Finned Tube Heat Exchangers: State of the Art for the Air Side

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Abstract

The findings of various investigators during the past ten years has greatly enhanced our ability to analyze, model design, and optimize the finned tube heat exchangers so widely used in the HVAC business. The effect of rows, fin pitch, and other geometric parameters are now well understood and generalized heat transfer correlations are available. The relation between heat and mass transfer has been much better defined and good correlations have been developed. Some advances have also been made in modeling techniques especially where partially dry cooling and dehumidifying coils are involved. All of these factors will be reviewed to concisely summarize what is now available to the heat exchanger analyist.

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Introduction

Interest in the design and optimization of finned tube heat exchangers was revived in about 1968 after some ten to fifteen years of limited activity following the work of Kays and London. Escalating manufacturing costs, the need for increased compactness, and lack of mass transfer data were responsible for this renewed emphasis. As a result of rather intense activity, the findings of various investigators during the past ten years has greatly enhanced our ability to analyze, design, model, and optimize the finned tube heat exchangers so widely used in the HVAC & R business. The effect of tube rows, fin pitch and other parameters are now well understood and generalized heat, mass, and momentum transfer correlations are available. The relation between heat and mass transfer has been better defined and surface effects have been investigated. Some advances have also been made in modeling techniques, especially with respect to partially dry cooling and dehumidifying coils. All of these factors will be reviewed beginning with an optimization study initiated in 1968.

Review of Literature

An ambitious project was undertaken by the author (1) to define the parameters which would optimize the finned tube heat exchanger with respect to space and weight under both heat and mass transfer conditions. To do this, generalized correlations for the heat, mass, and momentum transfer coefficients were required. It immediately became clear that very little data were available and practically no general correlations could be found. In fact experimental data for only two plate fin tube surfaces could be located. The scope of the initial effort was therefore reduced to cover only sensible heat transfer and considerable effort was expended to test two surfaces (2) and develop simplified general correlations for the heat transfer and friction coefficients. Figures 1 and 2 show these correlations. The data of Rich (3) was not available at the time of this work, but has been added for comparison. These correlations were not considered to be entirely satisfactory, but the absence of data made further development futile. The results of this project showed that tube diameter and fin spacing should be constrained to as small a value as possible; that fin thickness is a weak parameter and can be as small as possible; and for a given tube diameter there is an optimum fin diameter to give maximum heat transfer per unit of coil volume, face area, or weight. Figure 3 shows results for minimum volume. This work is still considered to be valid although the correlations used were not as refined as desired.

A paper by Rich (3), published in 1973, focused attention on the effect of fin spacing on the heat transfer and friction factors for one plate fin tube geometry. Fin pitch was systematically varied from zero to about 20 fins per inch with data collected over a wide operating range. A significant result of this work was the fact that the j factors and frictions factors could all be correlated quite well using a Reynolds number

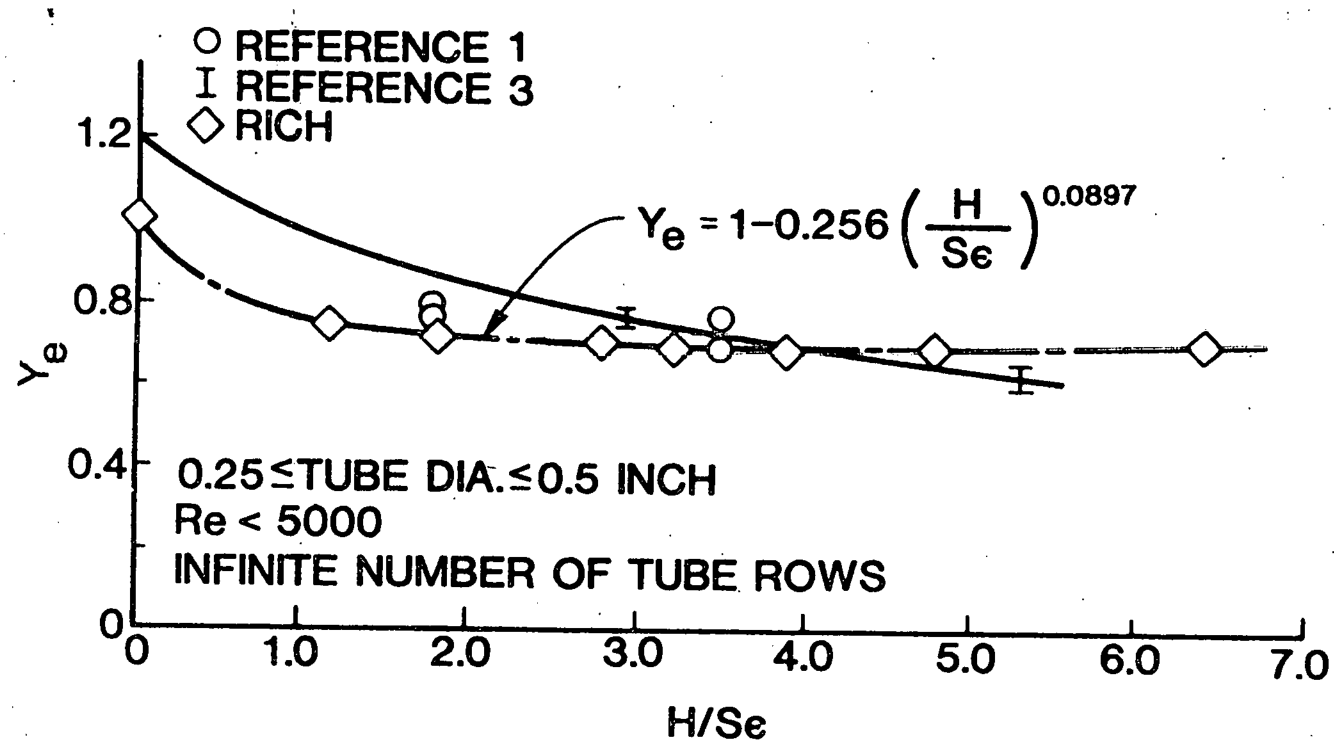

Figure 1. Heat Transfer Correlation for Finned Tube Heat Exchangers

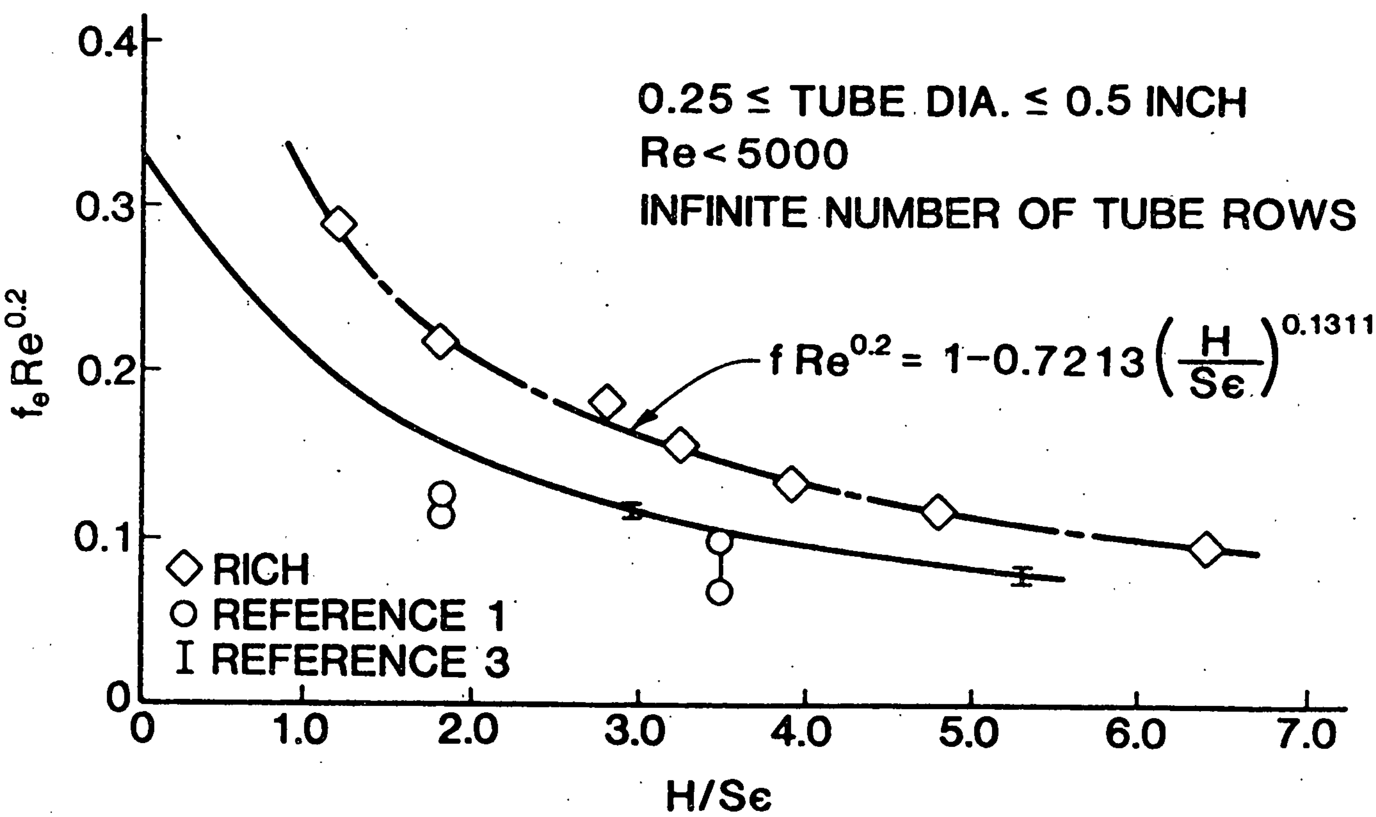

Figure 2. Friction Factor Correlation for Finned Tube Heat Exchangers

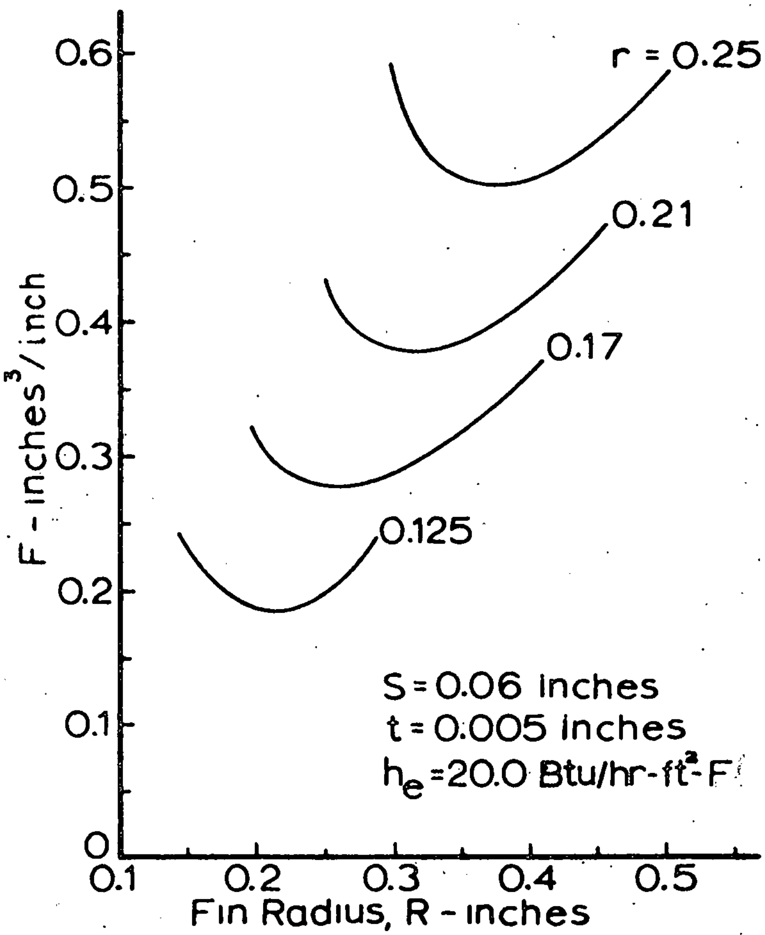

Figure 3. Effect of Tube and Fin Radii on Volume per Unit of Heat Transfer

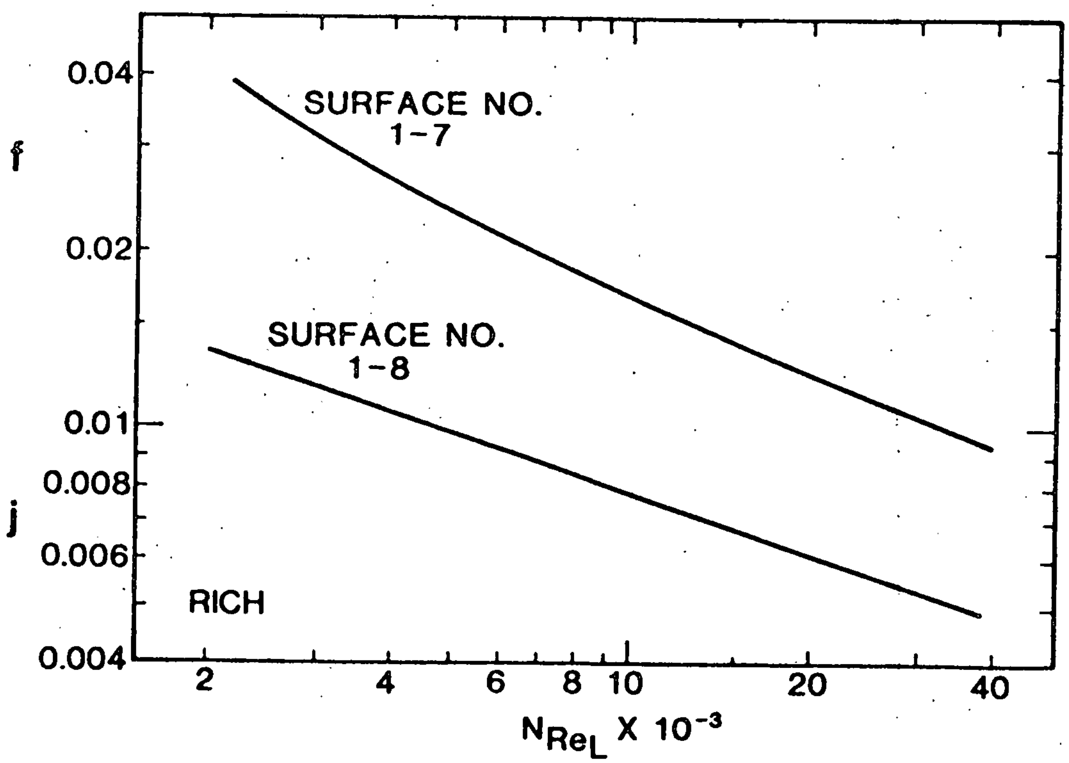

Figure 4. Correlation of Heat Transfer and Friction Results (RICH3)

based on the tube row spacing with fin pitch varying from about three to 15 fins per inch, figure 4. Although these results are for only one tube size and layout (0.5 in; 1.25 in x 1.25 in.) the Reynolds number based on tube row spacing should correlate other similar geometries equally well. Rich then proceeded to study the effect of tube rows.

The results of the study of tube row effects were published in 1975 (4). In this case one coil geometry indentical to that of reference 3 with 14 fins per inch was tested with the number of tube rows varying from one to six. One coil with four rows of tubes was tested and the performance of each row was determined. These very interesting and useful results are shown in figures 5 and 6. The average heat transfer coefficient is shown to decrease as the number of rows increase up to a Reynolds number (based on tube row spacing) of about 15,000 then the effect reverses. However, the usual range of operation is at the lower Reynolds numbers. It should be noted that this behavior is opposite to that of bare tubes where the effect of increased tube rows is to increase the average heat transfer coefficient. These results are significant because they show that the tube rows at the coil exit are not very effective in a deep coil. In the case of dehumidifying coils, the higher heat transfer rate experienced by the first row leads to a high surface temperature, reduced mass transfer, and often no mass transfer at all in this part of the coil. It can generally be assumed that the number of tube rows will affect the mass transfer coefficient in the same way as the heat transfer coefficient since they are related; however, the friction factor does not seem to be influenced very much by the number of tube rows. The literature is vague on this point. Rich (4) did not investigate the effect of rows on friction performance. The assumption is that since a contraction and expansion of the air stream occurs for each row, the friction factor is the same for each row. The last row actually has a larger friction factor but the difference is small

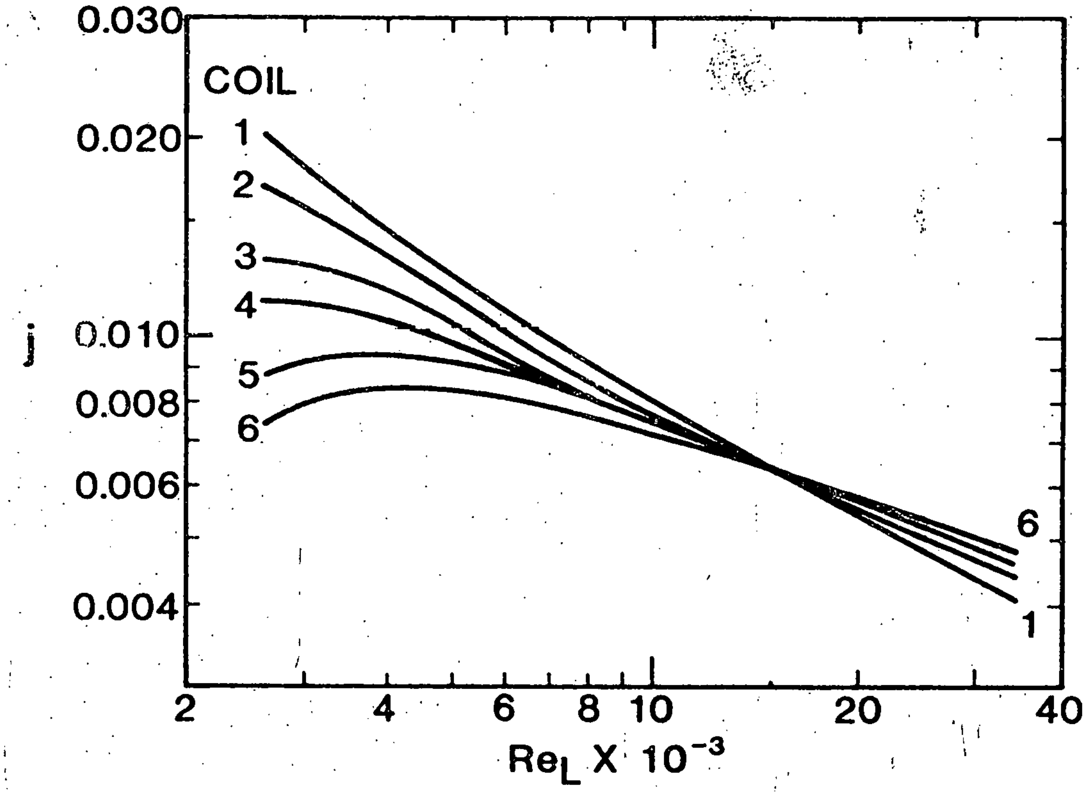

Figure 5. Average Heat Transfer Performance for Coils (RICH 4) with 1-6 Rows of Tubes

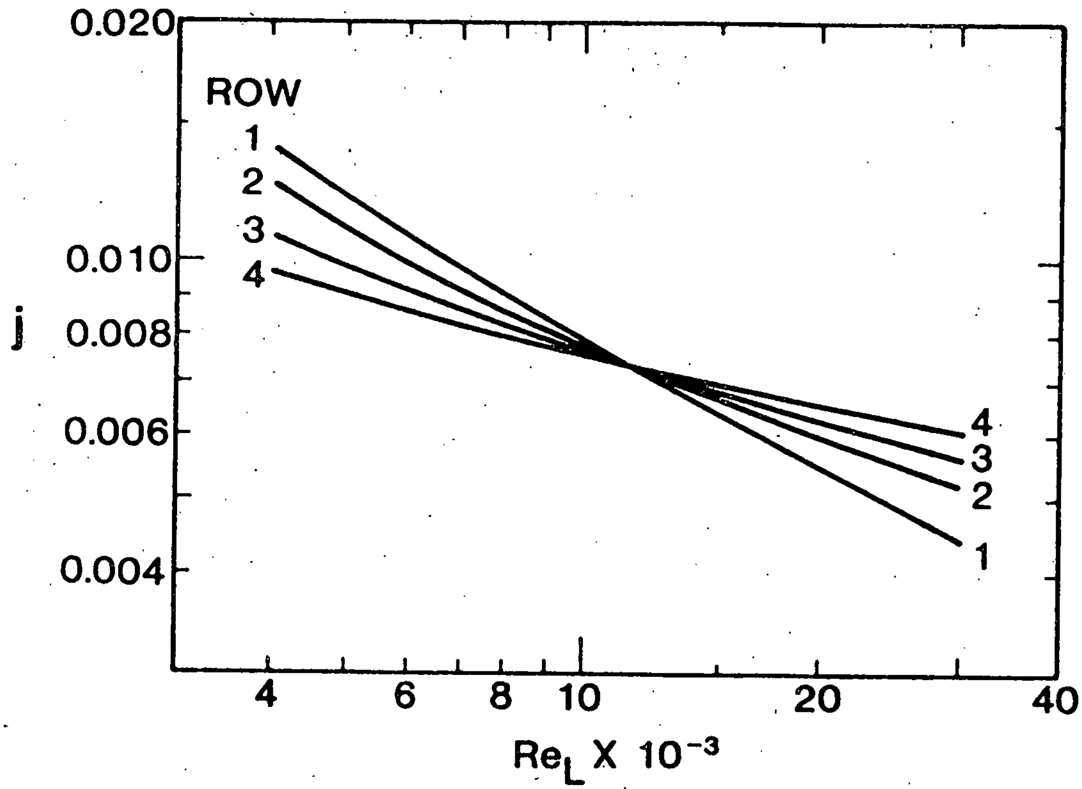

Figure 6. Summary of Heat Transfer Results for Individual Rows of a 4 Row Coil.

for the tube diameters and tube layouts of interest here. While Rich was studying fin pitch and row effects the author decided to pursue the question of the validity of the Colburn, j factor analogy given by

$$j_i = h_d(Sc^{2/3})/G_c = j = h(Pr^{2/3})/(G_c\ C_p) \tag{1}$$

where

h_d = mass transfer coefficient

h = heat transfer coefficient

Sc = Schmidt number

Pr = Prandtl number

G_c = mass velocity in the coil

c_p = specific heat capacity

Results of a simple (5) study involving a parallel plate exchanger suggested that the analogy was very poor when dry surface and wet surface j factors were compared. A much more refined investigation was then undertaken with the objective of determining the relationship between the dry surface sensible heat transfer and friction performance and the wet surface heat, mass, and friction performance of plate fin tube exchangers. The initial phase concentrated on the basic nature of the process and the effect of surface conditions (6). Figure 7 shows some of the results of this study. The presence of condensate on the surface influences the transport process. The clean surface corresponds to a film of water on the surface while the nonwetting surface was covered with droplets of water. The water on the surface has a roughness effect which is important when the Reynolds number is high and the boundary layer is thin, but has little effect when the Reynolds number is low and the boundary layer is thick. Note that the droplets have a greater effect than the film. These results prompted a continued study of surface conditions and the testing of 5 different plate fin tube coils with various fin spacings and under dry,

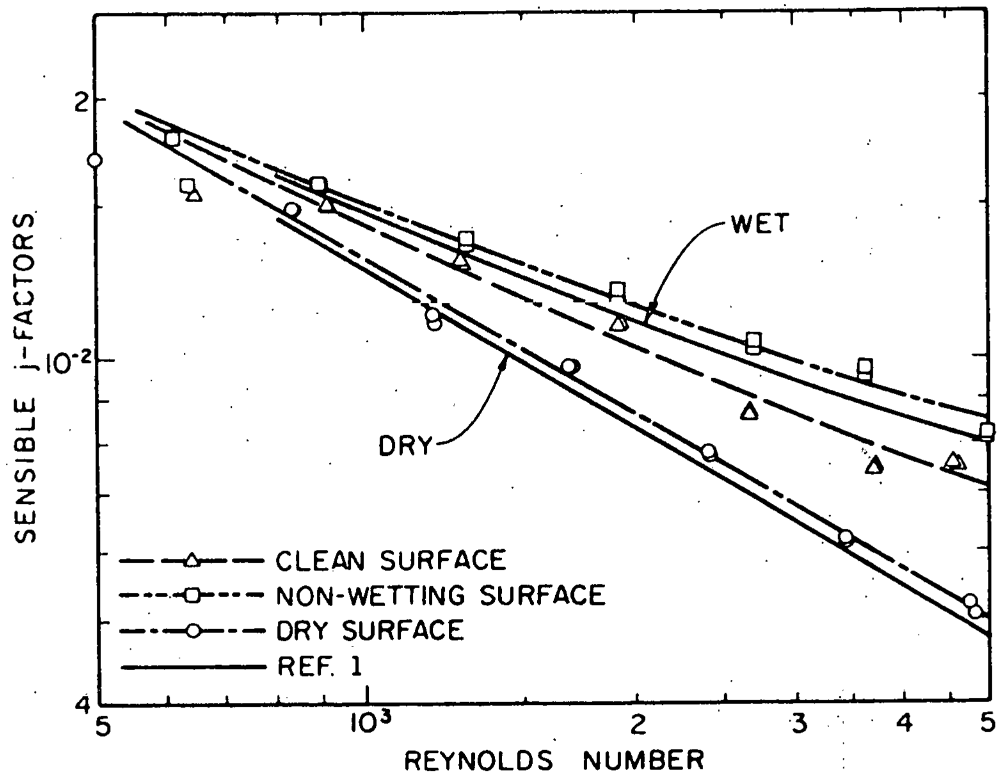

Figure 7. Sensible Heat Transfer j-Factors for the Parallel Plate Exchanger

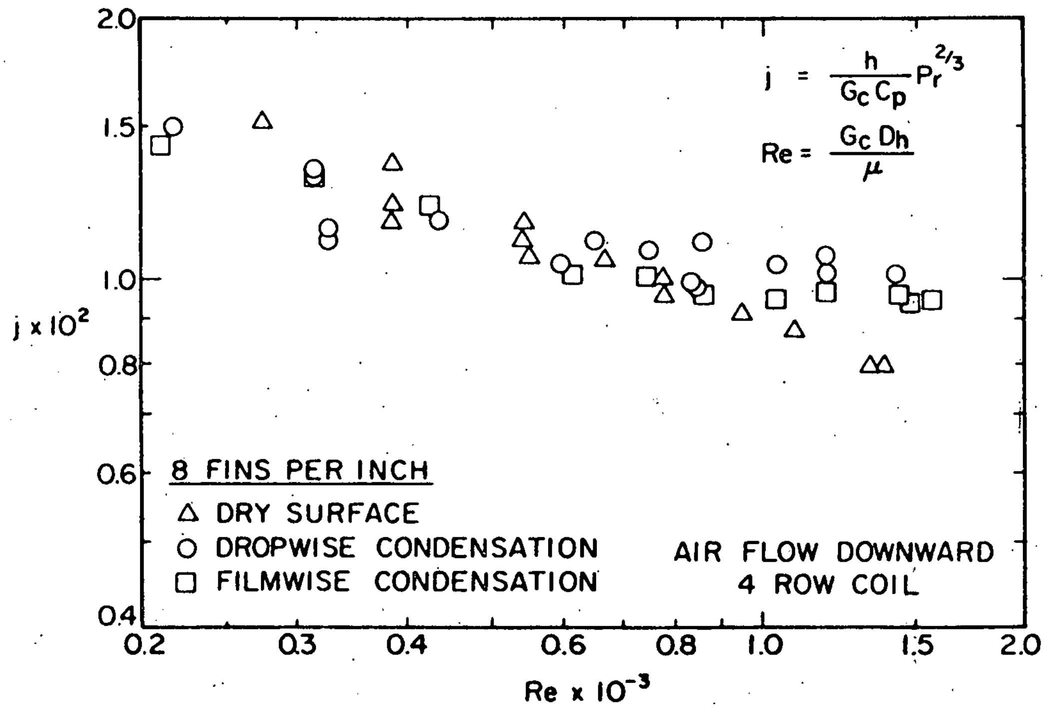

Figure 8. Sensible j Factors for the Plate-Fin-Tube Heat Exchanger with 8 Fins per Inch. Air Flow Downward; 4 Rows of Tubes

dry, film, and drop surface conditions (7). The coils were typical copper tube-aluminum fin exchangers. Extensive bench tests revealed that the aluminum fins would not operate in the film condensing mode following conventional degreasing methods, but exhibited drops instead. However, when the coil was cleaned using a harsh detergent and steam or boiling water, stable film condensation occured. Figures 8, 9, and 10 show the sensible j factors, total j factors, and the friction factors for the coil with 8 fins per inch. The effects of the moisture on the surface is evident. Also note that the surface effect increases with increasing Reynolds number. Similar data were obtained for exchangers with 4, 8, 10, 12, and 14 fins per inch. It should be noted that in reducing the raw data for the above study, the effect of the number of tube rows was considered. The curves of figure 11 were developed from the results of Rich (4) for this purpose. This approach predicted higher surface temperatures at the coil inlet and lower surface temperatures at the coil outlet than those obtained using average transport coefficients at both inlet and outlet. This in turn alters the enthalpy driving potential and the derived mass transfer coefficient. The fin efficiency concept of reference 8 which accounts for mass transfer was also used in the data reduction. As a result of all these refinements, the data reported in reference 7 are thought to be quite accurate. However, the j versus Reynolds number representation of the data is grossly inadequate to relate heat and mass transfer. It is also desirable to have correlations to account for variations in geometry to avoid many additional tests.

Extensive review of the literature (9) revealed that some investigators in the process industry were able to correlate data for large circular fin air coolers using the Reynolds number based on tube diameter and ratio of the total surface area to the surface area of the bare tubes.

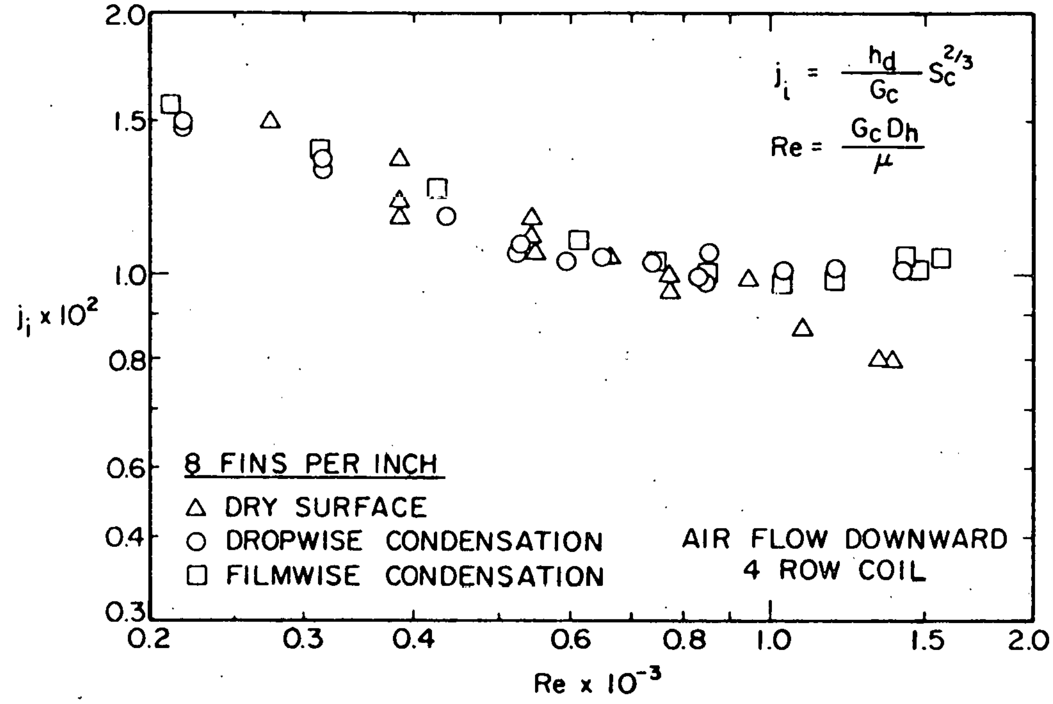

Figure 9. Total j Factors for the Plate-Fin-Tube Heat Exchanger with 8 Fins per Inch. Air Flow Downward; 4 Rows of Tubes

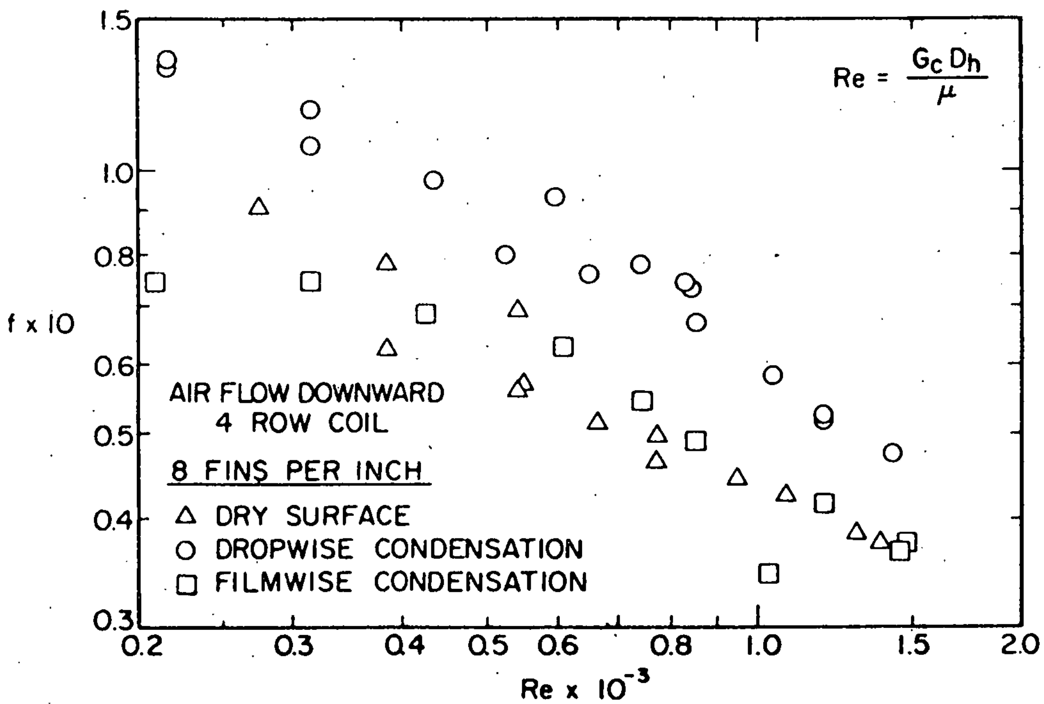

Figure 10. Friction Factors for The Plate-Fin-Tube Heat Exchanger with 8 Fins per Inch. Air Flow Downward; 4 Rows of Tubes

The parameter is defined as

$$JP = Re^{m}_{D}(A/A_t)^n \tag{2}$$

Note that the ratio A/A_t is equal to one for a bare tube bank and for this case m is about -0.4 at low Reynolds numbers. The parameter A/A_t contains all the geometric parameters of the coil except the number of tube rows. The final form of the correlation is shown in figure 12 where the data of reference 5 are plotted. Additionally, the data of references 2 and 3 are shown to fit nicely. Coils with tube diameters ranging from 3/8 inch to 5/8 inch; tube patterns ranging from 1 inch x 1 inch to 1 3/4 inch x 1 3/4 inch; and fin pitch ranging from three to 20 fins per inch are correlated. However, the correlation is for four rows of tubes.

Analysis of the data shown in figure 5 which shows the tube row effect resulted in the following correlation

$$j_n/j_1 = 1 - 1280\, N_r\, Re_b^{-1.2} \tag{3}$$

where

N_r = number of rows of tubes

j_n = j factor corresponding to N_r

j_1 = j factor for a 1 row coil

Re_b = Reynolds number based on tube row spacing

Since the correlation of figure 12 is based on four row coils, equation 3 can be modified to obtain

$$j_n/j_4 = \frac{1 - 1280\, N_r\, Re_b^{-1.2}}{1 - (4x1280)\, Re_b^{-1.2}} \tag{4}$$

where j_4 is found from figure 12.

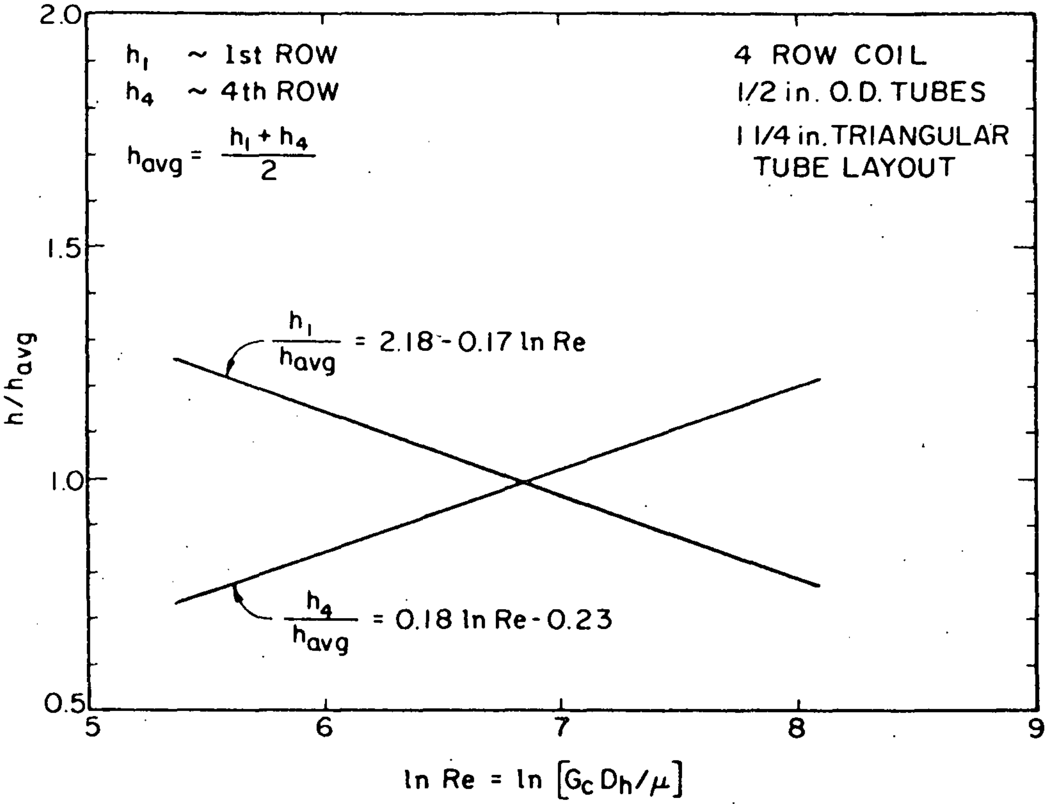

Figure 11. The Influence of Tube Rows on the Heat Transfer Coefficient (adapted from reference 3)

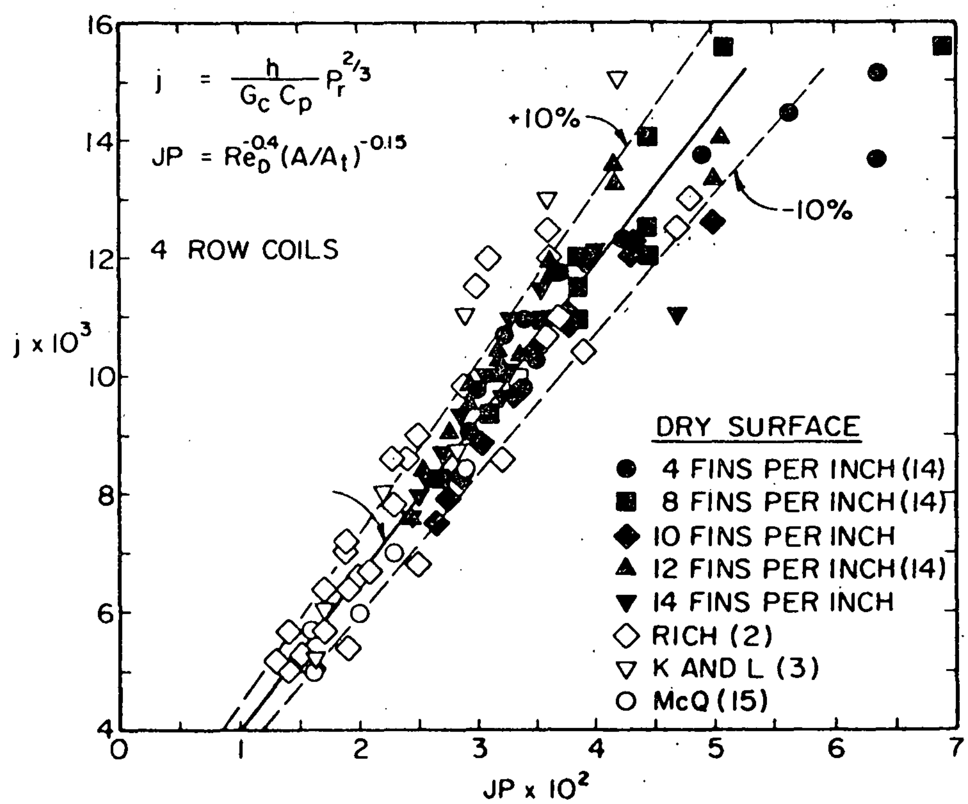

Figure 12. Correlation of Dry Surface Heat Transfer Data

The friction data of references 2, 3, and 7 were correlated in a similar way using a parameter defined as follows

$$FP = Re_D^{-0.25} \left(\frac{R}{R^*}\right)^{0.25} \left[\frac{(Xa-2R)Ps}{4(1-Ps\ y)}\right]^{-0.4} \left[\frac{Xa}{2R^*} - 1\right]^{-0.5} \quad (5)$$

where

$$\frac{R}{R^*} = \frac{(Xa - 2R)Ps + 1}{(A/A_t)} \quad (6)$$

and

R = effective fin radius

Xa = tube spacing normal to the air flow

Ps = fin pitch

y = fin thickness

Figure 13 shows the friction factor f plotted versus FP, the data of reference 7 correlate very well with the data from all sources scattered somewhat. Friction data from different sources are usually difficult to correlate. With the dry sensible heat tranfer and friction coefficients correlated reasonably well, new parameters to modify JP and FP were sought to relate the wet surface sensible and total j factors and friction factors to the dry surface values.

The wet surface effect seemed to depend strongly on the fin spacing and Reynolds number (reference 7). Therefore, two new parameters were devised. These were a Reynolds number based on fin spacing Re_s and the ratio of fin spacing to distance between the fins, F_s. Correlating parameters of the form

$$j_{wet} = fcn[(JP)J(s)]$$
$$f_{wet} = fcn[(FP)F(s)] \quad (7)$$

were developed. In the case of the film type surface condition the parameters were as follows:

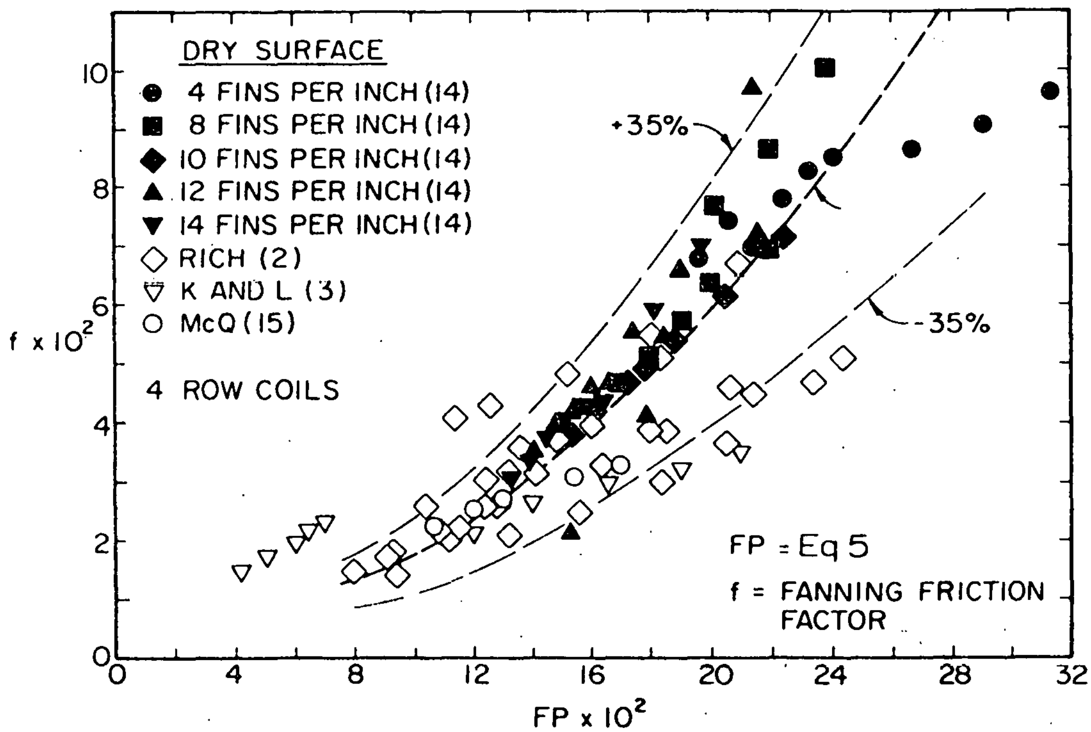

Figure 13. Correlation of Dry Surface Friction Data

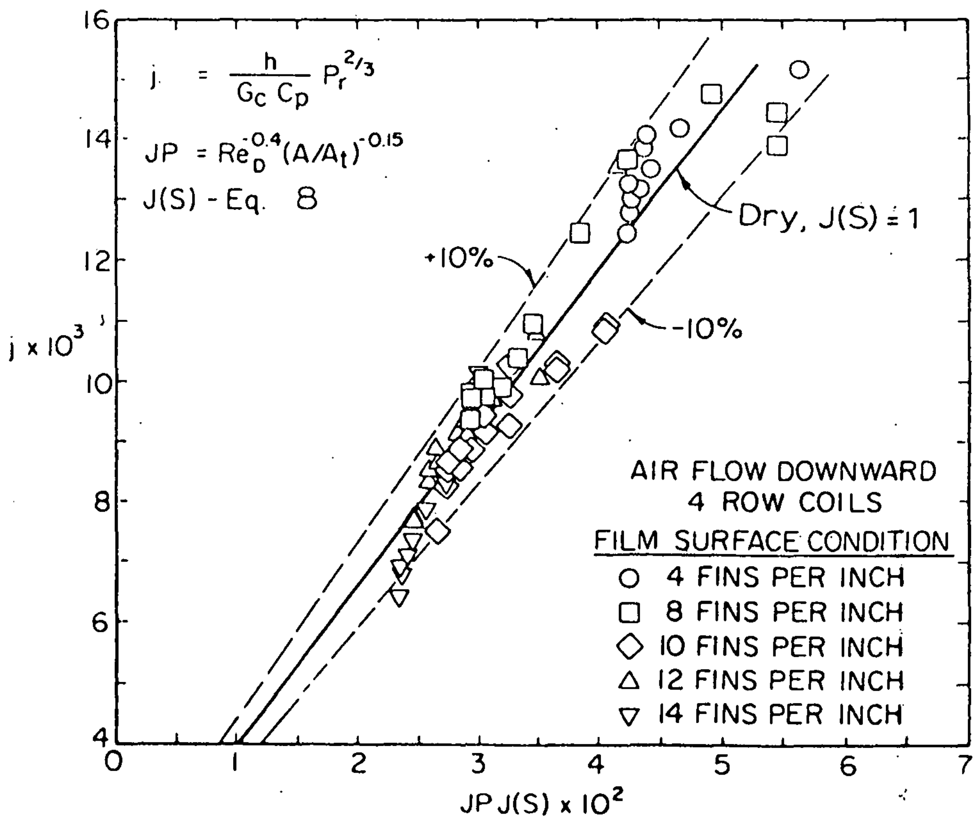

Figure 14. Sensible j Factors Correlated by J(s)

Sensible j Factor

$$J(s) = 0.84 + 4.0 \times 10^{-5} Re_s^{1.25} \quad (8)$$

Total j Factor

$$J(s) = [0.95 + 4 \times 10^{-5} Re_s^{1.25}] F_s^2 \quad (9)$$

Fanning Friction Factor

$$F(s) = [1 + Re_s^{-0.4}] F_s^{1.5} \quad (10)$$

Figures 14, 15, and 16 show all the wet surface data from reference 7. With the exception of the 14 fin per inch coil, all the data correlates reasonably well. Note that J(s) and F(s) are equal to one when the surface is dry. Therefore the correlations of figures 14, 15, and 16 may be used to obtain all the transport coefficients for any smooth plate fin tube coil operating either wet or dry.

Discussion

The generalized correlations described above provide excellent tools to investigate the performance, optimize, and design finned tube heat exchangers. Strictly speaking, the correlations shown are for smooth plate fin tube coils. Experience has shown, however, that circular fins, wavy fins, and other such variations correlate in the same general way with a slight displacement to the right or left and a small change in slope. Therefore, with a few data points from dry sensible tests, acceptable correlations can be developed for a whole family of coils with similar geometric characteristics. It must be assumed that the wet surface performance will follow the same trends as given for the smooth plate finned tube coils. This approach has been found to work well for smooth circular fins and corrugated circular fins where only four data points were available for the j and f factors.

Practical use of the forgoing correlations in heat exchanger analysis

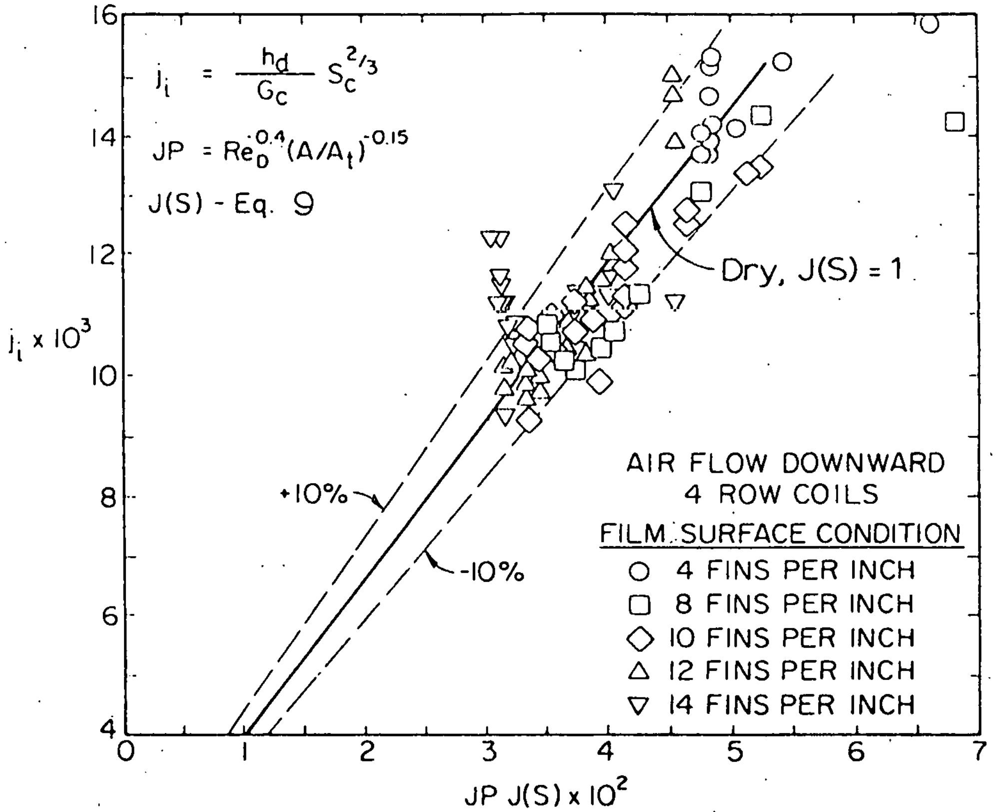

Figure 15. Total j Factors Correlated by J(s)

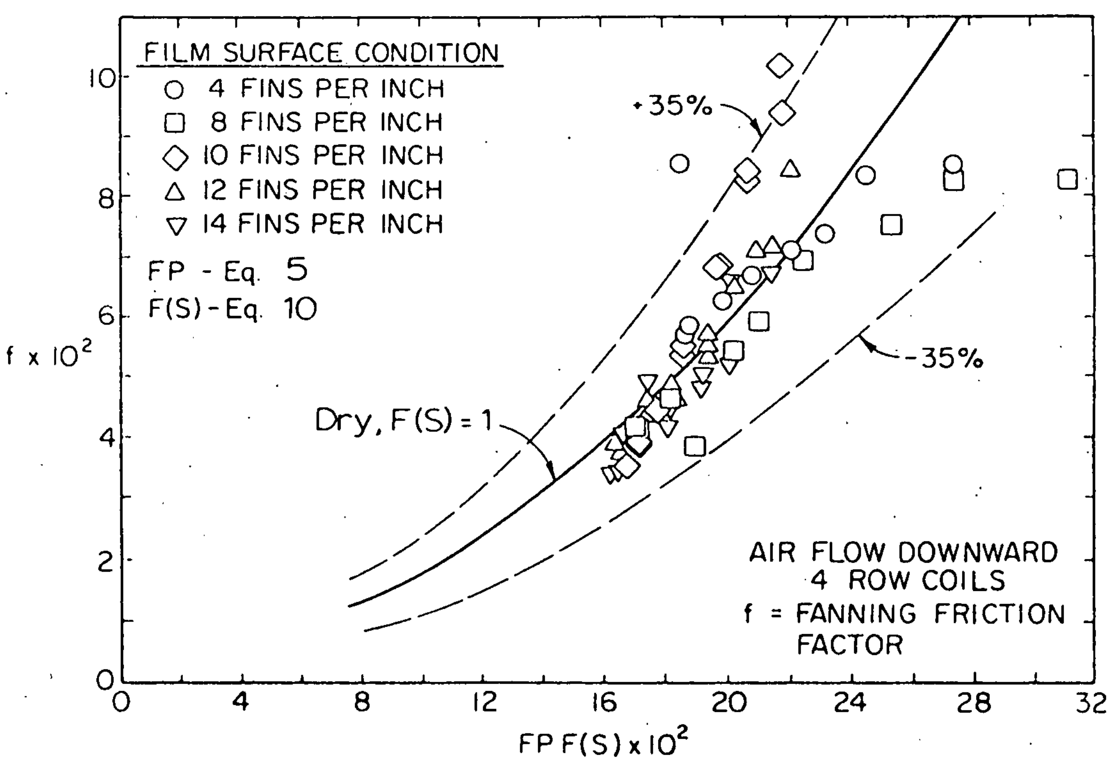

Figure 16. Friction Factors Correlated by F(s)

requires a computer, especially when mass transfer is occuring. This is because iteration is required. When only sensible heat transfer is occuring, the calculation procedure may be the classical LMTD or NTU-ε methods using the overall heat transfer coefficient U. In the case of simultaneous heat and mass transfer a different approach is required because the latent heat transfer due to moisture condensation depends on the surface temperature and not the water temperature directly. An iterative procedure may be used to compute the coil surface temperature at the inlet and outlet taking into account the difference in heat and mass transfer rates at inlet and outlet as discussed in the literature review (4). With these temperatures and the entering and leaving moist air conditions the driving potential (log mean enthalpy difference) is established and the total heat transfer rate may be computed. The sensible heat transfer is computed in the usual way, but a U factor based on wet surface conditions is used. In the course of carrying out the iterations suggested above the computed surface temperatures may be used to check for a partial or totally dry coil as follows:

1. Surface Temperature at inlet less than dewpoint temperature indicates a completely wetted coil.
2. Surface Temperature at exit greater than dewpoint temperature indicates a completely dry coil.
3. Surface Temperature at inlet greater than dewpoint temperature indicates a partially dry coil.

Figures 17 and 18 illustrate the conditions for a partially dry coil which is the most troublesome to analyze because it is necessary to determine the location in the coil where condensation begins and the air temperature at that point. Again this is accomplished by iteration. Using the known dewpoint temperature and humidity ratio and by checking the energy balances between the air and refrigerant for both the dry and wet portions of the coil, the location where condensation begins may be found. This iteration

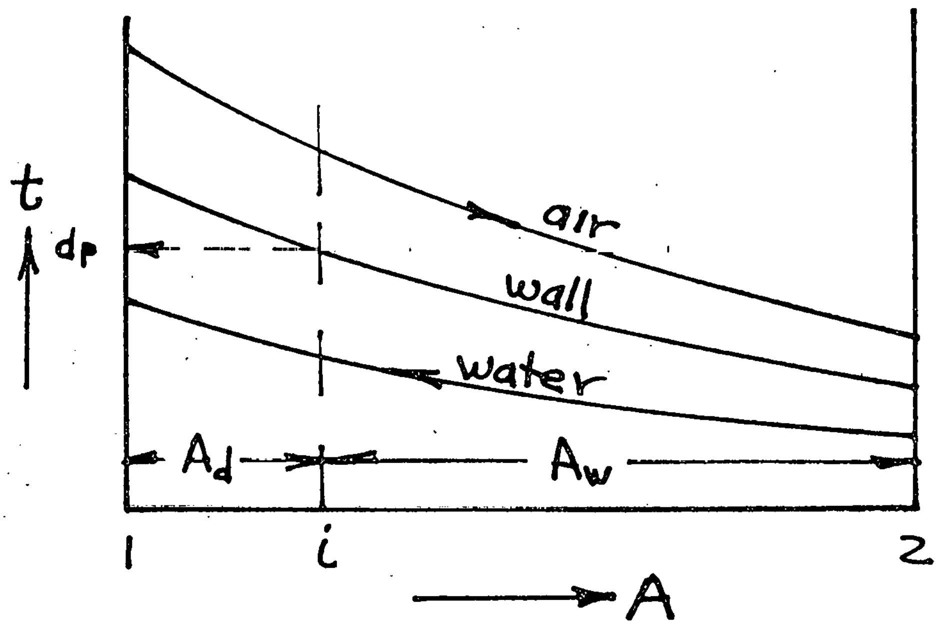

Figure 17. Temperatures in a Partially Wet Dehumidigying Coil

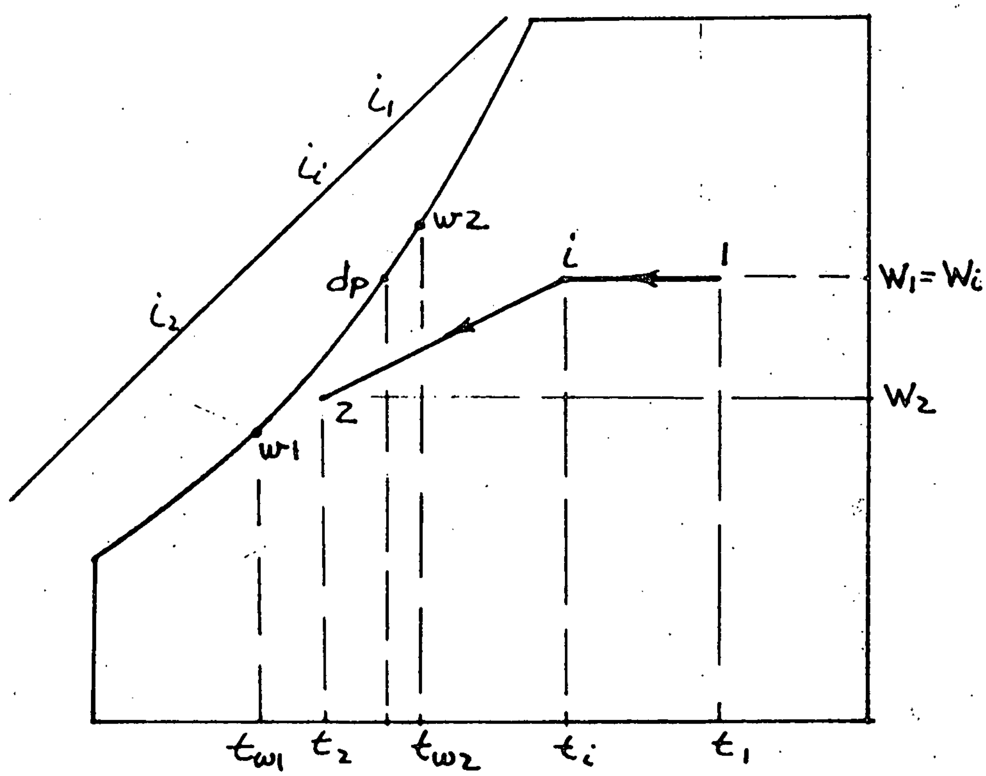

Figure 18. Psychrometric Process for a Partially Wet Dehumidifying Coil

is carried on each time the overall iteration for the coil exit conditions is done. At convergence all energy balances must be satisfied. This general approach has been used successfully for typical air conditioning type calculations as well as cases where air temperatures and pressures were rather extreme. In a number of cases computed and actual performance have been compared and found to agree within the accuracy of the general correlations.

Summary

Recent developments in the correlation of heat, mass, and momentum transport data for finned tube heat exchangers have been reviewed. It was shown that the effects of all geometric parameters on performance are well understood. Surface effects have been much better defined and the relation between heat and mass transfer is fairly well understood. However, with all this progress there still are a number of assumptions to be questioned. The assumption that the number of tube rows affects mass transfer in the same way as heat transfer should be verified. It may also be true that the number of tube rows has an effect on the friction factor, especially when the coil is wet. Additional research should also be done to better define the parameters which relate wet surface and dry surface performance. The correlations of figures 14, 15, and 16 can no doubt be improved.

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