta_o h_o A_o} = \frac{1}{UA} - \frac{1}{h_i A_i} - \frac{t}{k A_t} \tag{6}$$

For the tube-side heat transfer coefficients, the correlation[19] 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.

$$Nu_w = 0.0172 Re_w^{1.12} \mathrm{Pr}_w^{0.3} \quad 300 \le Re_w \le 21000 \tag{7}$$

$$Nu_w = 0.0376 Re_w^{0.81} \mathrm{Pr}_w^{0.3} \quad 21000 \le Re_w \le 40000 \tag{8}$$

For the characteristic length scale of the Reynolds and Nusselt number in Eqs. (7) and (8), tube diameter to the fin root of the micro-fin was used.

The surface efficiency η_o in Eq. (6) is obtained from

$$\eta_o = 1 - \frac{A_f}{A_o}(1-\eta) \tag{9}$$

The fin efficiency is obtained from Schmidt[20] equation. The heat transfer coefficient is expressed as the j-factor.

$$\mathrm{Re_D} = \frac{\rho_a V_{max} D_c}{\mu_a} \tag{10}$$

$$j = \frac{h_o}{\rho_a V_{max} c_{pa}} \mathrm{Pr}_a^{2/3} \tag{11}$$

All the fluid properties are evaluated at an average air temperature. The core friction factor is calculated from the measured pressure drop following Kays and London[21].

$$f = \frac{A_c}{A_o}\frac{\rho_m}{\rho_{in}}\left[\frac{2\Delta P \rho_{in}}{(\rho_m V_{max})^2} - (1+\sigma^2)\left(\frac{\rho_{in}}{\rho_{out}} - 1\right)\right] \tag{12}$$

In Eq. (12), the entrance and the exit loss coefficients are neglected following the suggestion by Wang et al.[22]

3. Results and discussion

The j and f factors of the one and two row samples are shown in Figs. 2 and 3 respectively. The figures show that effect of fin pitch on the j and f factor is negligible. It has been reported by many investigators that, for fin-and-tube heat exchangers, the effect of fin pitch on the heat transfer coefficient and the friction factor is not significant[1,8,10]. 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.

In Fig. 4, data are presented for different tube row at the fin pitch of 1.3 mm. The j factor increases as the tube row decreases. This trend is commonly observed for fin-and-tube heat exchangers[2,8,10], and could be explained by the boundary layer effect. At low Reynolds numbers, the boudary layer on the fin surface controls the heat transfer, and the heat transfer coefficient decreases as the tube row increases.

In Figs. 5 and 6, the present data (P_f = 1.3 mm) are compared with those of plain fin-and-tube samples having 7.3 mm diameter tubes[10](P_f = 1.5 mm). The smallest fin pitch available for the samples having 7.3 mm diameter tubes was 1.5 mm. Considering that the

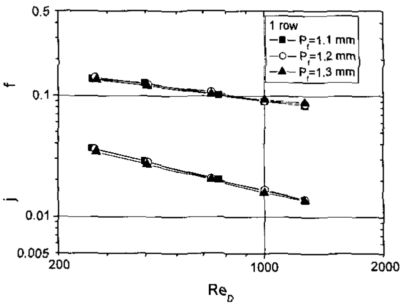

Fig. 2. Effect of fin pitch on the 1 row j and f factor.

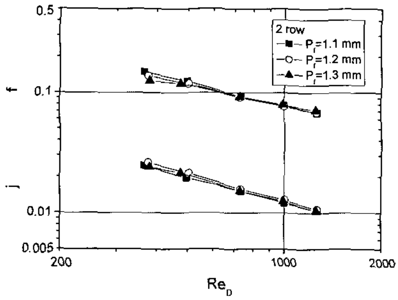

Fig. 3. Effect of fin pitch on the 2 row j and f factor.

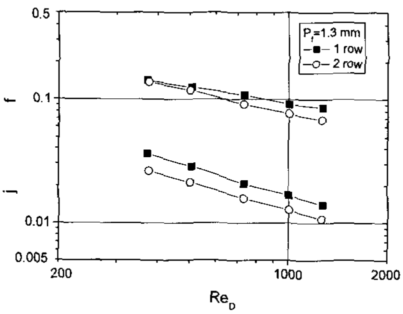

Fig. 4. Effect of tube row on the j and f for 1.3 mm fin pitch.

effect of fin pitch on j and f factor is not significant, the 0.2 mm fin pitch difference is believed not to cause significant problem for the comparison. For the samples having 7.3 mm diameter tubes, P_t = 21.0 mm and P_l = 12.5 mm, which are slightly larger than those of the present samples (P_t = 19.5 mm and P_l = 11.2 mm). The one row data are compared in Fig. 5, which shows that j and f factors of 7.3 mm sample are larger than the present 5.3 mm sample by 18% and 17% respectively. In Fig. 6, two row data are compared. Same as one row, 7.3 mm sample yields 15% larger j factor and 16% larger f factor compared with 5.3 mm sample. At first glance, this result appears against the notion that heat exchangers having smaller tubes yield superior performance. However, one thing to note is that the Reynolds number of this study is defined using tube diameter as the characteristic length.

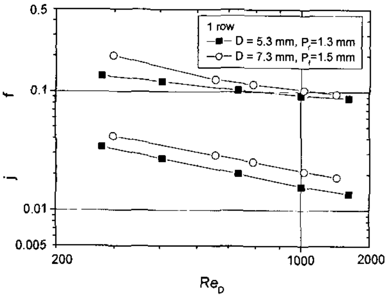

Fig. 5. Effect of tube diameter on the 1 row *j* and *f* factor.

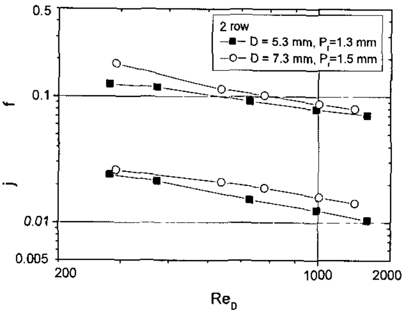

Fig. 6. Effect of tube diameter on the 2 row *j* and *f* factor.

Thus, at the same frontal velocity, the Reynolds number of the 5.3 mm sample is 27% smaller than that of 7.3 mm sample.

Figs. 5 and 6 show that the f factor difference between 7.3 mm and 5.3 mm sample increases as the Reynolds number decreases. The reason could be attributed to the recirculation zone, which exist behind the tubes at low Reynolds numbers. The size of the recirculation zone decreases as the tube diameter decreases, yielding smaller pressure drop. Thus, we may expect superior performance of the smaller diameter samples at low Reynolds numbers. In Fig. 7, the j/f ratios of the 5.3 mm samples are compared with those of the 7.3 mm samples. This figure shows that 5.3 mm samples yield higher j/f ratio below Reynolds numbers of 650~700.

The present data and those of samples having 7.3 mm diameter tubes[10] are compared with existing correlations for plain fin-and-tube heat exchangers [Gray and Webb[23], Kim et al.[24], Wang et al.[25]]. Note that the correlations were developed using the data of samples having larger than 7.0 mm diameter tubes. The j factors are compared in Fig. 8. The correlations by Wang et al.[25] and Kim et al.[24] overpredict most of the the data, whereas Gray and Webb correlation[23] underpredicts most of the data. Especially, the present 5.3 mm data are highly overpredicted, probably because tube diamete of the present data is

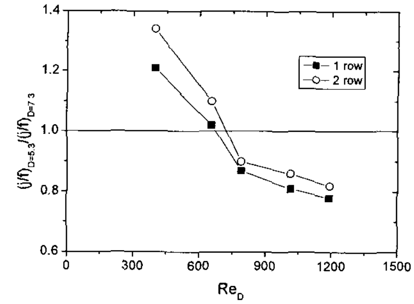

Fig. 7. (j/f) ratios of the present samples compared with those of the 7.3 mm tube diameter samples.

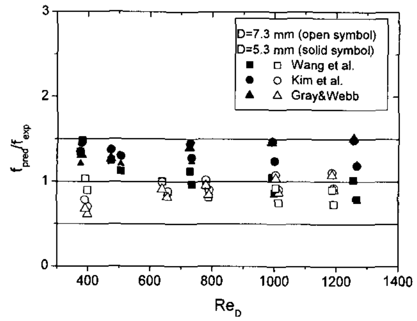

Fig. 8. Present j factors compared with predictions.

smaller than the applicable range of the existing correlations. The 7.3 mm data are best predicted by Kim et al.[24] correlation with 13% standard deviation, whereas 5.3 mm data are best predicted by Geay and Webb[23] correlation with 14% standard deviation. The f factors are compared in Fig. 9. Same as j factor, most of the 5.3 mm data are overpredicted. The 7.3 mm data are best predicted by Kim et al.[24] correlation with 14% standard deviation, whereas 5.3 mm data are best predicted by Wang et al.[25] correlation with 23% standard deviation.

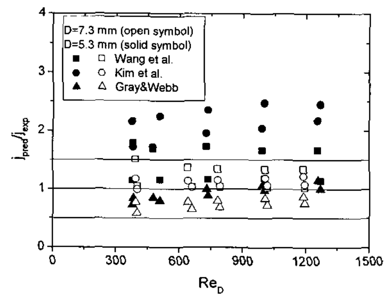

Fig. 9. Present f factors compared with predictions.

4. Conclusions

In this study, heat transfer and pressure drop tests were conducted for six samples(P_f = 1.1~1.3 mm, 1~2 row) having 5.3 mm diameter tubes.

1) The effect of fin pitch on j and f factors is negligible.
2) Both j and f factor decrease as tube row increases, although the effect is not significant for f factors.
3) The j and f factors of the samples having 7.3 mm diameter tubes are larger than those of samples having 5.3 mm tubes. However, the j/f ratio is larger for 5.3 mm samples especially at low Reynolds numbers.
4) The heat transfer coefficient is well predicted by Gray and Webb(23) correlation, and the friction factor is well predicted by Wang et al.(25) correlation.

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