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Technical Note

A heat transfer and friction correlation for wavy fin-and-tube heat exchangers

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Nomenclature

A_c minimum free-flow area [m^2]
A_f fin surface area [m^2]
A_t tube surface area [m^2]
A_0 total surface area [m^2]
D_c fin collar outside diameter, $D_0+2\delta_f$ [m]
D_h hydraulic diameter, $4A_cL/A_0$ [m]
f friction factor, dimensionless
F_p fin pitch [m]
G_c mass flux of the air based on the minimum flow area [kg m^{-2} s^{-1}]
h heat transfer coefficient [W m^{-2} K^{-1}]
j $Nu/Re\,Pr^{1/3}$, the Colburn factor, dimensionless
k thermal conductivity [W m^{-1} K^{-1}]
L depth of the heat exchanger [m]
N number of longitudinal tube rows, dimensionless
NTU UA/C_{min}, number of transfer units, dimensionless
ΔP pressure drop [Pa]
P_l longitudinal tube pitch [m]
P_d waffle height [m]
P_t transverse tube pitch [m]
Pr Prandtl number, dimensionless
Re_{Dc} Reynolds number based on tube collar diameter, dimensionless
S fin spacing [m]
X_f projected fin length [m].

Greek symbols
ε $\dot{Q}/\dot{Q}_{max}$, heat exchanger effectiveness, dimensionless
θ corrugation angle [°]
δ_f fin thickness [m]
μ dynamic viscosity of fluid [kg m^{-1} s^{-1}]
ρ mass density of fluid [kg m^{-3}]
σ contraction ratio of cross-sectional area, dimensionless.

Subscripts
air air side
i tube side
f fin surface
0 total surface
1 inlet
2 outlet.

1. Introduction

Finned tube heat exchangers are widely used in a variety of applications in the air-conditioning, refrigeration, and process industry. Generally, the heat exchangers consist of a plurality of spaced parallel tubes through which water, oils, or refrigerants is forced to flow while air is directed across the tubes. Since the dominate resistance is usually on the air-side, therefore, the use of enhanced fin surfaces is very common to effectively improve the overall heat transfer performance. One of the very popular fin patterns is the wavy fin configuration as shown in Fig. 1.

Despite there being several efforts devoted to the air-side performances [1–6], as pointed out in [7], significant differences of air-side performances may occur owing to the differences of reduction method. The objective of the present study is to construct the correlations for the air-side performances of wavy fin-and-tube heat exchangers based on consistent reduction methods.

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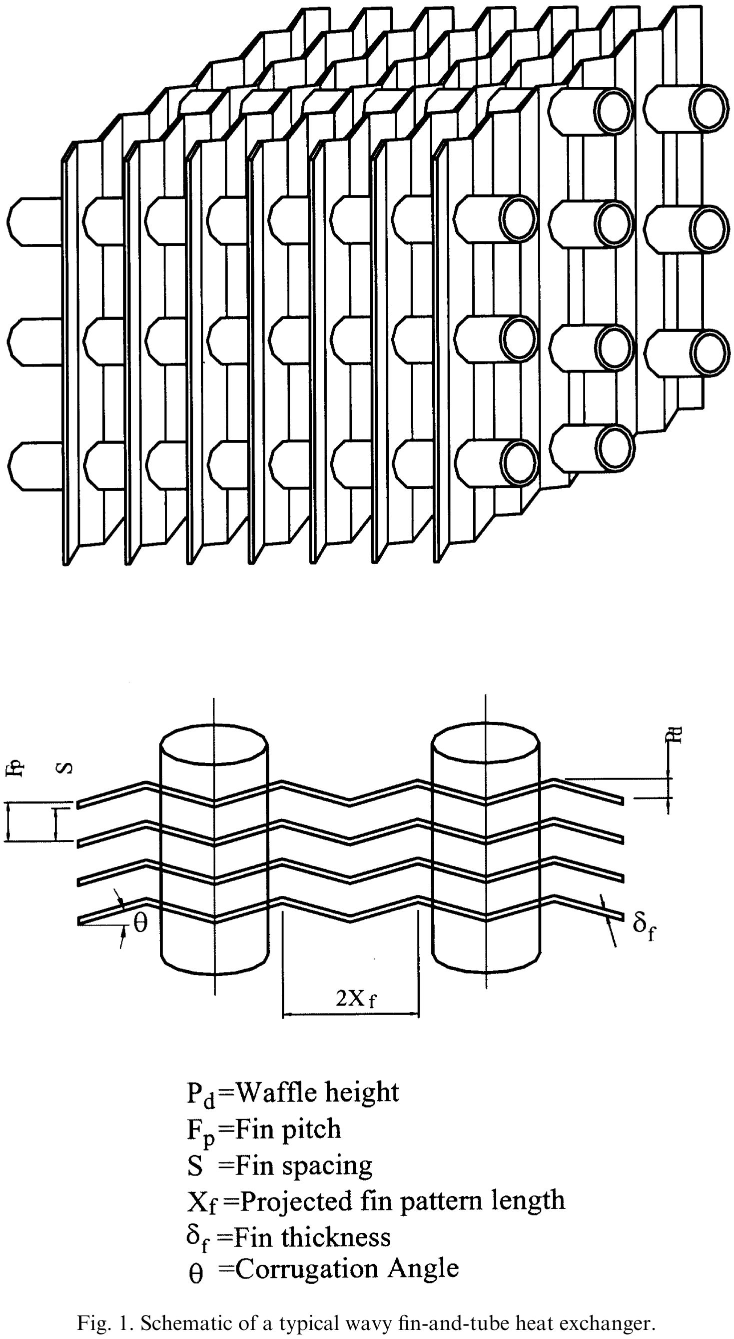

Fig. 1. Schematic of a typical wavy fin-and-tube heat exchanger.

2. The data bank

The database for the present study are taken from [4,5] and our newly tested samples. A total of 27 samples are used for the development of correlations. The database includes those by Wang et al. [5] (7 samples), Wang et al. [6] (8 samples), and additional twelve test samples conducted in this new study. Detailed geometry for the test samples is tabulated in Table 1.

The detailed experimental facility data had been described by several previous studies [8, 9], and will not repeat here. The heat transfer coefficients for the present

Table 1
Geometric dimensions of the wavy fin-and-tube heat exchangers

No	Fin pitch (mm)	Fin thickness (mm)	D_c (mm)	P_t (mm)	P_l (mm)	Waffle height (mm)	Row No.
1	2.54	0.115	8.58	25.4	19.05	1.32	1
2	1.21	0.115	8.58	25.4	19.05	1.32	1
3	2.54	0.115	8.58	25.4	19.05	1.32	2
4	1.69	0.115	8.58	25.4	19.05	1.32	2
5	1.21	0.115	8.58	25.4	19.05	1.32	2
6	2.54	0.115	8.58	25.4	19.05	1.32	4
7	1.21	0.115	8.58	25.4	19.05	1.32	4
8	1.70	0.12	8.62	25.4	19.05	1.18	2
9	1.69	0.12	8.62	25.4	19.05	1.58	2
10	3.09	0.12	8.62	25.4	19.05	1.18	2
11	3.17	0.12	8.62	25.4	19.05	1.58	2
12	1.65	0.12	8.62	25.4	19.05	1.18	4
13	1.70	0.12	8.62	25.4	19.05	1.58	4
14	3.11	0.12	8.62	25.4	19.05	1.18	4
15	3.14	0.12	8.62	25.4	19.05	1.58	4
16	2.85	0.12	10.38	25.4	19.05	1.18	6
17	3.09	0.12	8.62	25.4	19.05	1.58	6
18	1.63	0.12	10.38	25.4	19.05	1.18	6
19	2.87	0.12	10.38	25.4	19.05	1.18	6
20	1.59	0.12	10.38	25.4	19.05	1.18	6
21	1.67	0.12	8.62	25.4	19.05	1.58	6
22	2.85	0.12	10.38	25.4	19.05	1.18	1
23	2.95	0.12	8.62	25.4	19.05	1.58	1
24	1.65	0.12	8.62	25.4	19.05	1.58	1
25	3.58	0.12	8.62	25.4	19.05	1.58	1
26	1.62	0.12	10.38	25.4	19.05	1.18	1
27	3.66	0.12	8.62	25.4	25.4	1.68	1

Note: the sample heat exchangers are all staggered layout.

study were reduced using the ε–NTU methods, and the fin efficiency is calculated using the Schmidt [10] approximation.

The core friction of the heat exchanger is calculated from the pressure drop equation proposed by Kays and London [11]. The relation for the Fanning friction factors which includes the entrance and exit pressure loss, and is expressed as below:

$$f=\frac{A_c\rho_1}{A_0\rho_m}\left[\frac{2\Delta P}{G_c^2\rho_1}-(1+\sigma^2)\left(\frac{\rho_1}{\rho_2}-1\right)\right] \quad (1)$$

where A_0 and A_c stand for the total surface area and the flow cross-sectional area, respectively. The term, σ, is the ratio of the minimum flow area to frontal area.

3. The correlation

Attempts are made to correlate the test results by using a multiple regression technique. The basic forms of the correlations are:

$$j=C_1\,Re_{\mathrm{Dc}}^{C_2} \quad (2)$$

$$f=C_3\,Re_{\mathrm{Dc}}^{C_4}. \quad (3)$$

It is assumed that C_1, C_2, C_3, and C_4 are dependent on the physical dimensions of the heat exchanger. A separate multiple linear regression was proceeded to determine the exponents, C_2, and C_4 of the heat exchangers. The determinations of C_1 and C_3 are analogous to, C_2, and C_4. After a trial and error process, the final equation forms for the j factors are given as follows:

$$j=0.324\,Re_{D_c}^{J1}\left(\frac{F_p}{P_l}\right)^{J2}(\tan\theta)^{J3}\left(\frac{P_l}{P_t}\right)^{J4}N^{0.428} \quad (4)$$

where

$$J1=-0.229+0.115\left(\frac{F_p}{D_c}\right)^{0.6}\left(\frac{P_l}{D_h}\right)^{0.54}\times N^{-0.284}\log_e(0.5\tan\theta) \quad (5)$$

$$J2=-0.251+\frac{0.232N^{1.37}}{(\log_e(Re_{D_c})-2.303)} \quad (6)$$

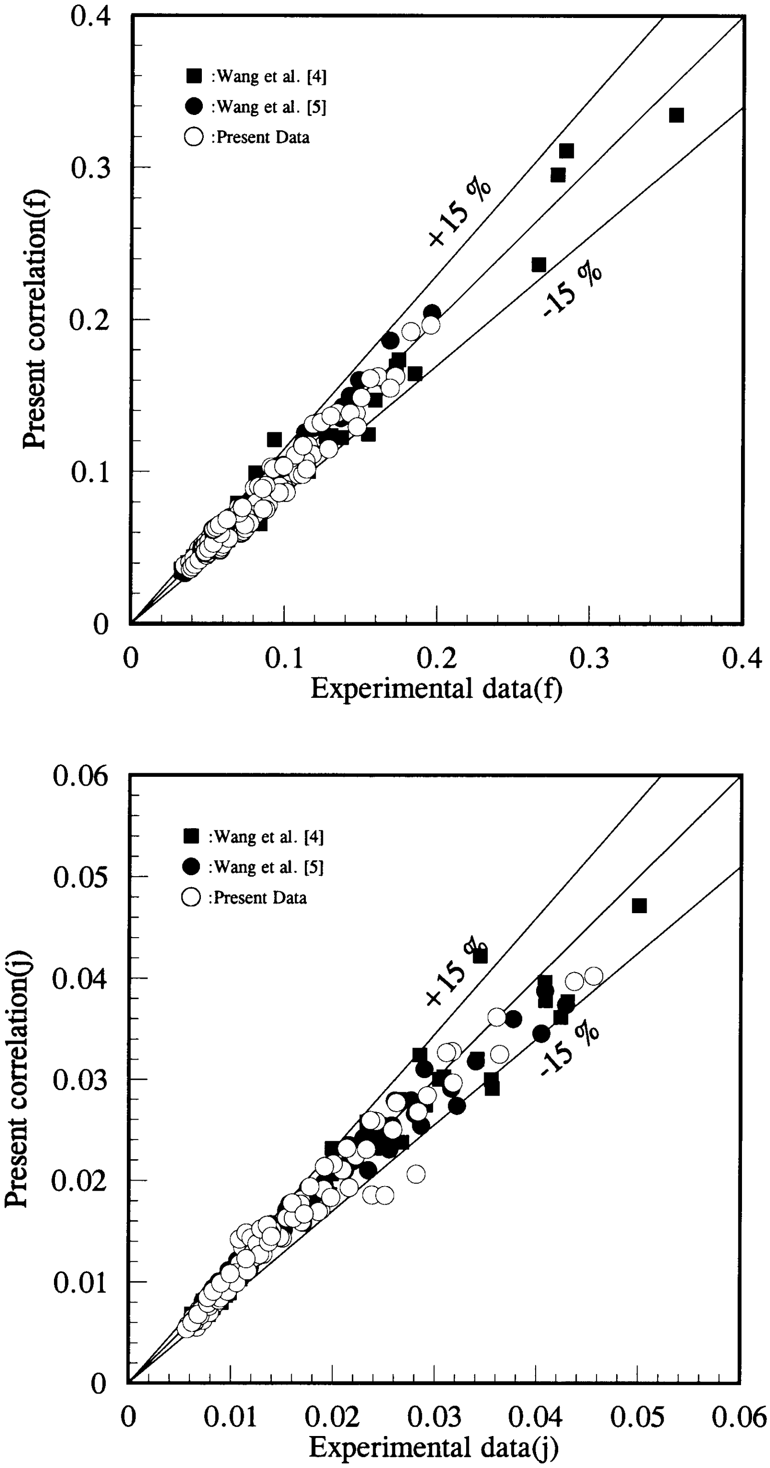

Fig. 2. Comparisons of the present heat transfer and friction correlations with the experimental data.

Table 2
Comparison of the proposed correlation with the experimental data

Deviation	$\pm 10\%$ (%)	$\pm 15\%$ (%)	$\pm 20\%$ (%)	$\pm 25\%$ (%)	Mean deviation	Average deviation
j	79.6	95.1	97.4	98.5	6.44	0.35
f	84.9	97.3	99.6	100	5.01	0.24

$$\text{Average deviation} = \frac{1}{M}\left[\sum_{1}^{M}\frac{j(f)_{\text{pred}} - j(f)_{\text{exp}}}{j(f)_{\text{exp}}}\right] \times 100\%.$$

$$\text{Mean deviation} = \frac{1}{M}\left[\sum_{1}^{M}\frac{|j(f)_{\text{pred}} - j(f)_{\text{exp}}|}{j(f)_{\text{exp}}}\right] \times 100\%.$$

M: number of data points.

$$J3 = -0.439\left(\frac{F_p}{D_h}\right)^{0.09}\left(\frac{P_l}{P_t}\right)^{-1.75} N^{-0.93} \tag{7}$$

$$J4 = 0.502(\log_e(Re_{D_c}) - 2.54). \tag{8}$$

The correlation for friction factor is given as:

$$f = 0.01915\, Re_{D_c}^{F1}(\tan\theta)^{F2}\left(\frac{F_p}{P_l}\right)^{F3}\left(\log_e\left(\frac{A_0}{A_t}\right)\right)^{-5.35} \times\left(\frac{D_h}{D_c}\right)^{1.3796} N^{-0.0916} \tag{9}$$

where

$$F1 = 0.4604 - 0.01336\left(\frac{F_p}{P_l}\right)^{0.58}\log_e\left(\frac{A_0}{A_t}\right)(\tan\theta)^{-1.5} \tag{10}$$

$$F2 = 3.247\left(\frac{F_p}{P_t}\right)^{1.4}\log_e\left(\frac{A_0}{A_t}\right) \tag{11}$$

$$F3 = \frac{-20.113}{\log_e(Re_{D_c})}. \tag{12}$$

Figure 2 shows the comparisons of the experimental data with equations (4) and (9). Equation (4) can describe 95.1% of the j factors within $\pm 15\%$ and equation (9) can correlate 97.3% of the friction factors within $\pm 15\%$. The results of the comparisons of the present correlations with all the test data are tabulated in Table 2. As seen, the present heat transfer correlation gives a mean deviation of 6.44% while the proposed friction correlation shows a 5.01% mean deviation.

4. Conclusions

A generalized heat transfer and friction correlation for wavy fin geometry is proposed from a total of 27 samples of fin-and-tube heat exchangers. The proposed heat transfer correlation can describe 95.1% of the test data within $\pm 15\%$ with a mean deviation of 6.44% while the proposed friction correlation can describe 97.3% of the results within $\pm 15\%$ with a mean deviation of 5.01%.

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