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HEAT TRANSFER AND FRICTION CORRELATIONS FOR PLATE FINNED-TUBE HEAT EXCHANGERS HAVING PLAIN FINS

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ABSTRACT

This paper is concerned with heat exchangers having continuous, flat fins on a staggered array of circular tubes. Correlations are developed to predict the air-side heat transfer coefficient and friction factor as a function of the Reynolds number and the geometric variables of the heat exchanger. The results are applicable to any number of tube rows in the air flow direction. A multiple regression technique was used with data for 16 heat exchangers to develop the heat transfer correlation. The rms error of the resulting correlation is 7.3%. The friction correlation is based on a superposition model, which assumes that the drag force on the tubes and that on the fins may be added together. After subtracting the tube bank friction loss from the 18 sets of heat exchanger data, a multiple regression technique was used to correlate the fin friction data. The resulting heat exchanger friction factor correlated the 18 data set with an rms deviation of 7.8%.

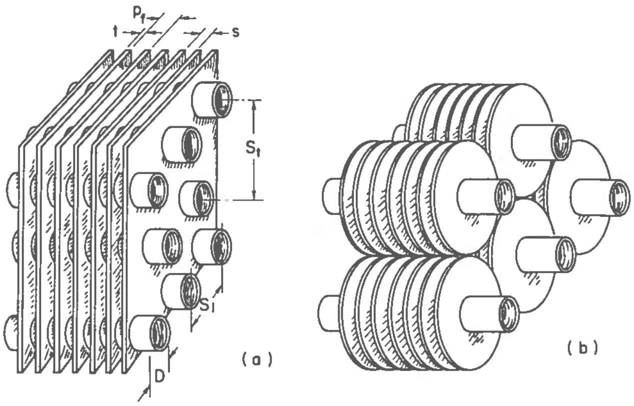


Fig. 1 Two types of finned-tube heat exchangers a. Plate fin type, b. Individually finned tubes.

1. INTRODUCTION

Air cooled, finned-tube heat-exchangers are used in many applications. There are two basic design types of these heat exchangers: Plate finned-tubes (Fig. 1a) and individually finned tubes (Fig. 1b). Designers have need of correlations to predict the heat transfer coefficient and friction factor as a function of Reynolds number and the geometric parameters. Heat transfer [1] and friction [2] correlations have been developed for the individually finned tube (circular fins) geometry with staggered tubes. McQuiston [3] has proposed a correlation for the heat transfer coefficient for the plate finned-tube geometry with plain fins and a staggered tube layout. Although McQuiston also developed a friction correlation, its accuracy was quite poor. "Most of the data" were correlated within 35%.

The purpose of this paper is to present heat transfer and friction correlations for the plate fined-tube geometry with plain fins and a staggered tube layout. Multiple regression techniques are used to develop the correlations. The dimensionless geometric parameters considered are the tube bank parameters ($S_t$, $S_l$, D, N), and the fin spacing (s). The fin thickness is not included, because it affects only the air flow velocity in the heat-exchanger, which is accounted for by the Reynolds number. The entrance and exit pressure drops are influenced by the fin thickness. However, the friction correlation excludes the entrance and exit losses, so it is not necessary to include fin thickness as a variable.

The six data sources used to develop the correlations are described in Table 1. Data on 16 and 18 heat-exchangers were used to develop the heat transfer and friction correlations, respectively. These data sets are indicated by the H (heat transfer) and F (friction) entries on lines three and four of Table 1. Two to three data points were used for each heat exchanger, which spanned the range of Reynolds numbers tested. Care was taken to avoid biasing the correlation by including a large number of data points having a small range of particular geometric parameters, e.g., $S_l/D$. For example, all eight heat exchangers of Reference 4 have $S_t/D = 2.38$, so additional points for $S_l/D = 1.97$ (9) were added to the data bank.

As shown by line five of Table 1, five additional heat transfer data sets, which were not used in the development of the correlation, are included in the calculation of the "rms" deviation of the heat transfer correlation. Also, the ability of the heat transfer correlation to predict the data of References 11 and 12 will be evaluated.

The flow variables used in the correlations are the Reynolds number (based on the tube outside diameter) and the mass velocity in the minimum flow area ($A_c$). The heat transfer parameter used by the correlation is the j-factor, $StPr^{2/3}$. The friction factor correlation uses the Fanning friction factor.

There is no "correct" definition for the characteristic dimension used in the Reynolds number. Both the tube diameter (D) and the hydraulic dia-

TABLE 1
SOURCES USED IN DEVELOPMENT OF CORRELATIONS
(* Code is H=heat transfer; F=friction)

| Reference | 7 | 4 | 8 | 9 | 10 | 11 | 12 |
|---|---|---|---|---|---|---|---|
| Symbol | ⬡ | □ | ◇ | ▽△ | ○● | ◆ | |
| Correlation Code* | H/F | H/F | H | H/F | H/F | F | |
| No. H-X for H/F Correl. | 4/5 | 4/8 | 4/0 | 2/2 | 2/2 | 0/1 | 0/0 |
| Added H-X to check H/F correlation | 1/0 | 4/0 | 0/0 | 0/0 | 0/0 | 0/0 | 0/0 |
| D (mm) | 9.96 | 13.33 | 13.23 | 10.34 | 10.21/17.17 | 19.51 | 16.51 |
| $S_t/D$ | 2.55 | 2.38 | 2.40 | 1.97 | 2.22/2.49 | 2.60 | 1.82 |
| $S_1/D$ | 2.22 | 2.06 | 2.08 | 1.70 | 2.15/2.59 | 2.26 | 2.79 |
| s/D | 0.17/0.30 | 0.08/0.64 | 0.12 | 0.16/0.29 | 0.17/0.28 | 0.12 | 0.12 |
| t/D | 0.015 | 0.011 | 0.011 | 0.016 | 0.024/0.032 | 0.014 | 0.018 |

meter ($D_h$) have been used (1,4). The authors used both Reynolds numbers in the development of the heat transfer correlation. The tube diameter based Reynolds number yielded a slightly better correlation than that based on the hydraulic diameter. The tube diameter based Reynolds number was thus selected for the correlation. This is the same Reynolds number definition used by Briggs and Young [1]. Both Briggs and Young [1] and Rich [4] have shown that the j-factor is dependent on the mass velocity in the minimum flow area of the finned tube bundle ($G_c$).

Careful readers of Rich's papers [4,8] are aware that Rich assumed h = 56.8 W/m$^2$c for his fin efficiency calculations. The authors have completely re-reduced his test data, without using this assumption. These calculations show that Rich's data are correct within +2.8/-0.4%. The present correlation uses the heat transfer coefficient values reported in Rich's original publications [4,8].

## 2. MULTIPLE REGRESSION TECHNIQUE

The j-factor correlation is developed assuming

$$j = \text{const } F(x_i) \, Re^m \tag{1}$$

where F(x) accounts for the dimensionless geometric factors, and can be written in the form

$$F(x_i)=(x_1)^{n_1}(x_2)^{n_2}(x_3)^{n_3}....(x_j)^{n_j} \tag{2}$$

Equations 1 and 2 may be written as

$$\ln j=n_1 \ln x_1+n_2 \ln x_2+n_3 \ln x_3+...n_j \ln x_j+m \ln Re \tag{3}$$

The values of the exponents are determined by a linear regression program, which uses a least squares curve fit for their evaluation. The MINITAB statistical package [5] was used to develop the present correlations.

The accuracy of the correlation was evaluated using the "R-squared value," which is also known as the"coefficient of determination." This is defined as the fraction of the variability in the logarithm of the dependent variable (e.g., the j-factor) that can be accounted for by means of the straight line regression equation (eq. 3) using the parameters selected as the predictors (e.g., the dimensionless geometric variables). The coefficient of determination is given [6] by

$$R^2 = 1 - (SSE/SST) \tag{4}$$

where SSE is the variability of the data about the regression line, and SST is the total variability of the data [6]. An R-squared value of one indicates that all the experimental values lie exactly on the regression line. Thus, the objective is to select dimensionless parameters that give the highest R-squared value. This is accomplished using a trial and error method.

Engineers frequently use the "root mean square" (rms) deviation to evaluate the validity of correlations. This is the square root of the sum of the squares of the deviation of the data points from the correlating line. The rms value will also be reported for the correlations. It is a matter of choice whether one chooses to use the "rms" or the "R-squared" method for assessing the accuracy of the correlation.

## 3. HEAT TRANSFER CORRELATION

The heat transfer correlation was developed in two steps. The first step developed a correlation for finned tube geometries having four tube rows. Then a multiplier was developed to account for the "row effect," - number of tube rows (N) less than four.

The data used to develop the 4-row correlation are listed as references 4, 7, 9 and 10 in Table 1. All data were taken in a wind tunnel having the fan downstream from the heat-exchanger (induced draft). The data of Rich [4] and McQuiston [7] are for N = 4. The McQuiston and Tree data [9] were adjusted to the original 5-row geometry tested. Kays and London [10] do not report the number of tube rows, but they are believed to have N ≥ 4. The correlation was developed assuming that the j and f-factors for N > 4 are negligibly different from that of 4-row data.

After trying different combinations of the dimensionless parameters, and evaluating their effect on the R-squared value and the rms deviation, equation 5 was selected for the 4-row j-factor correlation.

$$j_4 = 0.14 \, Re^{-0.328} \left(\frac{S_t}{S_1}\right)^{-0.502} \left(\frac{s}{D}\right)^{0.0312} \tag{5}$$

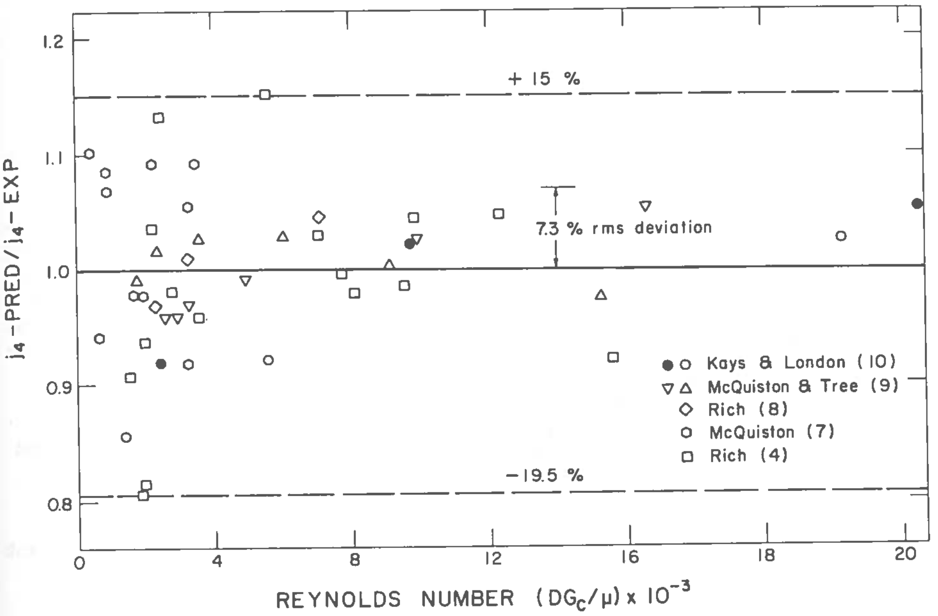


Fig. 2

Ratio of predicted (eq. 6) to experimental j-factor for N = 4 rows. Symbols defined in Table 1.

The R-squared value of equation 5 is 96.0% and the rms deviation is 7.3%. Examination of Fig. 2 shows that 89% of the data are correlated within 10%. The equation contains all of the geometric variables, except the fin thickness, which was previously discussed. The tube bank variable $S_t/S_1$ shows a strong dependence (-0.502 exponent). However, the dimensionless fin spacing s/D, has a very small influence (0.0312 exponent).

## 3.1 Comparison with Other Correlations

It is of interest to compare equation 5 with other applicable correlations. McQuiston [3] used data from references 4, 7 and 10 to develop the empirical correlation given by equation 6 for 4-row plate finned-tube heat-exchangers.

$$j_4 = 0.0014 + 0.2618\ Re^{-0.4}\left(\frac{A}{A_t}\right)^{-0.15} \quad (6)$$

The parameter $A/A_t$ is the ratio of the total surface area to the area of a bare tube bank. Equation 6 correlates the 4-row data of Table 1 with an rms deviation of 9.5%, as compared to 7.3% for equation 5.

The Briggs and Young correlation [1] is for triangular pitch banks of circular finned tubes with N ≥ 4. It is given by equation 7.

$$j = 0.134\ Re^{-0.319}\left(\frac{s}{l}\right)^{0.2}\left(\frac{s}{t}\right)^{0.11} \quad (7)$$

Equation 7 does not contain the tube bank variable, $S_t/S_1$, but it does contain an influence of the fin thickness (s/t). The fin thickness parameter is expected for individually fined tubes, because the fin thickness provides a blockage at each tube row. One may consider application of equation 7 to the plate finned tube geometry if the circular fin length (l) is such that the fin tips are touching; thus, for an equilateral triangular pitch, $l = S_t - D$. Or for an isosceles triangular pitch, $l = 0.5[\min(S_t, S_1)]$. Using these interpretations for the fin height, equation 7 was used to predict the data of Table 1. The rms error was 16%, compared to 7.3% for equation 5.

## 3.2 Correlation for the Row Effect

The reference [8] data of Table 1 were taken by Rich to determine the effect of the number of tube rows on the j-factor. Rich measured the j-factor for a plate finned-tube geometry having constant $S_t$, $S_1$, D and t for 1, 2, 3, 4, 5 and 6 tube rows. The previously described regression procedure was used to correlate Rich's data for N < 4, with results given by equation 8.

$$\frac{j_N}{j_4} = 0.991\left[2.24\ Re^{-0.092}\left(\frac{N}{4}\right)^{-0.031}\right]^{0.607(4-N)} \quad (8)$$

Equation 8 correlates Rich's data [8] with an rms deviation of 3.2%, or within +8/-4%. Some readers may be aware that Rich's original data showed the j-factor attains a maximum at $Re_L$ < 5000. Private discussions with Rich have suggested that this unexpected behavior may be due to experimental error at low air flow rates. Data for $Re_L$ < 5000 were not used in the correlation.

Equation 8 was used to predict the row effect for 1-4 rows. The predicted values and the data are shown on Figure 3. The performance decreases with increasing rows at low Re, but a cross-over occurs at Re ≅ 8000. The present authors suggest that users of equation 8 assume that the row effect is negligible for N > 4.

McQuiston [9] also developed an empirical correlation for the row effect, using Rich's data [8]. His correlation is given by equation 9, and it correlates Rich's data for N < 4 with an rms deviation of 4.3%, or within +16/-9%.

$$\frac{j_N}{j_4} = \frac{1 - 1\,280\ N\ Re_L^{-1.2}}{1 - 5\,120\ Re_L^{-1.2}} \quad (9)$$

The ability of equation 5 to predict data not used in development of the correlation was tested

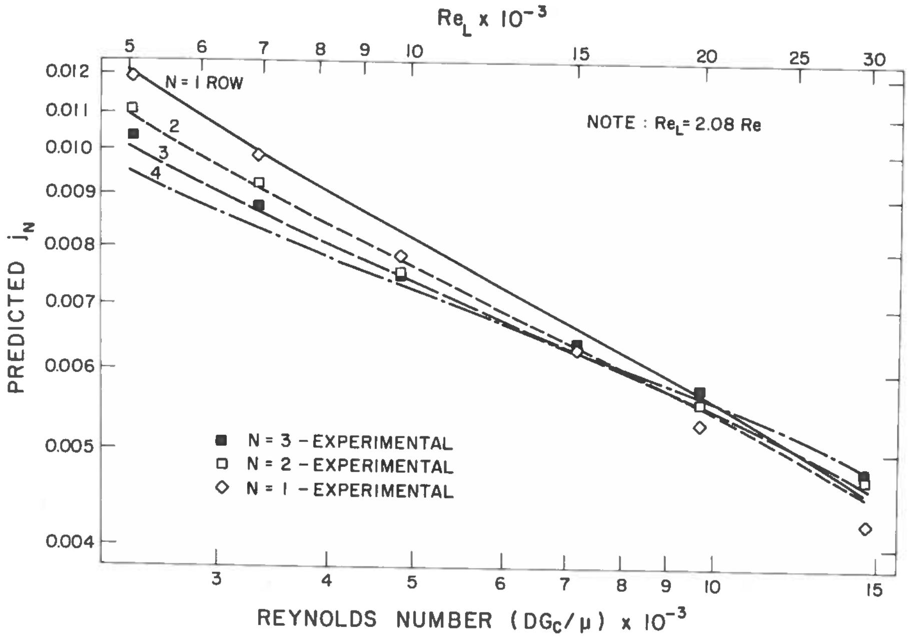


Fig. 3. Predicted j-factor (eq. 9) for $N \leq 4$.

using data of Cox and Jallouk [11], and of Vampola [12]. The geometries tested by Cox and Vampola are listed in Table 1. The 8-row Cox data were predicted within a maximum deviation of 5.9% of the experimental values using equation 5, which assumes there is no change of the heat transfer coefficient between four and eight rows. The 5-row Vampola heat-exchanger is made with five individual rows, each separated by a 4.0 mm space. These data were predicted within 11% of the experimental value using equation 5.

## 4. PRESSURE DROP CORRELATION

To date, the only correlation available for prediction of the friction factor of the plate finned-tube geometry is provided by McQuiston [3]. However, McQuiston's empirical correlation is quite poor, it "correlates most of the data within +35/-35%." Hence, there is need for a more accurate friction correlation.

### 4.1 Basis of the New Correlation

The friction correlation is based on a superposition model, that was initially proposed by Rich [4]. The basic model is written as

$$\Delta P = \Delta P_f + \Delta P_t \qquad (10)$$

where $\Delta P_t$ is the pressure drop component due to the drag force on the tubes, and $\Delta P_f$ is the pressure drop component due to friction on the fins. One may write equations for $\Delta P_t$ and $\Delta P_f$ as

$$\Delta P_f = f_f \frac{A_f}{A_c} \frac{G_c^2}{2\rho} \qquad (11)$$

$$\Delta P_t = f_t \frac{A_t}{A_{c,t}} \frac{G_c^2}{2\rho} \qquad (12)$$

The mass velocity (G) used in equation 11 is evaluated at the same G that exists in the finned-tube exchanger. Several correlations are available for the friction factor of flow normal to tube banks. This work used the Zukauskas correlation which may be found in most heat transfer text books. The present authors have compared the Zukauskas correlation [13] with the Jakob correlation [14] for the Re, $S_t$, and $S_1$ of interest here (1.5 to 2.6). the two equations agreed within +6/-6%.

Data reported in References 4, 7, 9, 10 and 11 were correlated using equations 11 and 12. First, the Zukauskas correlation was used to obtain the bare tube bank friction factor ($f_t$) and calculate $\Delta P_t$ using equation 12. Substituting equation 11 and 12 in equation 10 and solving for $f_f$ gives

$$f_f = \frac{A}{A_f} \left[f - f_t \left(1 - \frac{A_f}{A}\right) \left(1 - \frac{t}{p_f}\right)\right] \qquad (13)$$

### 4.2 The Pressure Drop Correlation

The previously discussed linear regression analysis technique was used to correlate $f_f$ as a function of the dimensionless geometric variables. The best correlation was provided by equation 14.

$$f_f = 0.508 \; Re^{-0.521} \; (S_t/D)^{1.318} \qquad (14)$$

The friction factor of the finned tube heat-exchanger is obtained by substituting equations 11 and 12 in equation 10 and solving for f. The result is

$$f = f_f \frac{A_f}{A} + f_t \left(1 - \frac{A_f}{A}\right) \left(1 - \frac{t}{p_f}\right) \qquad (15)$$

Figure 4 shows the ability of the correlation to predict the heat-exchanger friction factor. The $f_f$ is calculated by equation 14, and $f_t$ was obtained from the Zukauskas correlation for bare tube banks. The Zukauskas friction factor ($f_{tz}$) is defined differently than the definition of equation 12 ($f_t$). The relation between the two definitions is given by equation 16,

$$f_{tz} NX = f_t \frac{A_t}{A_{c,t}} = f_t \frac{\pi D}{S_t - D} \qquad (16)$$

where X is an empirical function of Re and $S_t/S_1$;

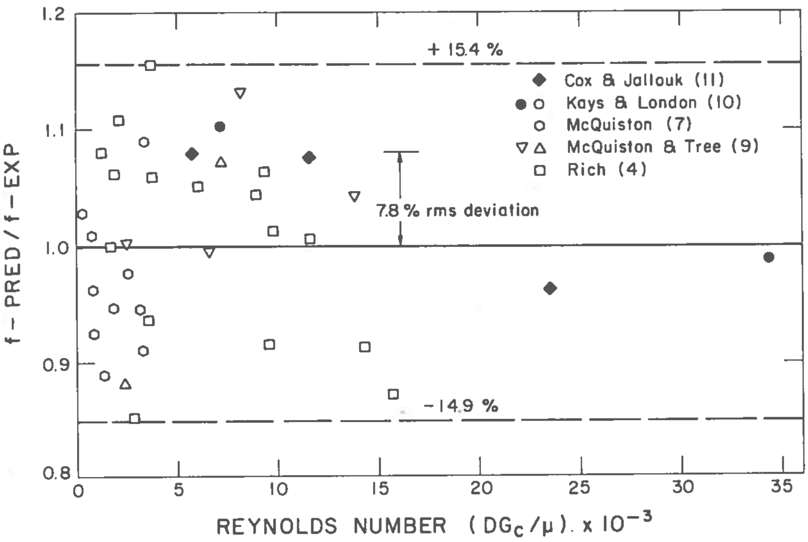


Fig. 4

Ratio of predicted-to-experimental friction factor using the present correlation (eq. 15). Symbols defined in Table 1.

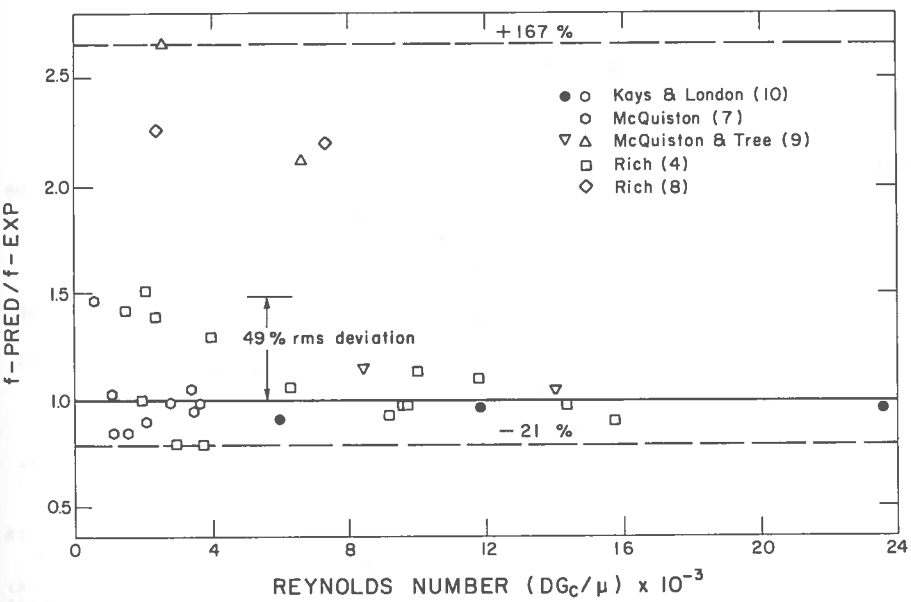


Fig. 5

Ratio of predicted-to-experimental friction factor using McQuiston's correlation. Symbols defined in Table 1.

it is not given here, because of space limitations.

Figure 4 shows the friction factor correlated using equations 14 and 15. The rms deviation is 7.8%, and 95% of the data are correlated within ±13%.

Figure 5 shows the ability of McQuiston's correlation [3] to correlate the same data shown in Figure 4. McQuiston's correlation is given by equations 17.

$$f = 0.0004094 + 1.382\, F_1^{-0.8}\, F_2^{-1} \left(\frac{D}{D^*}\right)^{0.5} Re^{-0.5} \qquad (17a)$$

$$F_1 = \frac{(S_t - D)}{4(p_f - t)} \qquad F_2 = \frac{S_t}{D^*} - 1 \qquad (17b)$$

$$\frac{D^*}{D} = \frac{A}{A_t}\, \frac{P_f}{(S_t - D) + P_f} \qquad (17c)$$

Figure 5 does not include the Kays and London data [10] of Table 1 for 7.75 fins/in. McQuiston's correlation yields a negative friction factor for these data. The rms deviation on Figure 5 is 49.1%, and the maximum error limits are +167/-21%.

## 5. RANGE OF VALIDITY OF THE CORRELATIONS

Because the present correlations are empirical, it is important to establish the range of dimensionless parameters used in developing the correlations. The range of dimensionless variables are $500 < Re < 24\,700$ (equation 5); $2\,400 < Re < 24700$ (Equation 8); $1.97 < S_t/D < 2.55$; $1.70 < S_l/D < 2.58$; $0.08 < s/D < 0.64$; $1 < N < 8$ or more.

## 6. CONCLUSIONS

Empirical linear regression equations have been developed to predict the heat transfer coefficient and friction factor of plate finned-tube heat exchangers having plain fins.

The heat transfer correlation (equation 5) is valid for 4 or more rows, has an rms error of 7.3%. The correlation assumes that the heat transfer coefficient does not change after 4-rows. Equation 8 is a multiplier to account for the row effect when $N < 4$.

The friction correlation has an rms deviation of 7.8%, and 95% of the data are correlated within +13/-13%.

Users are cautioned to be careful in extrapolating the correlations to conditions beyond the range of dimensionless variables used to develop the correlations.

## 7. NOMENCLATURE

| | |
|---|---|
| $A$ | air-side surface area (fins and tubes), $(m^2)$ |
| $A_f$ | surface area of fins $(m^2)$ |
| $A_t$ | surface area of bare tube, $(m^2)$ |
| $A_c$ | minimum flow area, $A_{c,t}$ (for bare tube bank), $(m^2)$ |
| $D$ | tube diameter, (m) |
| $D_h$ | hydraulic diameter ($D_h = 4LA_c/A$), (m) |
| $f$ | Fanning friction factor (dimensionless) |
| $f_f$ | friction factor associated with fin area (dimensionless) |
| $f_t$ | friction factor associated with bare tube area |
| $G_c$ | mass velocity based on flow area $A_c$, $kg/m^2s$ |
| $h$ | heat transfer coefficient, $(W/m^2\text{-}C)$ |
| $j$ | Colburn j-factor, $StPr^{2/3}$ |
| $j_4$ | j-factor for 4-row heat exchanger, (dimensionless) |
| $j_N$ | j-factor for N-row heat exchanger, (dimensionless) |
| $L$ | air exchanger flow length (m) |
| $l$ | circular fin height, (m) |
| $N$ | number of tube rows in flow direction |
| $\Delta P$ | pressure drop across heat exchanger (Pa) |
| $\Delta P_f$ | pressure drop assignable to fin area, (Pa) |
| $\Delta P_t$ | pressure drop assignable to bare tube bank area (Pa) |
| $p_f$ | Fin pitch (s + t), (m) |
| $Pr$ | Prandtl number (dimensionless) |
| rms | root mean square deviation, (dimensionless) |
| $R^2$ | Coefficient of determination, defined by Eq. 4, (dimensionless) |
| $Re$ | Reynolds number, $L_cG_c/\mu$, Re for $L_c = D$, $Re_D$ for $L_c = D_h$, $Re_L$ for $L_c = S_1$, (dimensionless) |
| $s$ | spacing between adjacent fins, (m) |
| $S_t$ | tube spacing normal to flow (m) |
| $S_1$ | tube spacing in air flow direction |
| $S$ | $[(S_t/2)^2 + (S_1)]^{2\ 0.5}$ (m) |
| $St$ | $h/G_cc_p$ (dimensionless) |
| $t$ | fin thickness, (m) |
| $\chi$ | Term in equation 16, $X = X(Re, S_t/S_1)$ |

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