CORRELATION OF HEAT, MASS, AND MOMENTUM TRANSFER COEFFICIENTS IN PLATE-FIN-TUBE HEAT EXCHANGERS

By

KEVIN FRANCIS O'BRIEN

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Oklahoma State University

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Thesis Approved:

Roger C. McLinston
Thesis Adviser

Dennis K McLaughlin

Gerald D Parker

Norman N Durham
Dean of the Graduate College

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NOMENCLATURE

English Letter Symbols

A	Area
A_t	Total tube area without fins
c_p	Specific heat at constant pressure
D	Tube outside diameter
D_h	Hydraulic diameter
D^*	Effective diameter
f	Fanning friction factor
FP	Friction correlation parameter
F_s	Dimensionless parameter based on fin spacing
G	Mass Velocity (ρV)
h	Surface convective heat transfer coefficient
i	Enthalpy
j	Colburn j factor
JP	Heat and mass transfer correlation parameter
K	Convective mass transfer coefficient
k	Thermal conductivity
M	Mean molecular weight
N_r	Number of tube rows
P	Pressure
Pr	Prandtl number ($c_p\mu/k$)
P_s	Fin pitch

$\dot{q}$	Heat transfer rate
Re	Reynolds number based on hydraulic diameter (GD_h/μ)
Re_B	Reynolds number based on longitudinal tube spacing (GX_b/μ)
Re_D	Reynolds number based on tube diameter (GD/μ)
Re_s	Reynolds number based on fin spacing (Gs/μ)
s	Center to center fin spacing $(1/P_s)$
Sc	Schmidt number (ν/D_c)
St	Stanton number (h/c_pG)
t	Temperature
u	Local velocity in flow direction
V	Average (bulk) velocity
W	Humidity Ratio
X_a	Transverse tube spacing
X_b	Longitudinal tube spacing
y	Fin thickness

Greek Letter Symbols

δ	Condensate layer thickness
μ	Absolute viscosity
ρ	Density
ν	Kinematic viscosity (μ/ρ)
σ	Ratio of free flow area to total area

Subscripts

d	Total transfer (i.e. based on enthalpy potential)
fg	Property associated with liquid vapor phase change
i	Total transfer (i.e. based on enthalpy potential)
ℓ	Latent (i.e. property associated with a change of phase)

m	Mass transfer (i.e. based on concentration potential)
s	Liquid-vapor interface
v	Vapor
w	Wall
∞	Free stream

CHAPTER I

INTRODUCTION

In man's early attempts to lower the temperature of his immediate environment there was little or no need to accurately predict the effects of his actions. Indeed for many centuries the work in producing artificially lowered temperatures was measured in terms of the number of slaves necessary to carry the snow from the mountains or fan the water on a cool desert night (4). Investigations into the nature of heat, however, were severely limited except for the last one hundred years. Recorded instances speak of Western Europeans and Americans criticizing the makers of artificial ice as late as the nineteenth century for intruding in such a God-provoking enterprise (4).

Interest in the industrial and environmental applications of refrigeration increased tremendously in the late nineteenth and early twentieth centuries with the need to transport meat. In the 1920's and 1930's the finned evaporator was developed and it is still in wide use today (5). Unfortunately the design procedures for many types of finned evaporators of today are still based on a "cut and try" process.

The purpose of this work is to correlate the heat and momentum transfer characteristics of a plate-fin-tube heat exchanger operating in a dry sensible heat transfer mode, to the heat, mass, and momentum transfer characteristics of the same exchanger operating in a wet or mass transfer mode. The obvious advantage of such a correlation would be that a single test of an exchanger in the dry mode would be sufficient to predict its operation in the vastly more complicated mass transfer, or dehumidification, mode.

Extensive heat, mass, and momentum transfer data has been reported by Burchfield (14) and this shall be used as a basis for the correlations. The distinguishing characteristic of this data is that it is the only data available which, for the same coils, gives not only the dry sensible heat transfer characteristics, but those for mass transfer as well. This data is discussed further in the next chapter.

It has been known for some time that, for a given temperature difference, a finned evaporator will operate more effectively, with respect to sensible heat transfer, if dehumidification is occuring simultaneously. The explanation involves the interaction of the boundary layer of the flow field with the moisture deposited on the fin surface. Therefore the primary experimental thrust of this report was the construction of an apparatus which would allow the visual observation of a fin surface during mass transfer. In this way a qualitative understanding of the effects of moisture deposition may be used to account for the heretofore unpredictable differences between dry and wet coil performance.

CHAPTER II

LITERATURE REVIEW

This paper is primarily concerned with correlation procedures for the prediction of wet coil performance on the basis of dry coil testing; however, literature pertaining to this area is highly dependent on the literature in the general area of condensation heat and mass transfer.

Within the literature it is noted that the first analogy with regards to a heat transfer coefficient was presented by Osborne Reynolds in 1874. In his proposal Reynolds assumed that the turbulent diffusivities, the mechanisms for heat and momentum transfer, are equal at any particular point in the flow. Armed with this assumption, Reynolds was able to show that there exists a relation between the heat transfer coefficient and the friction factor on a local basis. This relation can be expressed in nondimensional form as:

$$\frac{h}{Gc_p} = St = \frac{f}{2} \qquad (2.1)$$

where St = local Stanton number

G = mass velocity (lb_m/ft^2-hr)

c_p = specific heat at constant pressure (BTU/lb_m-°F)

h = local heat transfer coefficient (BTU/hr-ft^2-°F)

f = local fanning friction factor

Experiments have shown that for certain fluids, Prandtl number close to one, the Reynolds analogy is quite good. This is only reason-able since in his derivation Reynolds assumed turbulent diffusivities for heat and momentum to be equal, which is the same as specifying fluids with the turbulent Prandtl number equal to one.

The first analytical investigation into the effects of condensation can be traced to Nusselt in 1916 (7). His model was a vertical flat plate maintained at a temperature below the saturation temperature of the surrounding vapor as shown in Figure 1. Nusselt assumed that the weight of the condensate is balanced only by the shear stresses at the wall. By neglecting the effects of fluid acceleration and energy convection he was able to derive the velocity profile within the condensate layer

$$u = \frac{\rho}{2\mu}\,\delta^2\left(2\,\frac{y}{\delta} - \frac{y^2}{\delta^2}\right) \tag{2.2}$$

where u = local velocity in the x direction (ft/hr)

ρ = density of the condensate (lb_m/ft^3)

μ = absolute viscosity of the condensate (lb_m/ft-hr)

δ = condensate layer thickness (ft)

y = perpendicular distance from the plate (ft)

Nusselt further postulated that the resistance to heat transfer would be due solely to the liquid film, giving

$$h(x) = \frac{k}{\delta} = \left[\frac{k^3 \rho\, i_{fg}\, x^3}{4\nu(t_s - t_w)}\right]^{1/4} \tag{2.3}$$

where $h(x)$ = local heat transfer coefficient (BTU/hr-ft^2-°F)

k = thermal conductivity of the condensate (BTU/hr-ft-°F)

i_{fg} = latent heat of vaporization (BTU/lb_m)

x = distance from leading edge of the plate (ft)

ν = kinematic viscosity (ft^2/hr)

t_s = saturation temperature of the gas (°F)

t_w = wall temperature (°F)

Eckert and Drake demonstrated the validity of Nusselt's assertion through experiments (16). Deviation was found in the higher Reynolds numbers, based on the film thickness δ; however, this is due to the instabilities in the film at the larger film thicknesses.

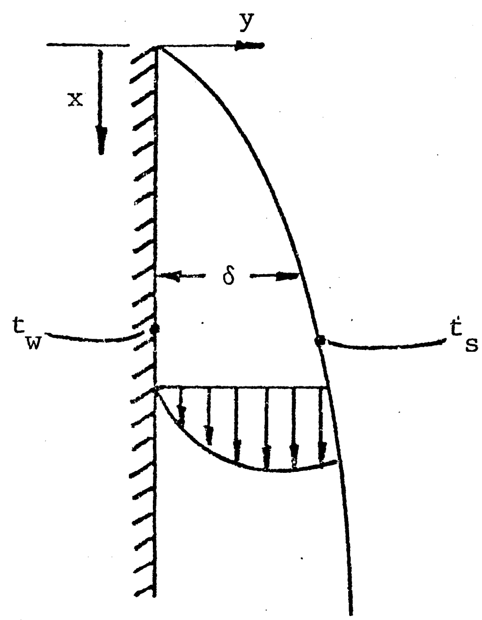

Figure 1. Nusselt's Model of Film Condensation

Refinements by Sparrow and Gregg (8) using the more precise boundary layer equations take into account the fluid acceleration and energy convection, or buoyancy, terms and result in the introduction of a Prandtl number dependence. As seen in Figure 2, the only significant difference from Nusselt's theory is at high heat rates or extremely low Prandtl numbers (liquid metals). These results demonstrate the wide range of fluids which are acceptable to Nusselt's theory.

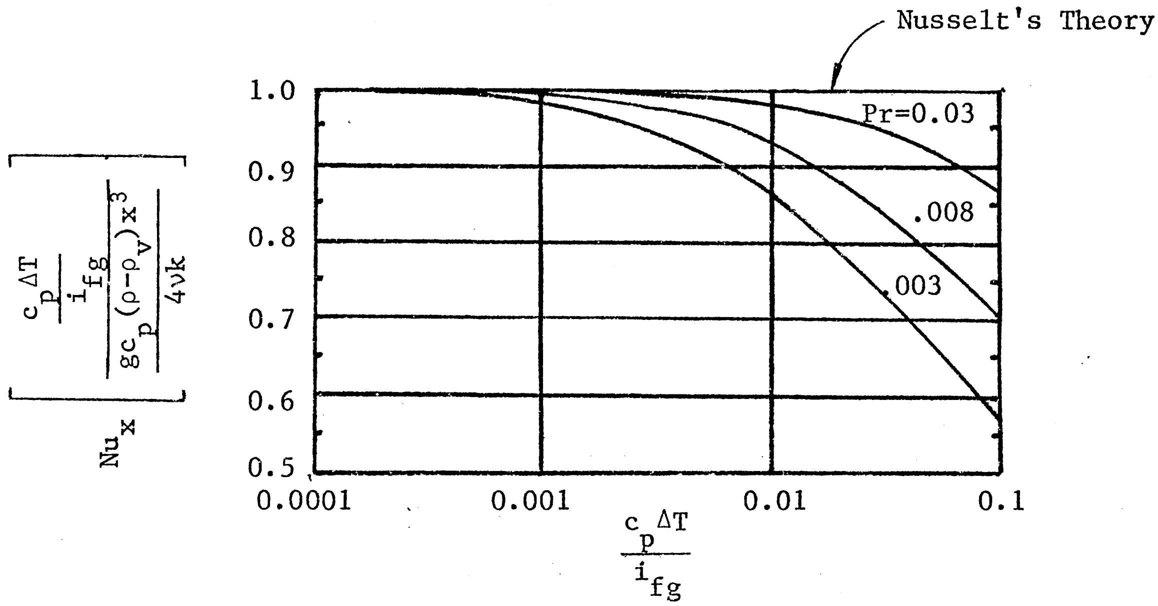

Figure 2. Prandtl Number Effect in the Heat Transfer Coefficient

Heat transfer with drop condensation was investigated by Graham and Griffith (28) in 1972. In their pure component analysis, steam was condensed on a copper surface which was polished to a mirror finish. Photographs of the surface, along with heat transfer measurements were made during the testing. Their results show that at atmospheric pressure, drops of diameter less than 40μ covering 23 percent of the surface area, transfer 90 percent of the heat. This is due to the increased resistance to conduction through the drops as the drops become larger.

All analysis to this stage has been concerned with pure component condensation. Another practical application of interest is that of a mixture of condensable vapors and noncondensable gases.

As a beginning in this area, Sparrow and Lin (12) analyzed a similar problem to that studied by Sparrow and Gregg with the difference of adding noncondensables to the vapor. The result was a reduction in heat transfer up to 50 percent when compared to the pure vapor case. This is a result of a diffusion process being controlled by a large buildup of the noncondensable at the liquid-vapor interface. A direct consequence is that the partial pressure of the vapor at the interface is reduced. This, in turn, lowers the temperature at the interface and thereby lowers the effective temperature difference. This is the cause of the observed reduction in heat transfer.

It is apparent that a model of the process of an actual heat exchanger, which should include such parameters as the bulk flow of the free stream, wall temperature variation, forms of condensate other than film type, and many more, would be a major, if not impossible task.

With so many factors to be considered in modeling, Colburn looked into the possibility of a correlation of experimental data in 1933 (9). Using his own data along with that of numerous other investigators, he obtained a correlating factor with Reynolds number for convective heat transfer in sensible operation. This j factor is

$$j = \frac{h}{Gc_p}\left(\frac{c_p\mu}{k}\right)^{2/3} \tag{2.4}$$

where k = thermal conductivity (BTU/hr-ft-°F)

μ = absolute viscosity (lb_m/ft-hr)

He also proved Reynolds assertion that friction and heat transfer may be related. In terms of his j factor

$$j = \tfrac{1}{2}f \tag{2.5}$$

where f is the Fanning friction factor. It should be noted, however, that these results apply only to turbulent flow.

In an extension of this work to mass transfer, Colburn and Chilton (10) in 1934, proposed a new correlation factor, with Reynolds number, of the form

$$j_m = \left(\frac{KP_{fg}M}{G}\right)Sc^{2/3} \tag{2.6}$$

where K = mass transfer coefficient (lb moles/hr-ft^2-atm)

P_{fg} = partial pressure potential (atm)

Sc = Schmidt number

M = mean molecular weight

The form of this correlation which is now in more general use is

$$j_m = \left(\frac{h_m}{G}\right)Sc^{2/3} \tag{2.7}$$

where h_m is the mass transfer coefficient with units of lb_{ma}/ft^2-hr defined by the equation

$$\dot{q}_\ell = h_m A(W_\infty - W_w) i_{fg} \tag{2.8}$$

in which $\dot{q}_\ell$ = latent heat transfer rate (BTU/hr)

A = total surface area (ft^2)

W_∞ = humidity ratio in the free stream (lb_{mw}/lb_{ma})

W_w = humidity ratio at the wall (lb_{mw}/lb_{ma})

i_{fg} = enthalpy of vaporization (BTU/lb_m)

and G = mass velocity of the air (lb_{ma}/ft^2-hr).

Chilton and Colburn presented data from only one source for flow across tube banks and noted that there was a 25 percent error. Despite such limited data and considerable error, Colburn asserted that

$$\tfrac{1}{2}f = j = j_m = j_i \tag{2.9}$$

where j_i is the total heat transfer j factor which is defined by the equation

$$\dot{q} = h_d A(i_\infty - i_w) \tag{2.10}$$

where $\dot{q}$ = total (sensible and latent) heat transfer rate (BTU/hr)

i_∞ = enthalpy of the freestream (BTU/lb_{ma})

i_w = enthalpy at the wall (BTU/lb_{ma})

h_d = total heat transfer coefficient (lb_{ma}/ft^2-hr)

This assertion has come to be known as Colburn's j factor analogy for momentum, heat, and mass transfer. Recent research (13,27) has shown that this analogy is quite poor for many cases.

In 1970, Bettanini (11) presented the results of his investigation into the effects of moisture deposition on heat transfer coefficients. When dehumidification tests were conducted on a single vertical flat plate, the total j factor, j_i, was 20 to 30 percent higher than the sensible j factor calculated for the dry tests. In addition, when gypsum nodules were sprayed onto the surface to simulate water deposition, an increase in the sensible j factor for the dry tests was noted. This demonstrated the significant effect of the condensate on heat transfer.

In 1973, Guillory and McQuiston (13) presented their work in the area of parallel plate exchangers. By removing many major sources for error and combining analysis with experimental data, plots for total, sensible, and mass transfer j factors are presented. Their conclusions show that the use of dry data in the design of wet exchangers can result in fin area overestimations and pressure drop underestimations on the order of 30 percent.

Additional work on parallel plate exchangers was published in 1974 by Helmer (25). In his work Helmer took velocity and temperature measurements between the plates of the exchanger during dehumidification operation. Helmer's results were always lower than the data of Guillory and McQuiston. Helmer also presented correlations for local heat and mass transfer data in turbulent flow.

McQuiston (26) reports on the continuation of his work with parallel plate exchangers in 1976. In his work McQuiston showed that the free stream turbulence is not a dominant factor in heat and mass transfer. Also tests conducted proved that the moisture content or driving potential has no effect on the j factors.

Extensive data has been obtained by Burchfield (14) for three plate-fin-tube heat exchangers. In this work Burchfield obtained total, sensible, and mass transfer j factors along with friction factors for three industrial grade heat exchangers. This report will be discussed further in the next chapter.

The end result of this investigation should be to provide useful correlations between wet and dry exchangers. Work in the literature dealing with correlations of this type were found in references (6,16-24, 1,2,29,33). Of major significance is that work presented by Jameson (6) in 1945. This work identified the important parameters in correlating pressure drop across a bank of helically finned tubes. This work can be modified for the plate-fin-tube heat exchangers.

Further investigations involving the identification of the significant coil parameters are presented by Gram, et. al. (29), and Briggs and Young (19). In both these papers forced convection heat transfer and momentum transfer (pressure drop) were considered. Recent work by Elmahdy (32) also deals with heat transfer correlations. These works will be discussed further in Chapter V.

Two papers by Rich (1,2) also contribute to an understanding of the effect of physical parameters. The first paper (1973) deals with the effect of fin spacing while the second paper (1975) is concerned with the number of rows on heat transfer performance. Both papers by Rich and the one by Jameson will be considered further in the chapter on correlations.

CHAPTER III

EXPERIMENTAL PROCEDURE

It is evident from previous investigations (13,27,14) that the performance of a heat exchanger under dehumidification operation will have different heat transfer and pressure drop characteristics than the same exchanger operated with only sensible heat transfer. It is therefore necessary, as a minimum, to qualitatively understand the influence of condensate deposition on the water vapor-air mixture flow through the exchanger. For this reason, the apparatus shown in Figure 3 was constructed.

The operation is as follows: compressed air from the laboratory system is regulated through a long tube in which the flow rate is measured. A spray nozzle humidifies the air to near saturation and psychometric readings are taken to monitor the process. The air is then channelled through the visual section where photographs may be taken. Figure 4 shows a side view of the visual section. Water from an ice bath is circulated on the backside of the plate in a direction counterflow to the airstream. The fin stock material is mounted to the plate with thermocouples between the two in order to monitor the plate temperature. The fin stock must at all times be below the dew point of the humidified air. The air passes between the fin stock and a plexiglas plate through which observations can be made. The depth of the air flow channel may be varied to simulate different fin spacings.

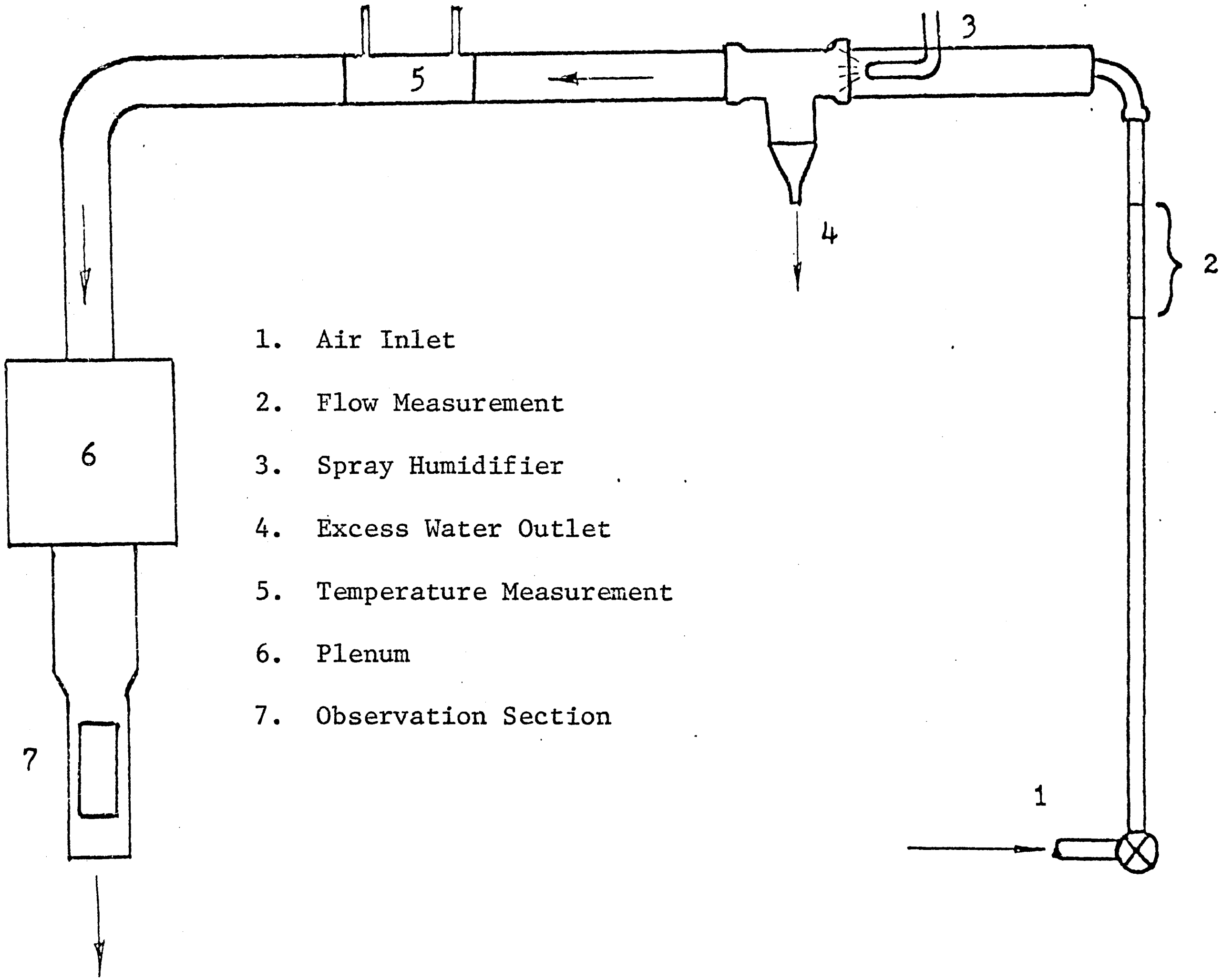

Figure 3. Schematic of Visual Test Facility

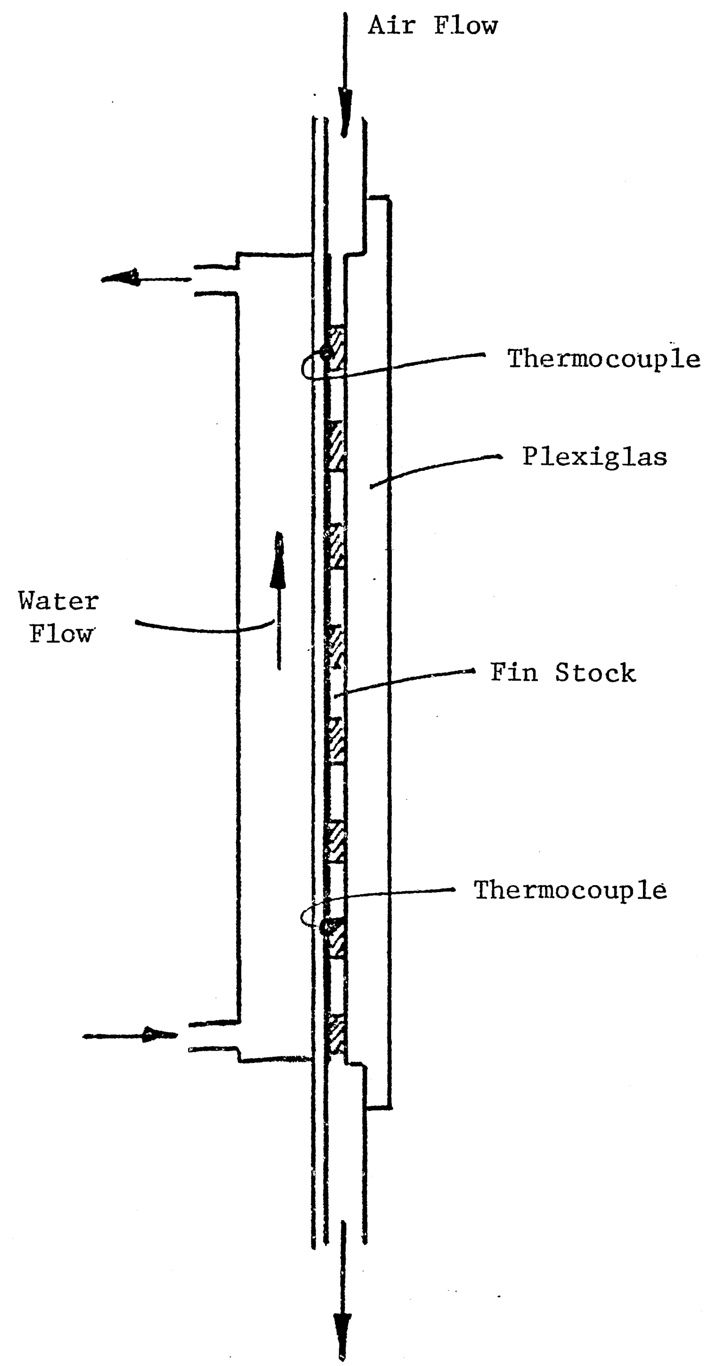

Figure 4. Schematic of the Observation Section

A typical test was run in the following manner. First a flow rate was selected to give the desired Reynolds number across the plate. Once this was adjusted the spray humidifier was turned on and the water drain was checked for excess water. Psychometric readings were then obtained from the wet and dry bulb temperature measurements to assure that the air was near saturation. Once all this was done and the operation showed no fluctuations in flow rate and temperatures, the pump for the ice water was turned on. The fin stock specimen was then observed until such time as the overall behavior of the moisture on the specimen showed no change.

A second apparatus, shown in Figure 5, was used to determine the heat, mass, and momentum transfer coefficients of different heat exchangers. A detailed description of the facility may be found in Reference 14 while only the basic operation will be described herein.

A multiple stage blower exhausts air through a butterfly valve used for flow rate control. The air is then heated when passed through a heat exchanger in order that the humidification section next in line can work more effectively. The air then passes through two 90 degree bends and over internal mixers to assure that no water droplets are carried to the temperature measurement section. From this point the air is channelled as high as the ceiling permits and is then passed over the test heat exchanger in a downflow manner. A catch drain is provided for the condensate run-off before passing through the outlet temperature measurement section. Finally, before returning to the blower, the flow rate of the air is measured by means of a pitot tube.

Raw data, in the form of temperature and pressure readings are then processed in a data reduction computer program. Output from this program

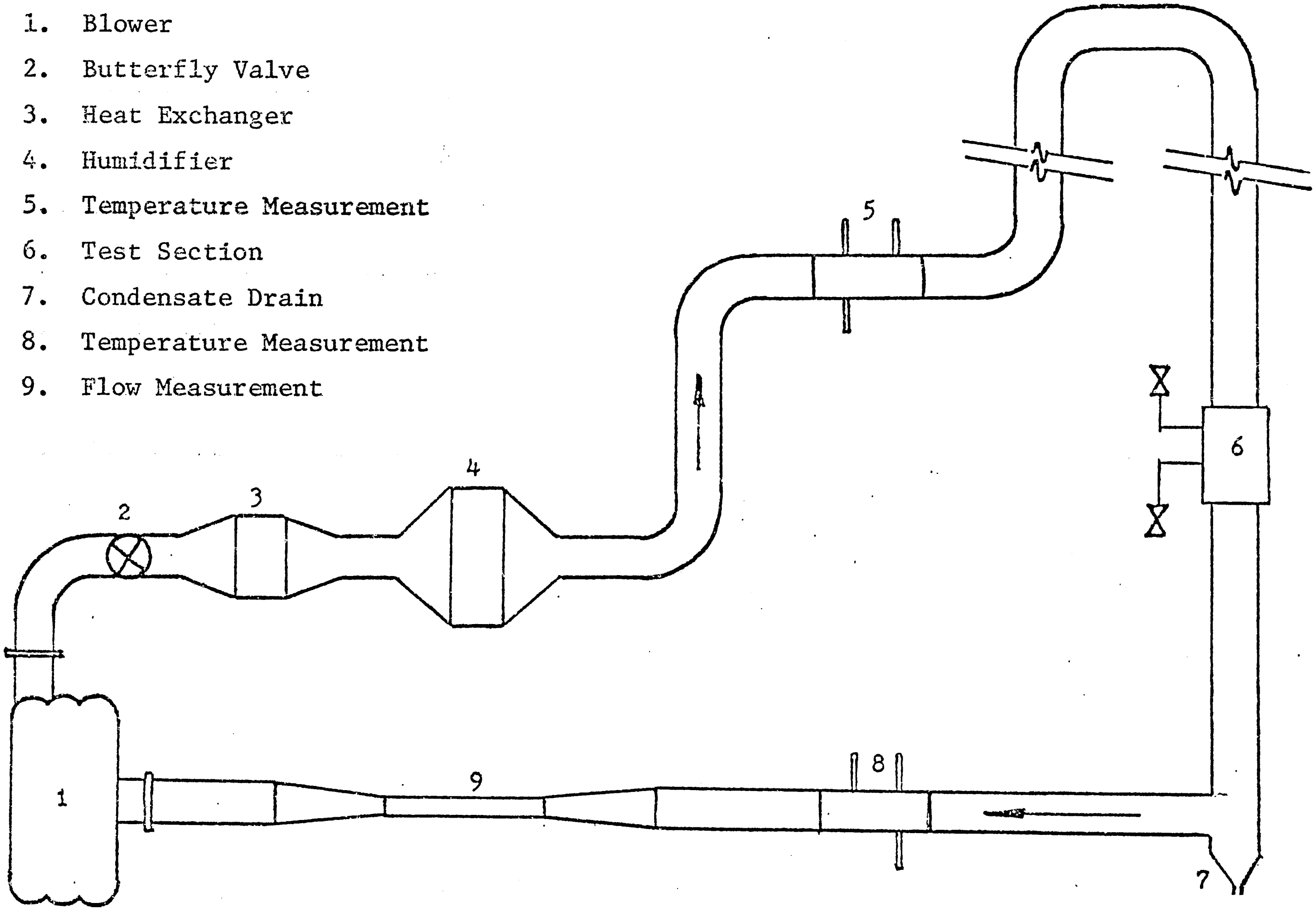

Figure 5. Schematic of the Closed Heat Transfer Loop

includes sensible j factors, total j factors, mass transfer j factors, Fanning friction factors, intermediate temperatures, Reynolds number, and numerous other pertinent information. A more detailed description of this program may also be found in Reference 14.

CHAPTER IV

EXPERIMENTAL RESULTS

Data obtained from the closed heat transfer loop for four, eight, and twelve fins per inch heat exchangers are presented in the report by Burchfield (14). In his report Burchfield states:

> At the lower Reynolds numbers, the j factors seem to be in fairly good agreement, but at the higher numbers the difference between the dry and wet j factors becomes very pronounced.... For a wet surface, the sensible and total j factors closely agree, with the j factors for dropwise condensation being slightly higher than the j factors for filmwise condensation (p. 36).

Noted by Burchfield was the deviation of the twelve fin per inch data from the above observations. Subsequent data obtained by Burchfield and the author are presented herein.

The behavior of the twelve fins per inch data, namely the sensible j factor for filmwise condensation being lower than the sensible j factor with no mass transfer, was felt to be a result of the interaction between the condensate and the air stream due to the reduction in area between the fins. Tests were made on two additional heat exchangers, one with ten fins per inch and the other with fourteen fins per inch, in order that this effect could be studied more closely. Results from these tests are presented in tabular form in the Appendix. In Figures 6 through 11 the j factor data is plotted versus the Reynolds number based on hydraulic diameter. Trends observed from these tests will be discussed in Chapter V.

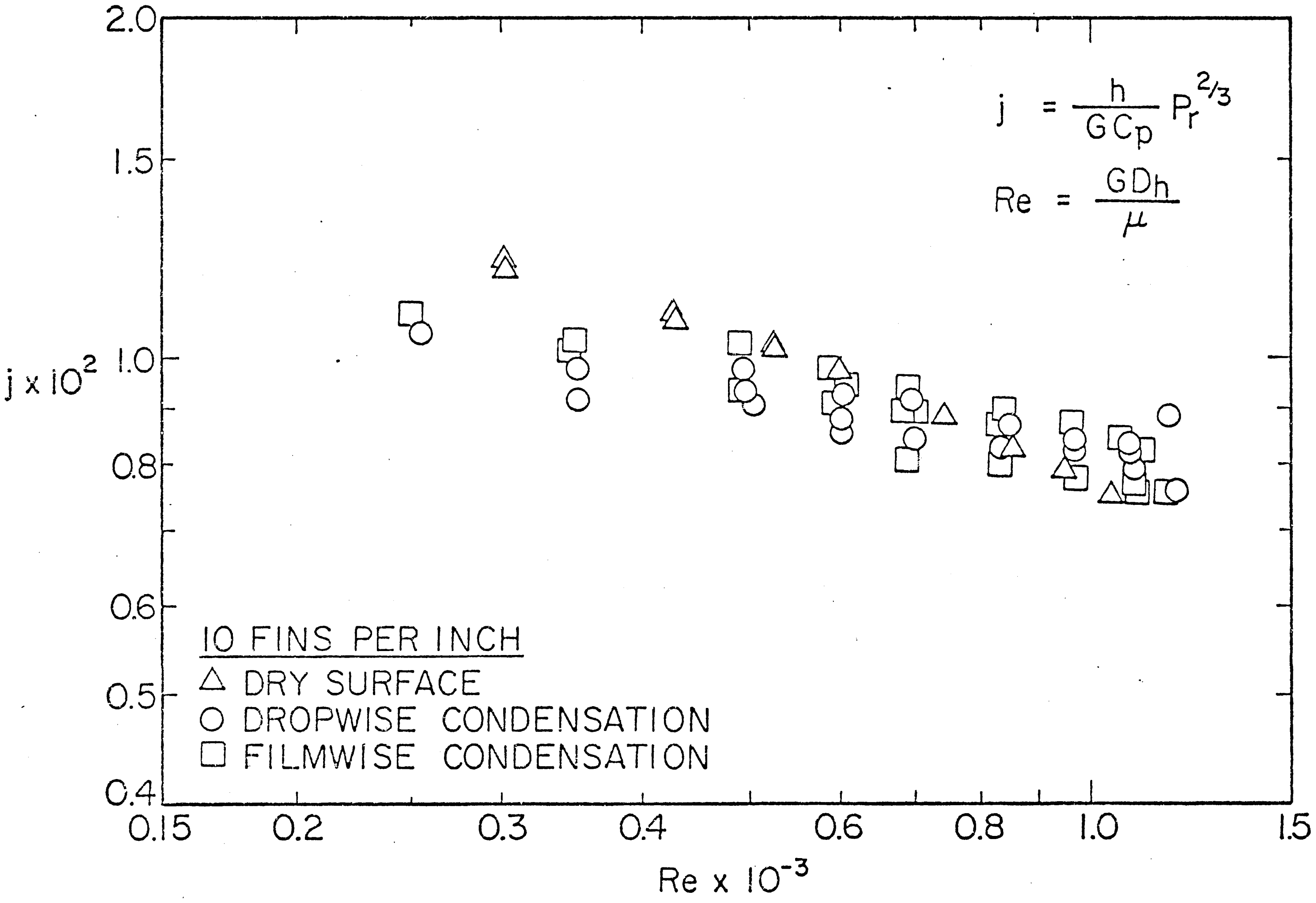

Figure 6. Sensible j Factors for the Plate-Fin-Tube Heat Exchanger With 10 Fins per Inch

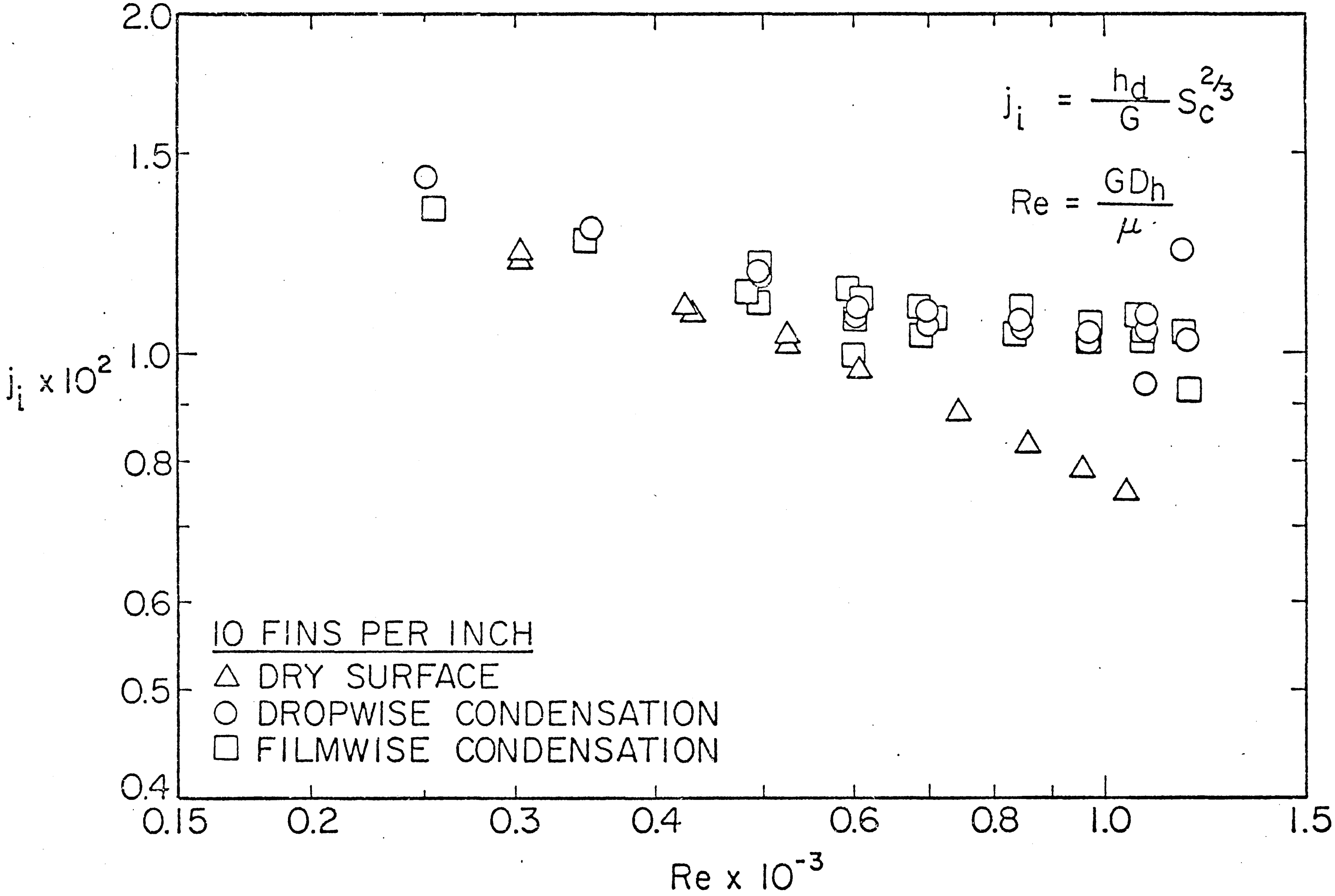

Figure 7. Total j Factors for the Plate-Fin-Tube Heat Exchanger With 10 Fins per Inch

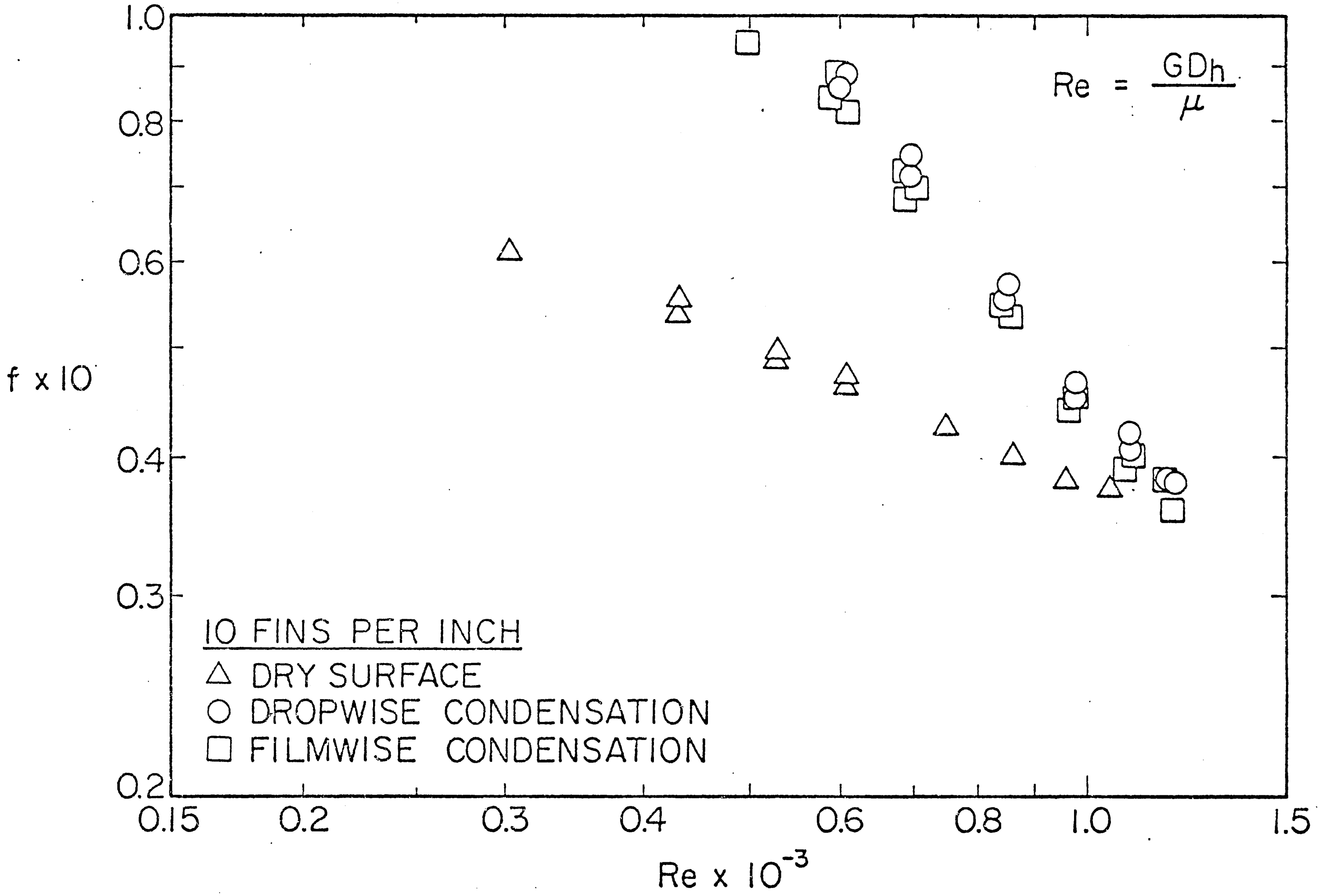

Figure 8. Friction Factors for the Plate-Fin-Tube Heat Exchanger With 10 Fins per Inch

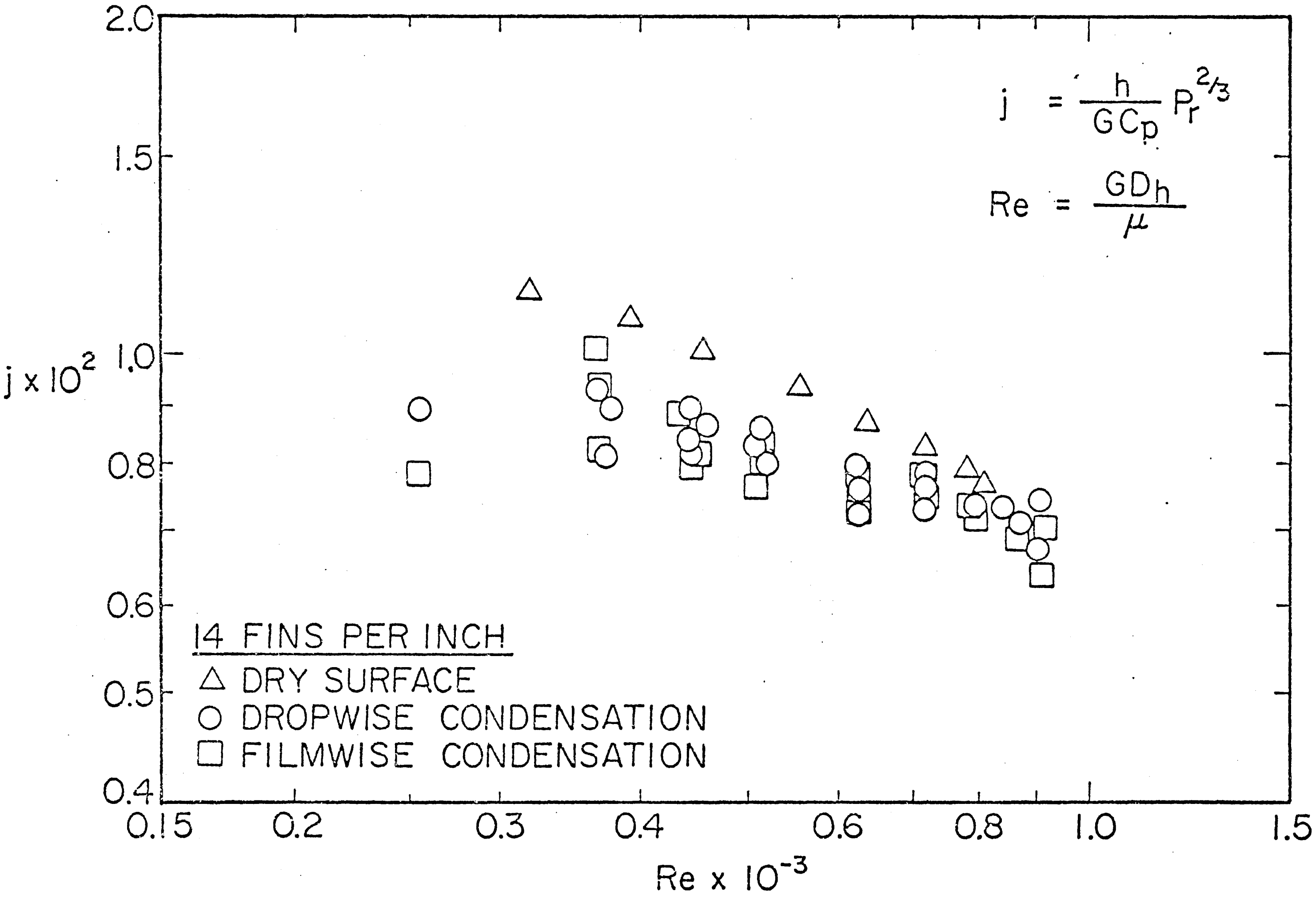

Figure 9. Sensible j Factors for the Plate-Fin-Tube Heat Exchanger With 14 Fins per Inch

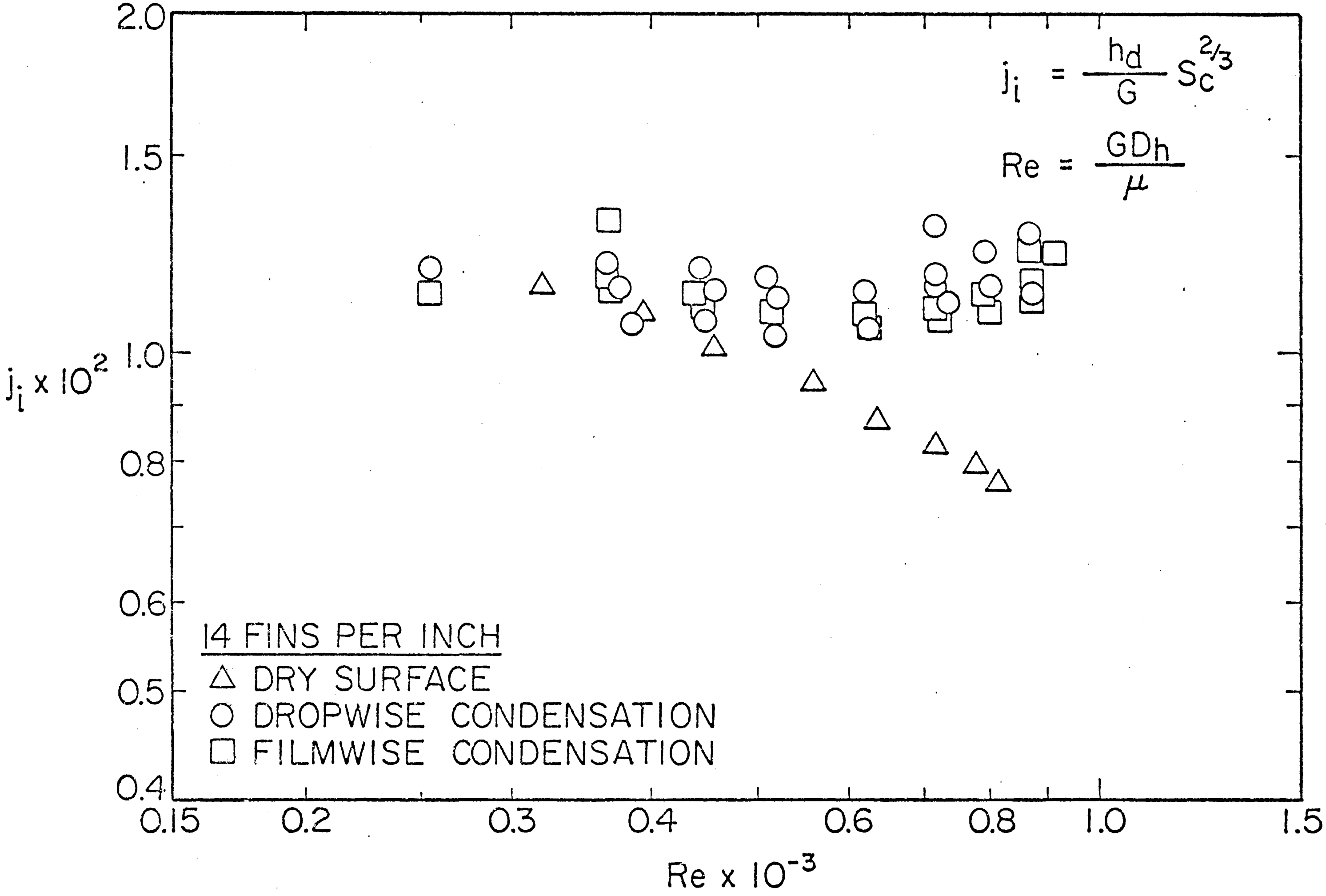

Figure 10. Total j Factors for the Plate-Fin-Tube Heat Exchanger With 14 Fins per Inch

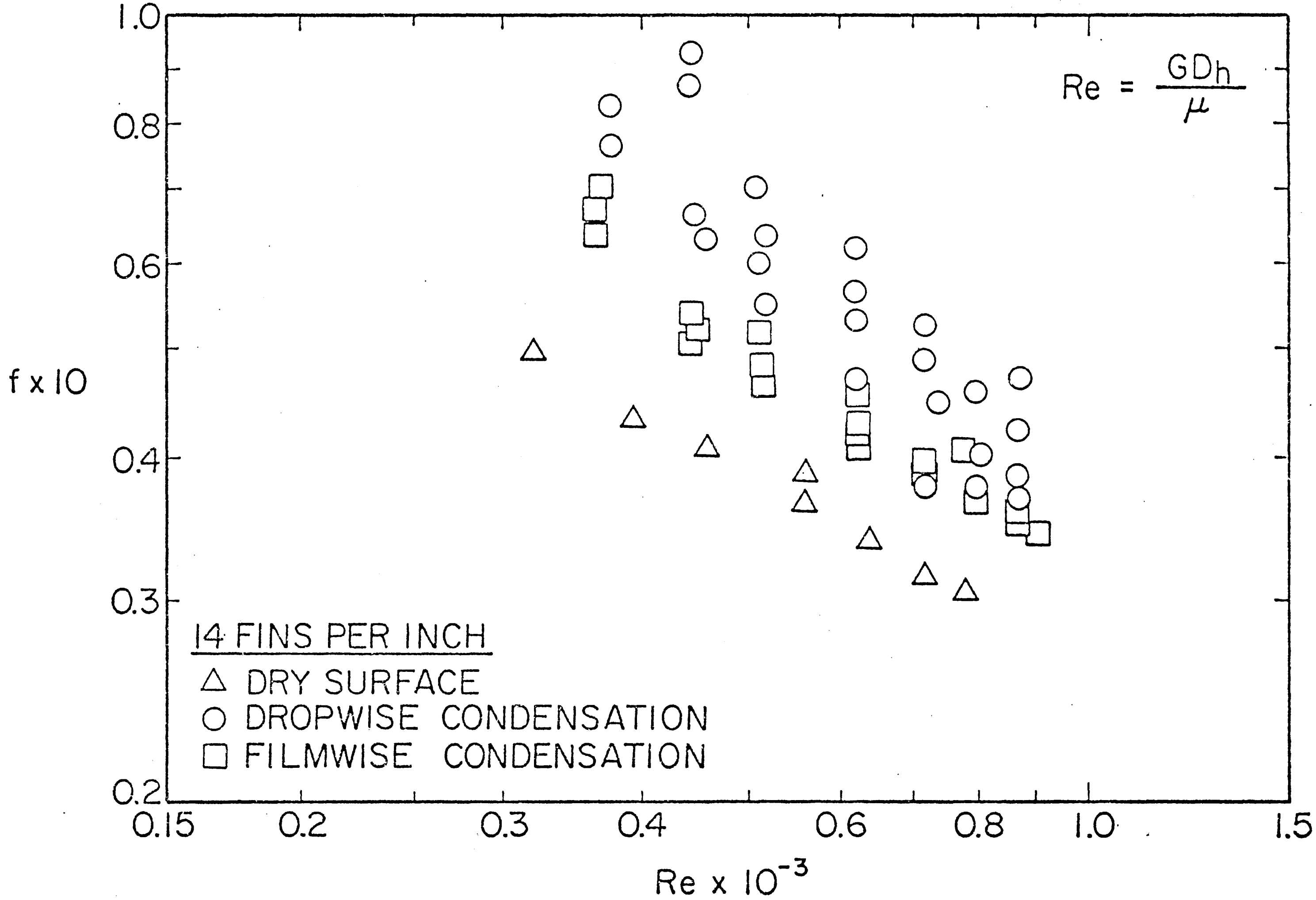

Figure 11. Friction Factors for the Plate-Fin-Tube Heat Exchanger With 14 Fins per Inch

A major portion of this study was devoted to the observation of condensation on fin material in a simulated air passage. Tests were run on the apparatus described in Chapter III for channel depths of 0.05 inches, 0.10 inches, and 0.13 inches corresponding to fin pitches of approximately 17, 9, and 7 fins per inch respectively. The Reynolds number, based on hydraulic diameter, was varied at each of the channel depths and numerous photographs were taken to reinforce the visual observations.

No major differences were observed in going from one channel depth to another and only the drop growth rate appeared to be affected by the change in flow rate.

Drop formation was the greatest in the region between the collars while the area immediately below the collars showed little or no drop growth. This is seen in Figures 12 and 13. At the lower Reynolds numbers, for all channel depths, the drop growth was much slower than at the higher Reynolds numbers. It appeared that at the higher flow rates the overall drop size was smaller at the time of the first released drop than at the lower flow rates. It is thought that this is a result of the drops being 'blown out' at the higher flow rates.

The first drops released were from the region of rapid drop growth between the collars. The effect of a released drop was to clear a path along the fin with only an occasional small bead left behind, Figures 14 and 15. Eventually most drops released from the underside of the collars where the condensate would accumulate, Figure 16. Within the paths cleared by the drops, new drops would begin to form again. This can be seen in Figures 14 and 16.

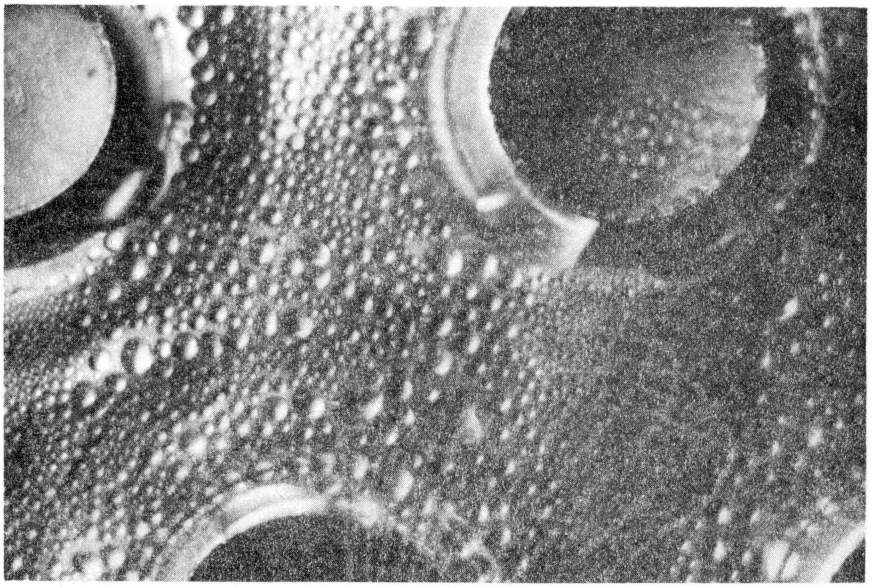

Figure 12. Visual Observation at Re=740
With 0.05 Inch Spacing

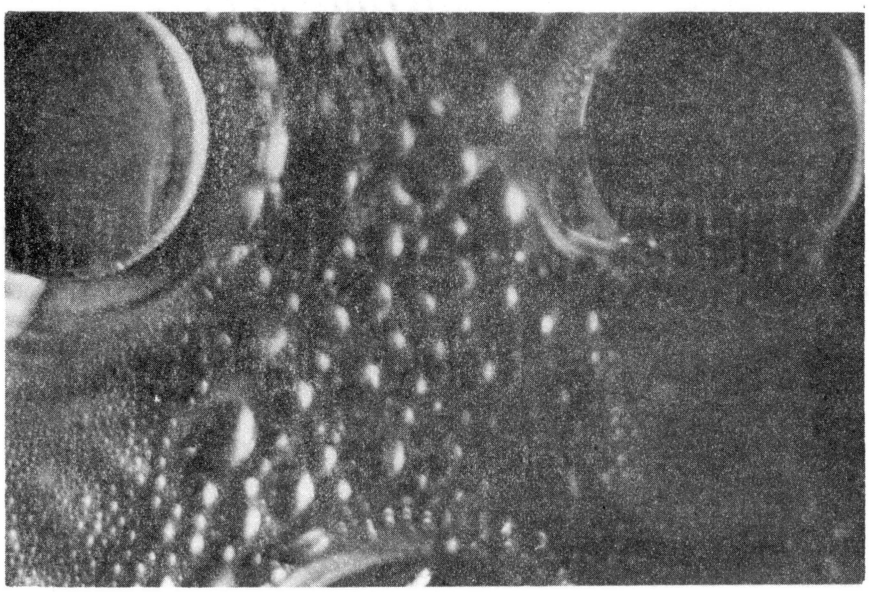

Figure 13. Visual Observation at Re=1100
With 0.10 Inch Spacing

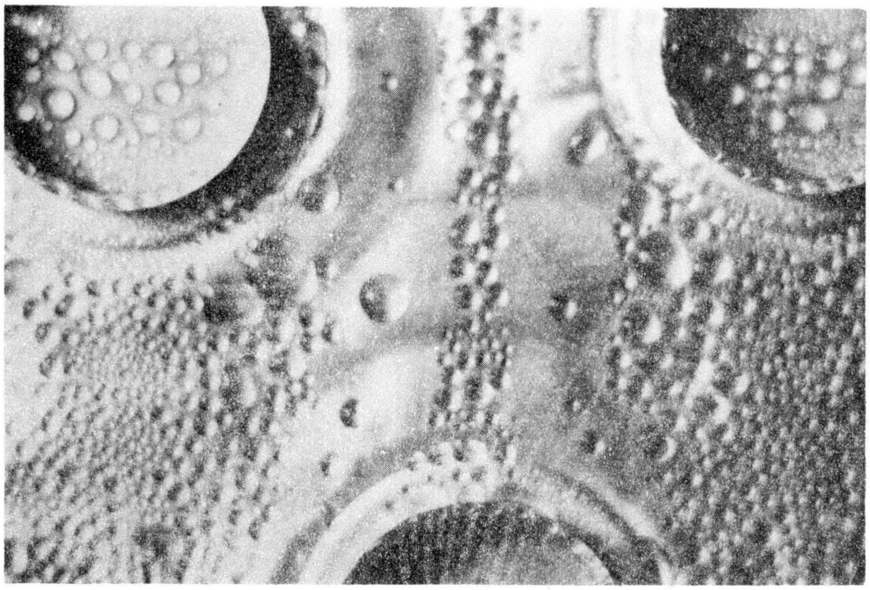

Figure 14. Visual Observation at Re=1488
With 0.05 Inch Spacing

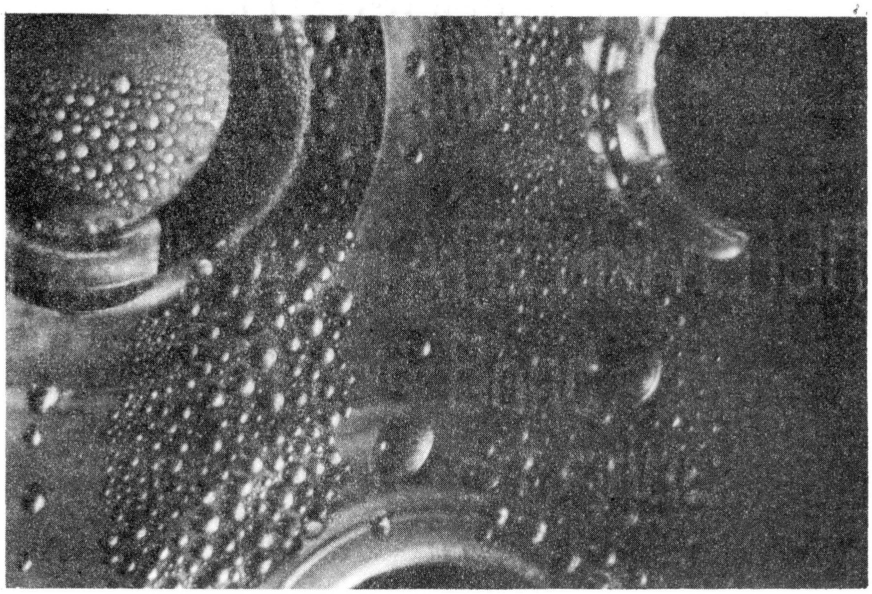

Figure 15. Visual Observation at Re=1530
With 0.10 Inch Spacing

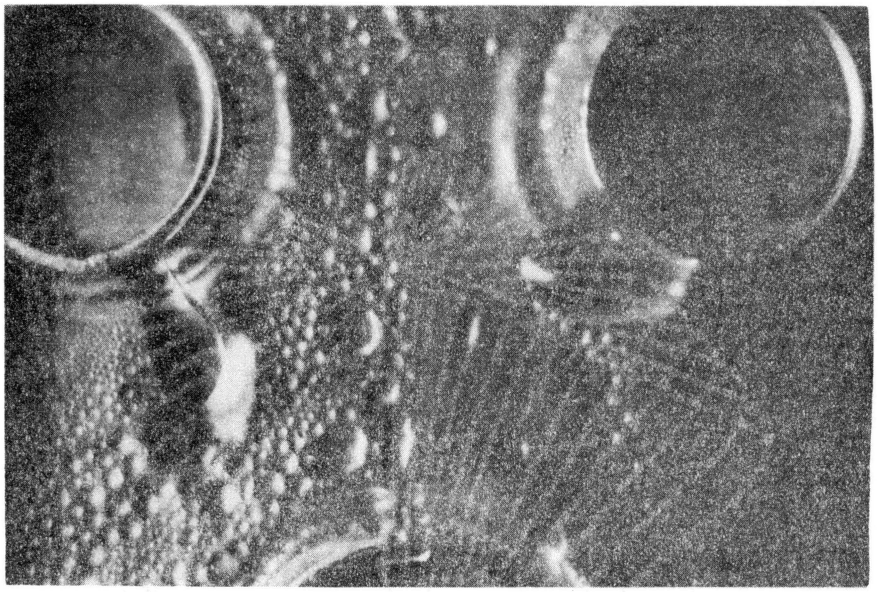

Figure 16. Visual Observation at Re=400
With 0.05 Inch Spacing

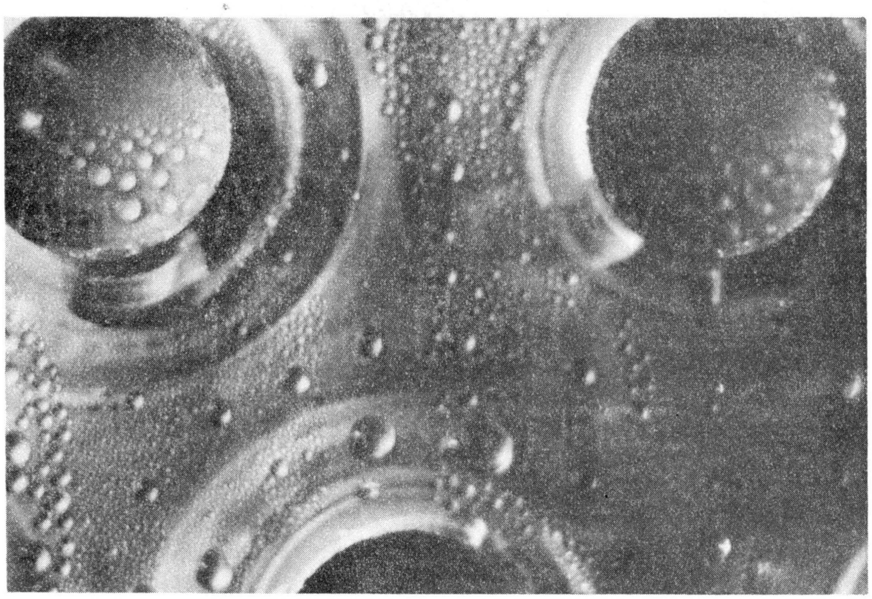

Figure 17. Visual Observation at Re=1100
With 0.10 Inch Spacing

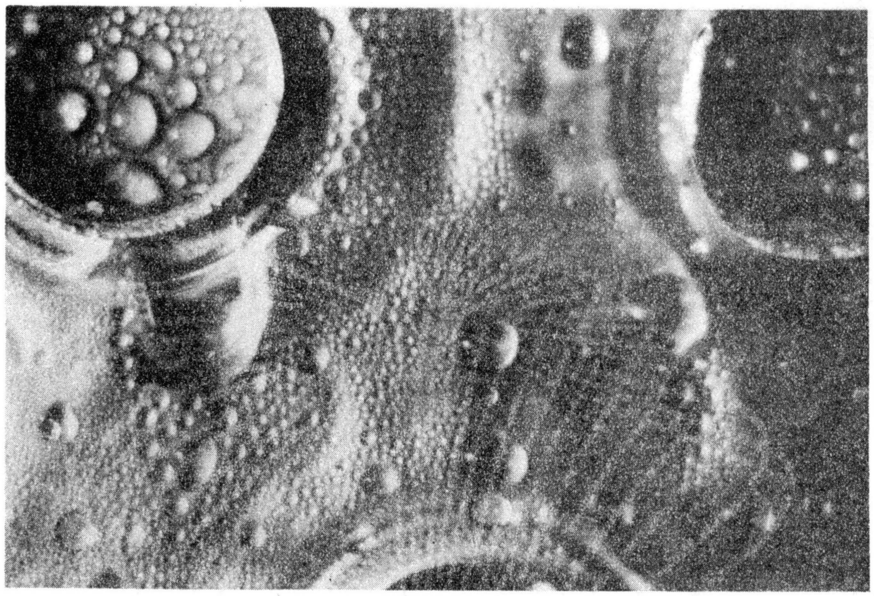

Figure 18. Visual Observation at Re=1488
With 0.05 Inch Spacing

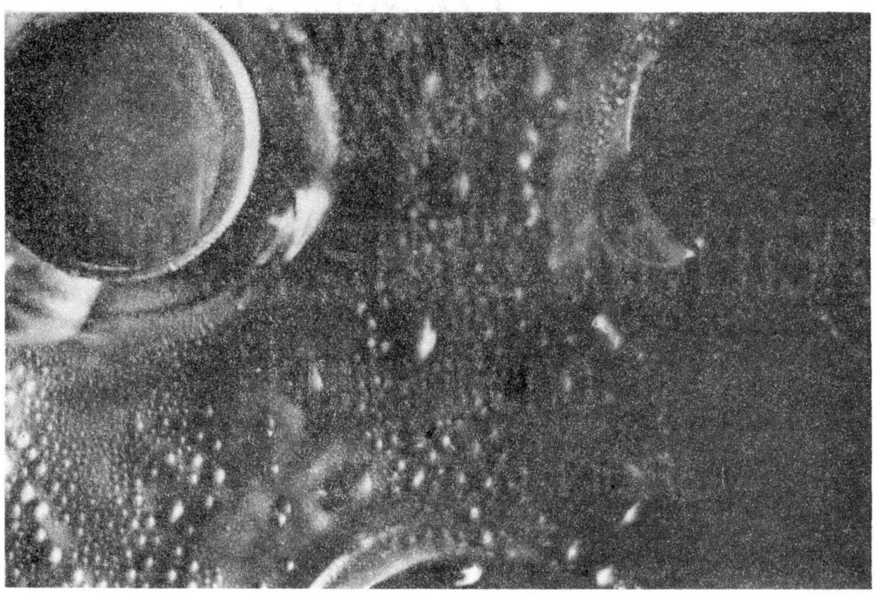

Figure 19. Visual Observation at Re=400
With 0.05 Inch Spacing

Each test was continued until the overall behavior of the condensate showed no change. In each case the general appearance of the fin was quite similar. As seen in Figures 17, 18, and 19, the fin was covered with drops of different sizes due to the passage of released drops.

Previous to these tests it was thought that blockage by the water in the region between the collars, or a radically different behavior of the condensate occurred at the smaller fin spacings. The previous results disprove these theories. The flow channel used in the tests differed from the actual process in only two respects. First the surface of the fin was maintained at a much more even temperature than the fins in an actual heat exchanger, which would tend to enhance condensation. Secondly the experimental apparatus had condensation forming on only one side of the air passage. It is thought that this could make a difference with fin spacings corresponding to fin pitches of 20 fins per inch and up. The smallest fin spacing tested corresponds to a fin pitch of 17 fins per inch. At smaller fin spacings it may be possible for condensate below the collars to bridge across the gap in sufficient quantity to alter the flow field. Such an occurrence would change the local heat transfer coefficient and thereby the overall heat transfer characteristics of the exchanger.

From Figures 6 through 11 it is noted that the effect of film condensation is nearly the same as drop condensation. Film condensation, it is thought, can be more accurately described as a wavy film due to the shear forces of the air. This rippling of the film causes an increase in turbulence of the air the same as drop condensation. Since the increase in turbulence is the reason for higher heat and momentum transfer characteristics, the j and friction factors should be affected similarly for drop and film condensation.

CHAPTER V

CORRELATIONS

This chapter is broken down into four sections. The first deals with those parameters found in the literature which appear to dominate the various correlations. The second part demonstrates an ability to correlate dry surface data when the appropriate parameters are chosen. The next section examines the effect of mass transfer on the j and friction factors and the determination of the significant parameters. In the final section, correlations are presented which allow the prediction of the j and friction factors on the basis of dry surface heat transfer and friction data.

Dry Surface Parameters

Two different methods were found in the literature for the formulation of the parameters. The work of Briggs and Young (19) identifies the basic dimensions for a finned tube and uses an extensive regression analysis on all possible combinations to determine the significance of each group. Gram, et. al. (29) worked with air flowing over in-line tube banks. In their analysis, extensive plots were used to determine the effect of different dimensionless groupings on the j and friction factors.

A simplified plate-fin-tube heat exchanger is shown in Figure 20. From the literature, a Reynold's number based on tube diameter was prevalent in nearly all of the correlations.

$$Re_D = \frac{GD}{\mu} \tag{5.1}$$

where G = mass velocity (lb_m/ft^2-hr)

D = tube outside diameter (ft)

μ = absolute viscosity (lb_m/ft-hr)

Therefore a correlation factor for the dry surface data of the form

$$JP = Re_D{}^n C^m \tag{5.2}$$

was sought where C is some combination of coil parameters representative of its particular geometry, and n and m are constants.

The geometric parameter of the form

$$C = \frac{A}{A_t}$$

where A = total air-side surface area (ft^2)

A_t = total tube area without fins (ft^2)

seemed to be the most meaningful. The ratio may be represented in terms of the coil parameters in the following manner.

$$\frac{A}{A_t} = \frac{4}{\pi} \frac{X_a}{D_h} \frac{X_b}{D} \sigma \tag{5.3}$$

where X_a = transverse tube spacing (ft)

X_b = longitudinal tube spacing (ft)

D = tube outside diameter (ft)

D_h = hydraulic diameter (ft)

σ = ratio of the minimum free flow area to the frontal area

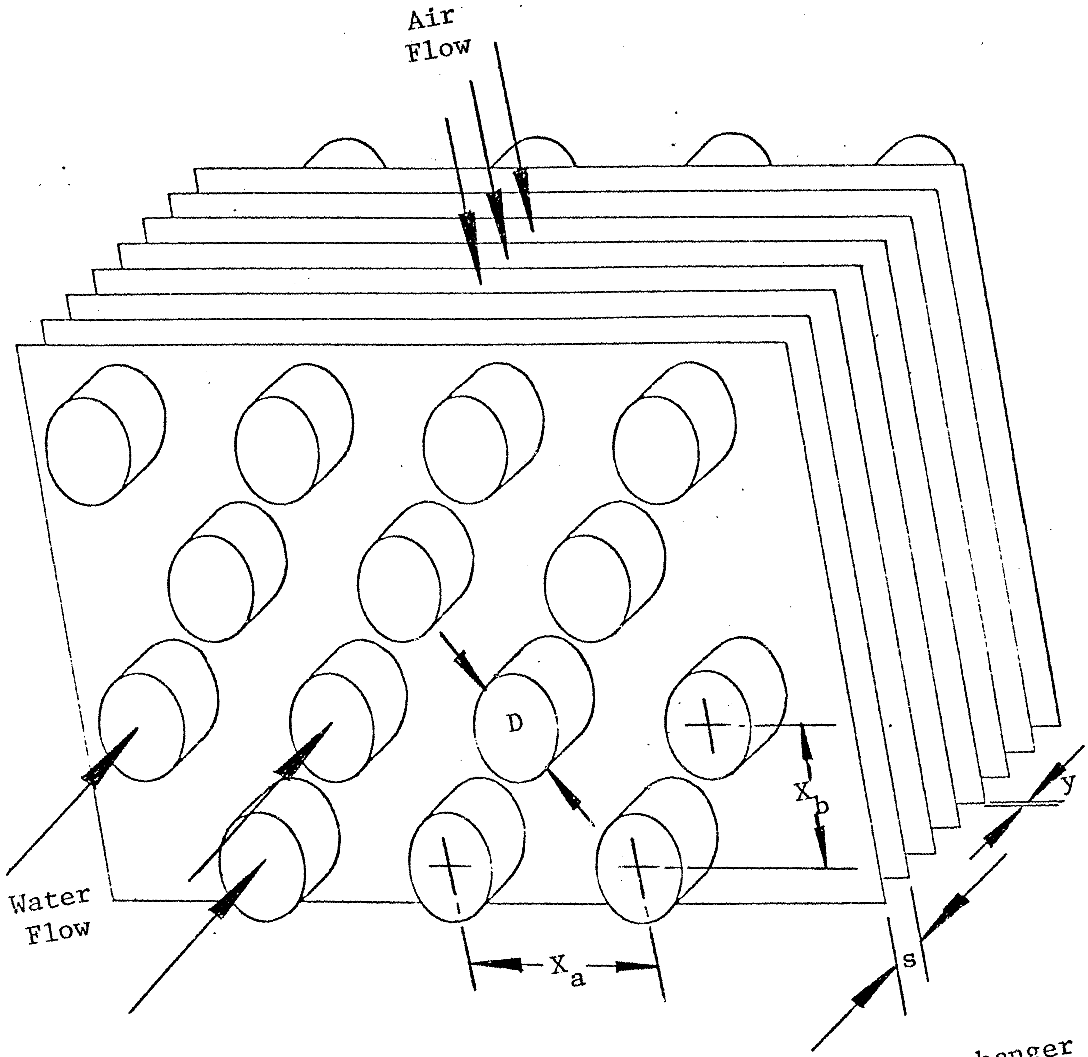

Figure 20. Simplified Plate-Fin-Tube Heat Exchanger

Correlation of the momentum transfer, or friction factors, is considerably more involved than the correlation of heat transfer data. The work of Jameson (6) has been modified from his work with large circular finned tube banks to plate-fin-tube heat exchangers with success.

Jameson defined an equivalent diameter for his finned tubes which, in plate-fin-tube terminology, is

$$D^* = \frac{\left(\frac{A}{A_t}\right) D}{(X_a - D)P_s + 1} \tag{5.4}$$

where D = tube outside diameter (ft)

P_s = fin pitch (fins/ft)

Jameson's correlation contains three other parameters as well. The first is the Reynolds number based on tube diameter as defined in equation (5.1). The second is

$$F_2 = \frac{(X_a - D)P_s}{4(1-P_s)y} \tag{5.5}$$

where y = fin thickness (ft)

The final parameter is

$$F_3 = \frac{X_a}{D^*} - 1 \tag{5.6}$$

where D^* is the effective diameter defined in equation (5.4).

Correlation of Dry Surface Data

Once the significant parameters have been identified the final step of the correlation process is to determine the relative importance of the individual terms. For the heat transfer correlation this requires the determination of the constants m and n. From the literature

(9,29) the exponent of the Reynolds number has been determined for finned tubes and tube banks to be about -0.4. With this substitution and a simple regression analysis the following correlation factor was obtained

$$JP = Re_D^{-0.4} \left(\frac{A}{A_t}\right)^{-0.15} \tag{5.7}$$

where $m = -0.15$.

Data from the study by Burchfield, the ten and fourteen fins per inch data presented in Chapter IV, and three other sources (2,3,5) are plotted in Figure 21. The coils included cover the range of 3/8 to 5/8 inch diameter tubes, fin pitch from 3 to 20 fins per inch, and tube layouts varying from 1 inch triangular to 1 1/2 x 1 3/4 inch triangular. The majority of the data is within ±10 percent of the mean with the data of Burchfield and this study behaving even better. The data of Figure 21 will be replaced by a line representing the mean. The equation of that line is

$$j = 0.0014 + 0.2618(JP) \tag{5.8}$$

This equation is based on data from heat exchangers with four rows of tubes. Rich (1) has studied the effect of the number of tube rows on the j factor and has found the Reynolds number based on longitudinal tube spacing to be of primary importance.

$$Re_b = \frac{GX_b}{\mu} \tag{5.9}$$

where X_b = longitudinal tube spacing (ft).

The results of reference 1 have been correlated as follows

$$\frac{j_n}{j_i} = 1 - 1280\, N_r\, Re_b^{-1.2} \tag{5.10}$$

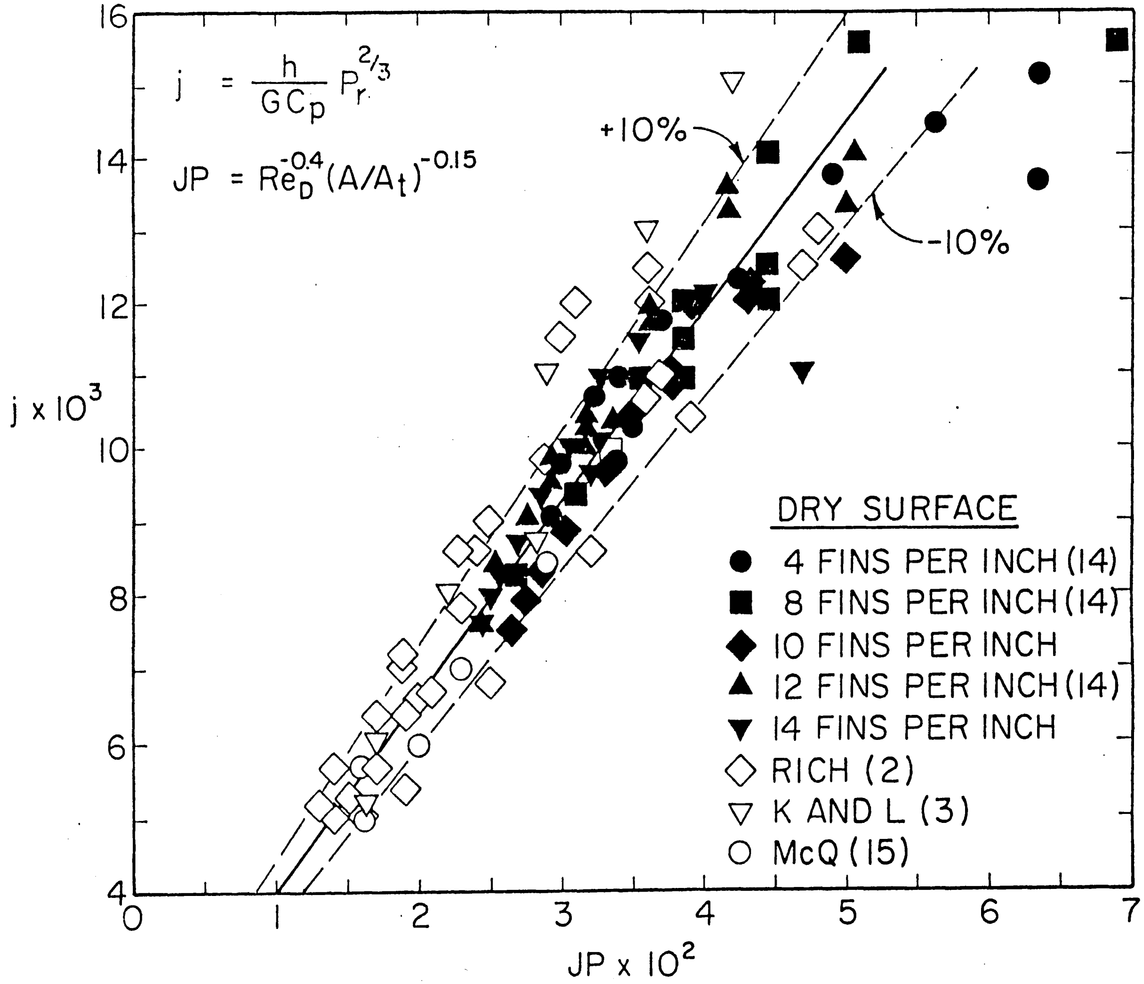

Figure 21. Correlation of Dry Surface Heat Transfer Data

where N_r = number of rows of tubes

j_n = j factor corresponding to N_r

j_i = j factor for a 1 row coil

Equation (5.10) is within ±7 percent of the data for Re_b between 3000 and 15000. This relation strictly applies to the geometry used by Rich; 1/2 inch diameter tubes on a 1 1/4 inch triangular layout with 14 fins per inch, however it is thought that the same effects are present in exchangers with tube sizes from 3/8 to 5/8 inch, triangular layouts of 1 to 1 1/2 inch, and fin pitch from 8 to 14 fins per inch. Mass transfer j factors will experience about the same effect as the sensible j factors.

Combining equations (5.9) and (5.10) gives

$$\frac{j_n}{j_4} = \frac{1 - 1280\, N_r\, Re_b^{-1.2}}{1 - 5120\, Re_b^{-1.2}} \tag{5.11}$$

and

$$j_n = \left\{\frac{1 - 1280\, N_r\, Re_b^{-1.2}}{1 - 5120\, Re_b^{-1.2}}\right\} \{0.0014 + 0.2618(JP)\} \tag{5.12}$$

The modified Jameson correlation for friction data is

$$FP = Re_D^{-0.25}\left(\frac{D}{D*}\right)^{0.25} \left(\frac{(X_a - D)P_s}{4(1-P_s)y}\right)^{-0.4} \left(\frac{X_a}{D*} - 1\right)^{-0.5} \tag{5.13}$$

where $$\left(\frac{D}{D*}\right) = \frac{(X_a - D)P_s + 1}{\frac{A}{A_t}}$$

Data from the same sources used for heat transfer correlations are plotted in Figure 22. Agreement is not as good as with the j factors, yet the data appears to be bracketed by ±35 percent of the mean. Once

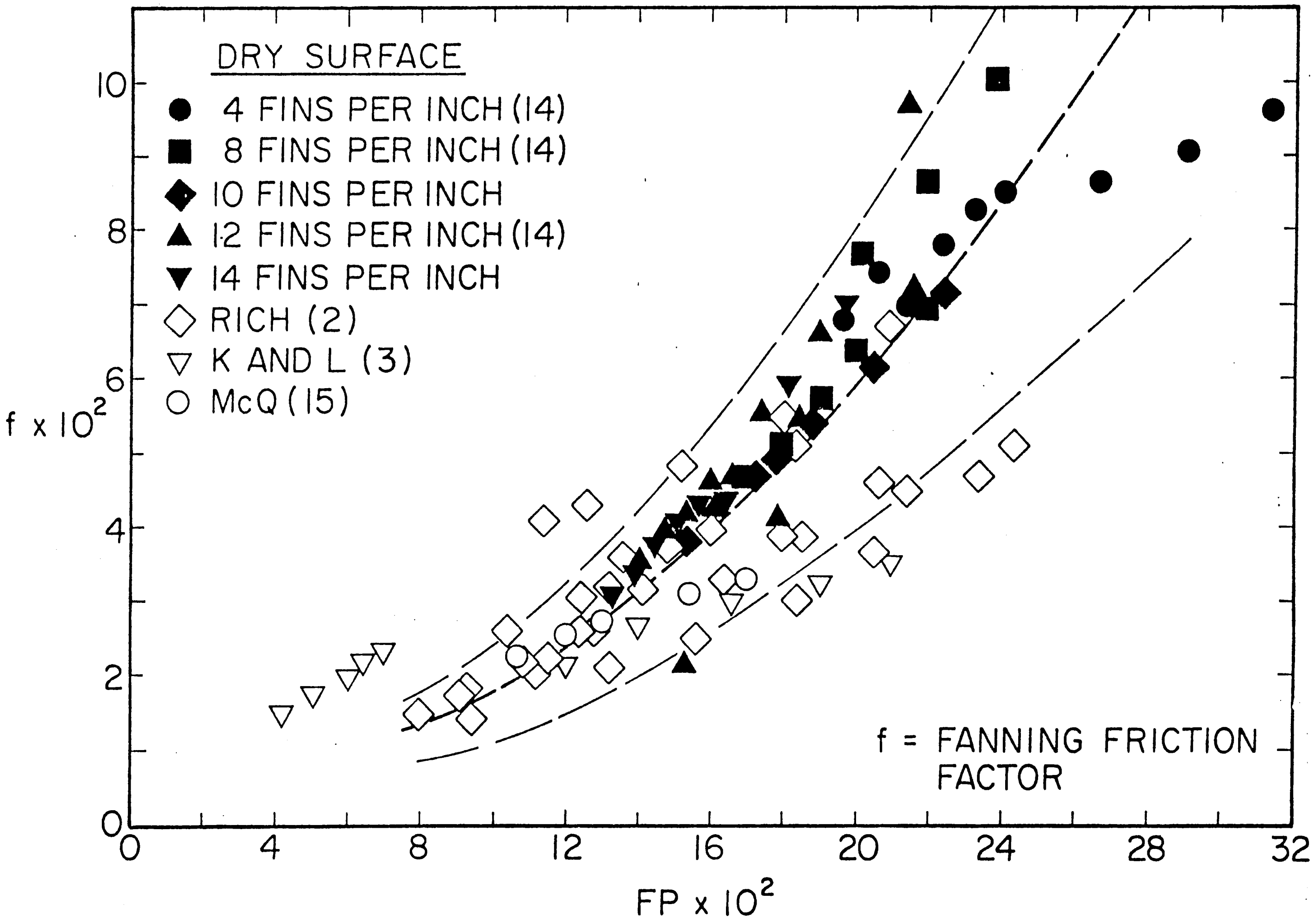

Figure 22. Correlation of Dry Surface Friction Data

again the data of Burchfield and this study behaves much better than the rest. Since the friction factors are highly dependent on the surface conditions it is not too surprising that the data are scattered. The exchangers used by Burchfield were industrial grade while those used by Rich (2) were laboratory grade with smooth, tinned, collarless fins metallurgically bonded to the tubes. The mean of the data of Figure 22 appears to lie along the line shown which may be represented by the equation

$$f = 4.094 \times 10^{-3} + 1.382(FP)^2 \tag{5.14}$$

when FP is in the range of 0.08 to 0.24. This correlation should be good for heat exchangers with three or more rows of tubes.

Mass Transfer Effects

The j factors obtained with wet surface conditions are shown in Figures 23 through 26 plotted versus the parameter JP. The effect of the condensate is apparent using this type of representation. The sensible j factors are greatly enhanced at the wider fin spacings and are depressed slightly, in relation to the dry surface, at the narrow fin spacing. The total j factors are also enhanced at the wide fin spacings and begin to follow the same trends as the sensible j factors; yet at the smaller fin spacings the total j factor once again increases substantially above the dry surface condition. It is thought that the interaction of the flow field and the condensate is responsible for the behavior of the total and sensible j factors at the smaller fin spacings. The increasing total j factor with depressed sensible j factors indicates that a greater percentage of the total heat transferred at the smaller fin spacings is latent heat.

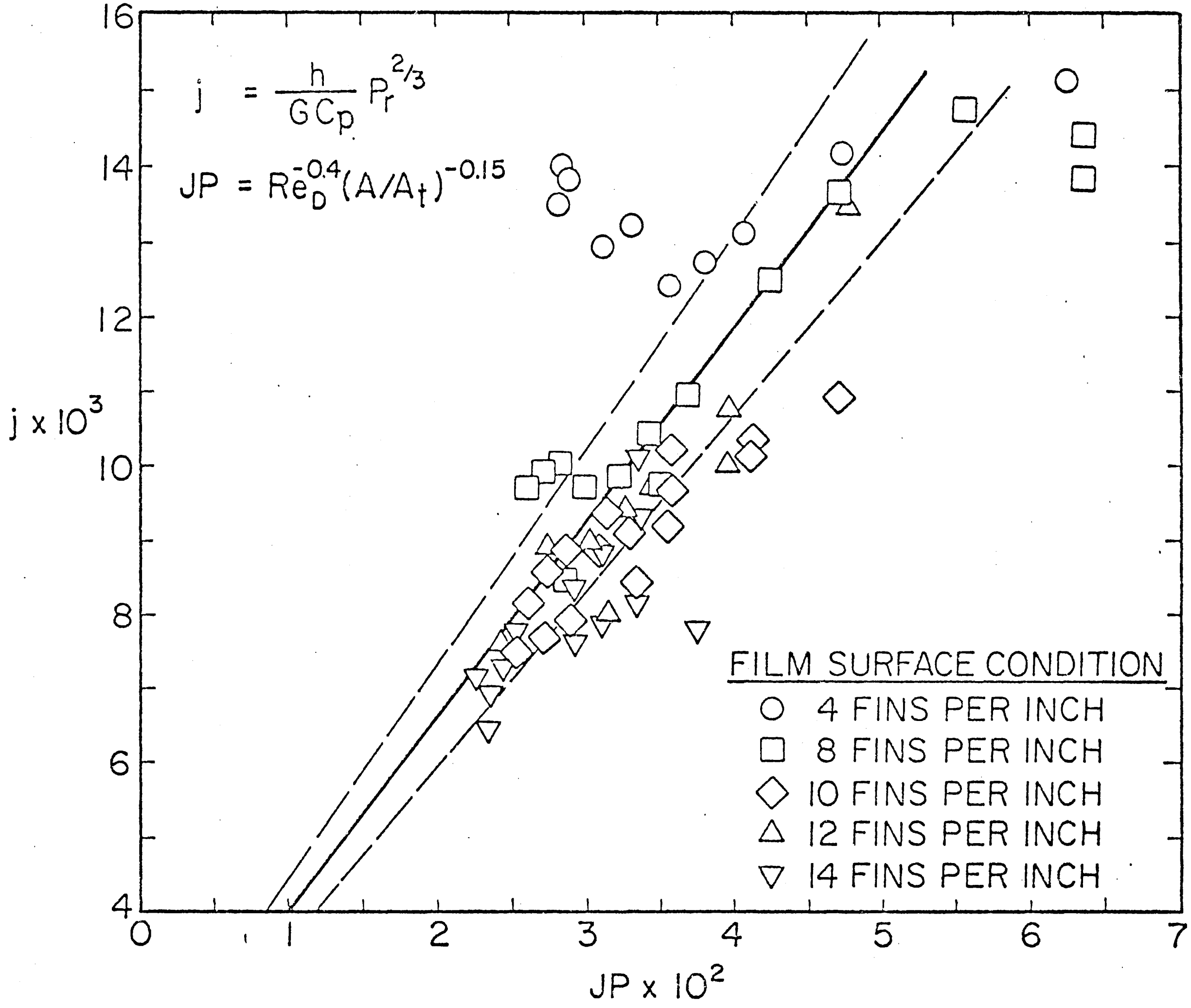

Figure 23. Effect of Film Condensation on Sensible j Factors

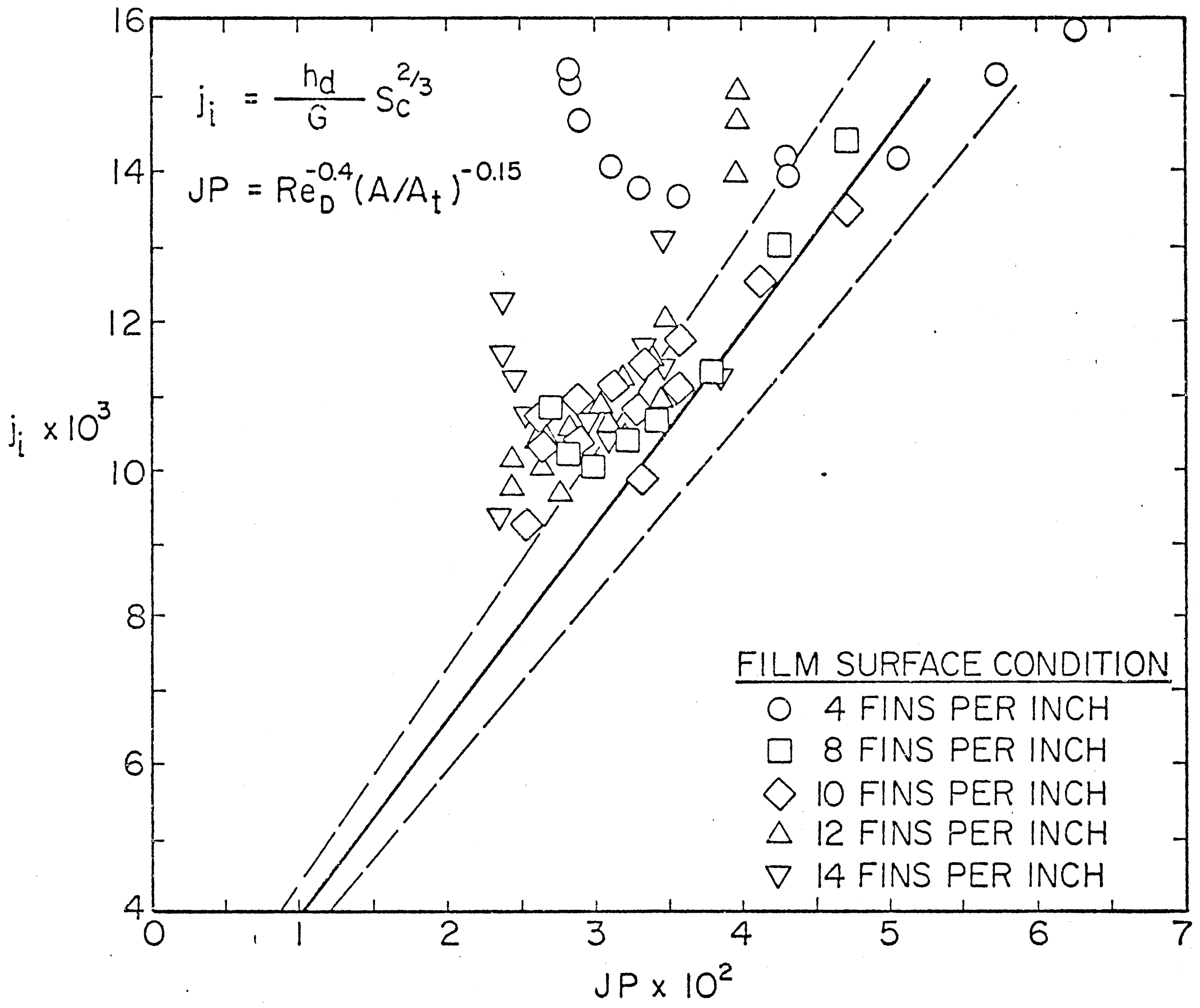

Figure 24. Effect of Film Condensation on Total j Factors

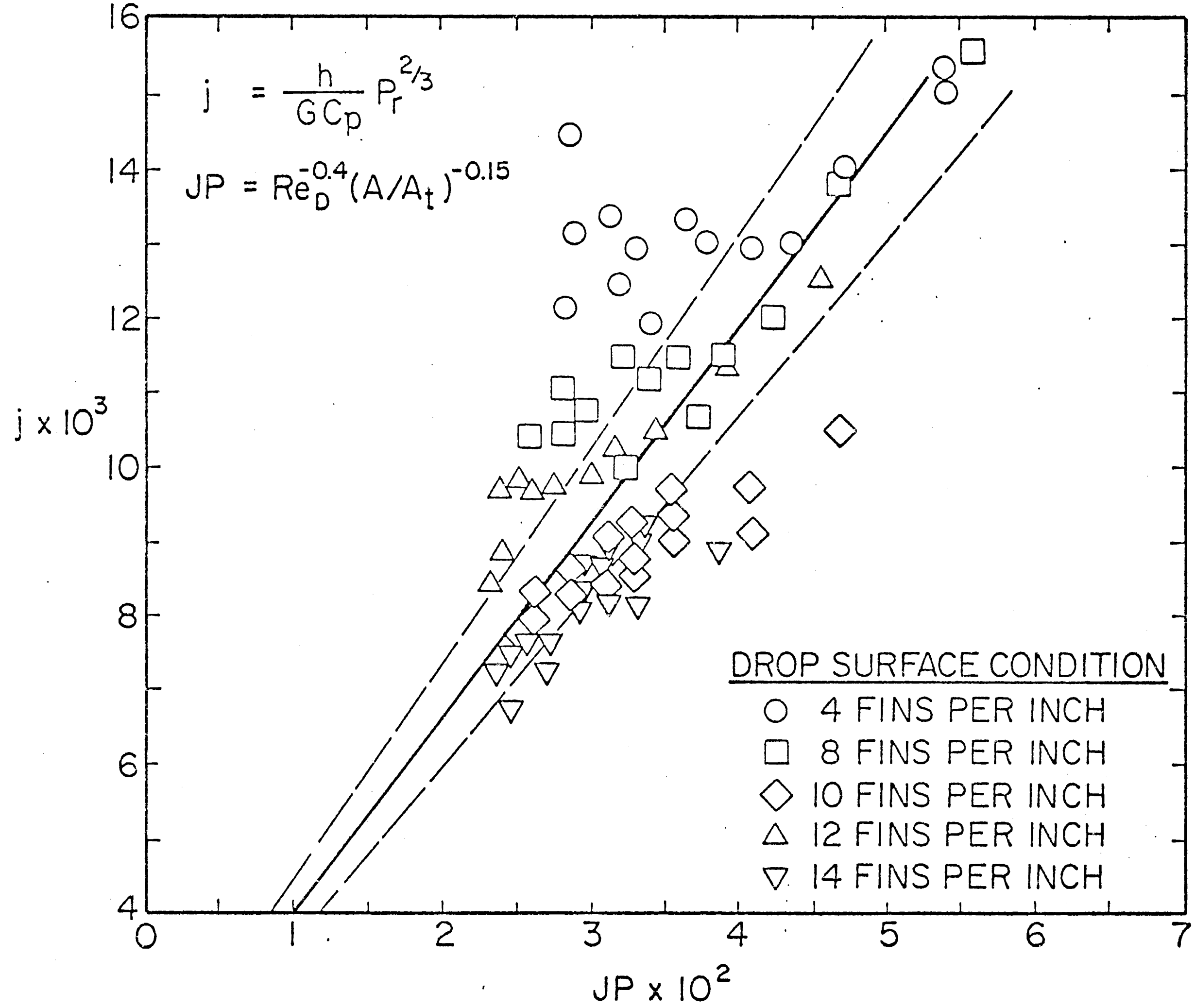

Figure 25. Effect of Drop Condensation on Sensible j Factors

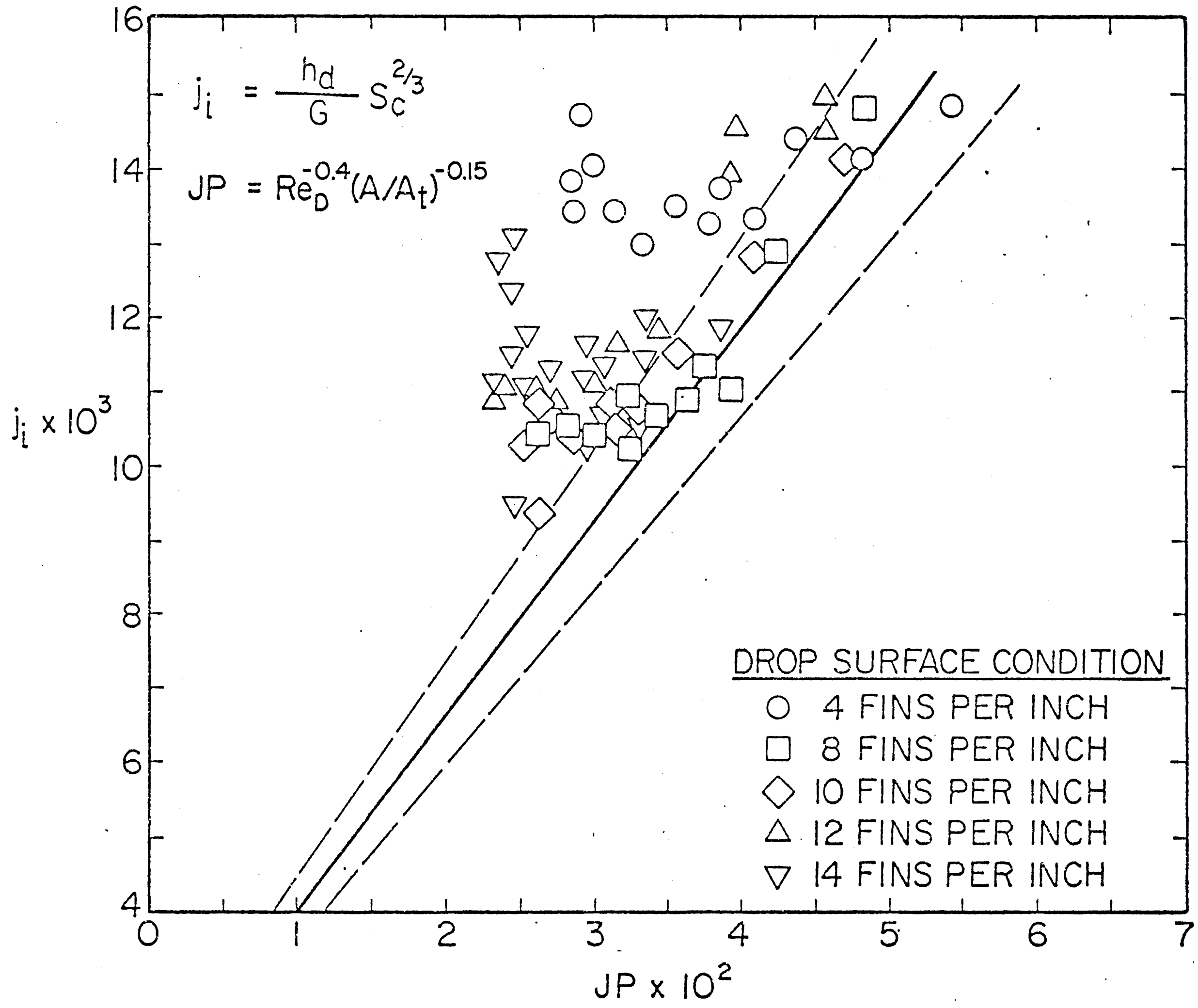

Figure 26. Effect of Drop Condensation on Total j Factors

Friction data for the film and drop condensation modes are presented in Figures 27 and 28. The increase in the friction factors with decreasing fin spacing is apparent and expected. The friction factors show the greatest increase at the lower Reynolds numbers and converge to the dry surface condition at the higher Reynolds numbers. The condensate has little effect at the wider fin spacings yet increases dramatically at the smaller fin spacings showing the increase in interaction between the condensate and the flow field.

From Figures 23 through 28 it is apparent that the fin spacing is a dominating factor in mass transfer. In order to account for the variation of the j factors with differing air velocity, a Reynolds number based on fin spacing is used.

$$Re_s = \frac{Gs}{\mu} \tag{5.15}$$

where s = center to center distance between the fins, $1/P_s$(ft/fin)

One additional parameter which is useful in mass transfer correlations is also based on fin spacing,

$$F_s = \frac{s}{s-y} \tag{5.16}$$

where y = fin thickness (ft)

This particular dimensionless parameter is constant for a specific heat exchanger; approaches one for large fin spacings; and increases with decreasing fin spacing. F_s varied from 1.0246 for 4 fins per inch to 1.0917 for 14 fins per inch. This parameter is most important with narrow fin spacing.

Correlations for the mass transfer j factors were sought in the form

$$j_{wet} = (j_{dry}) \{J(s)\} \tag{5.17}$$

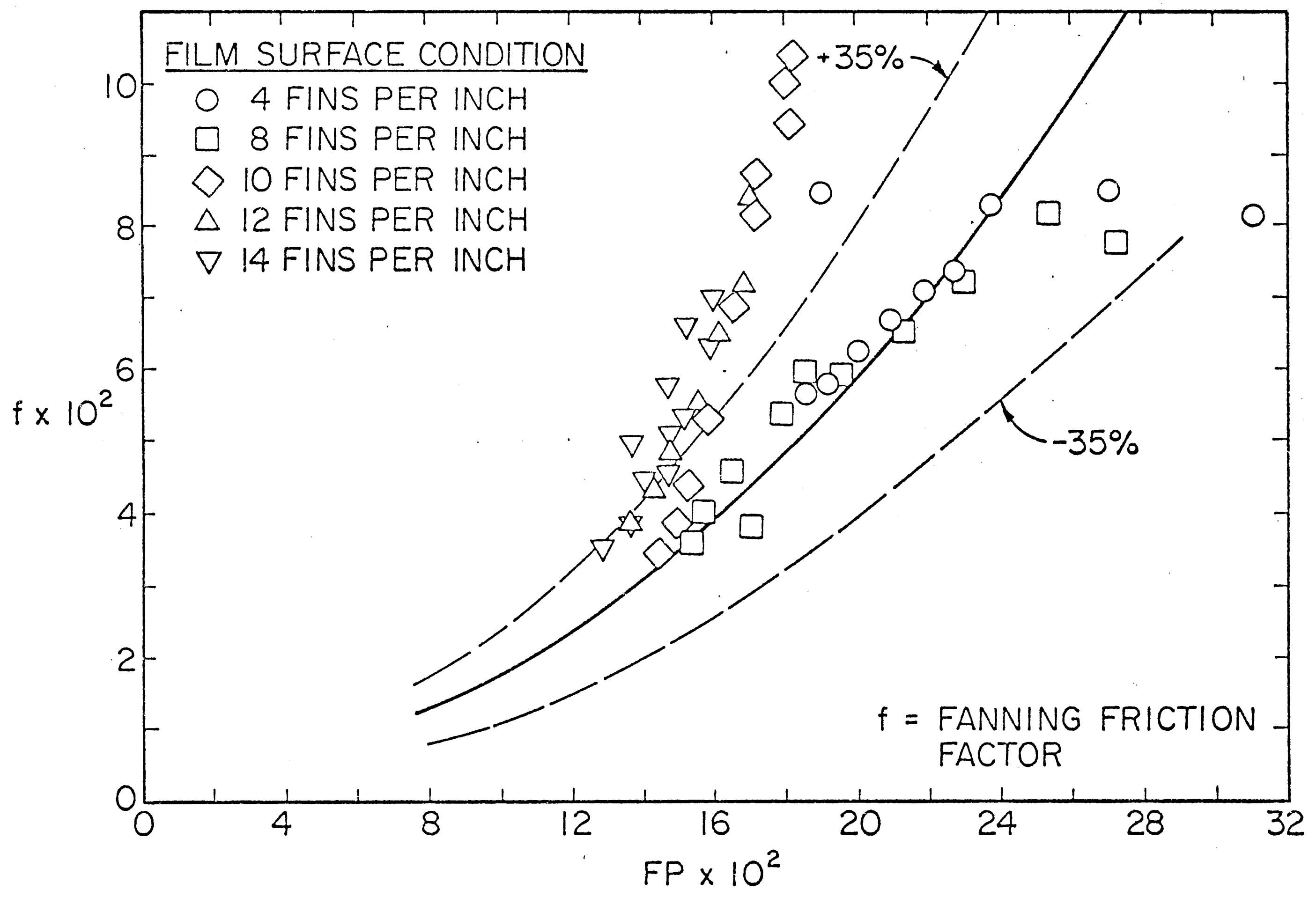

Figure 27. Effect of Film Condensation on Friction Factors

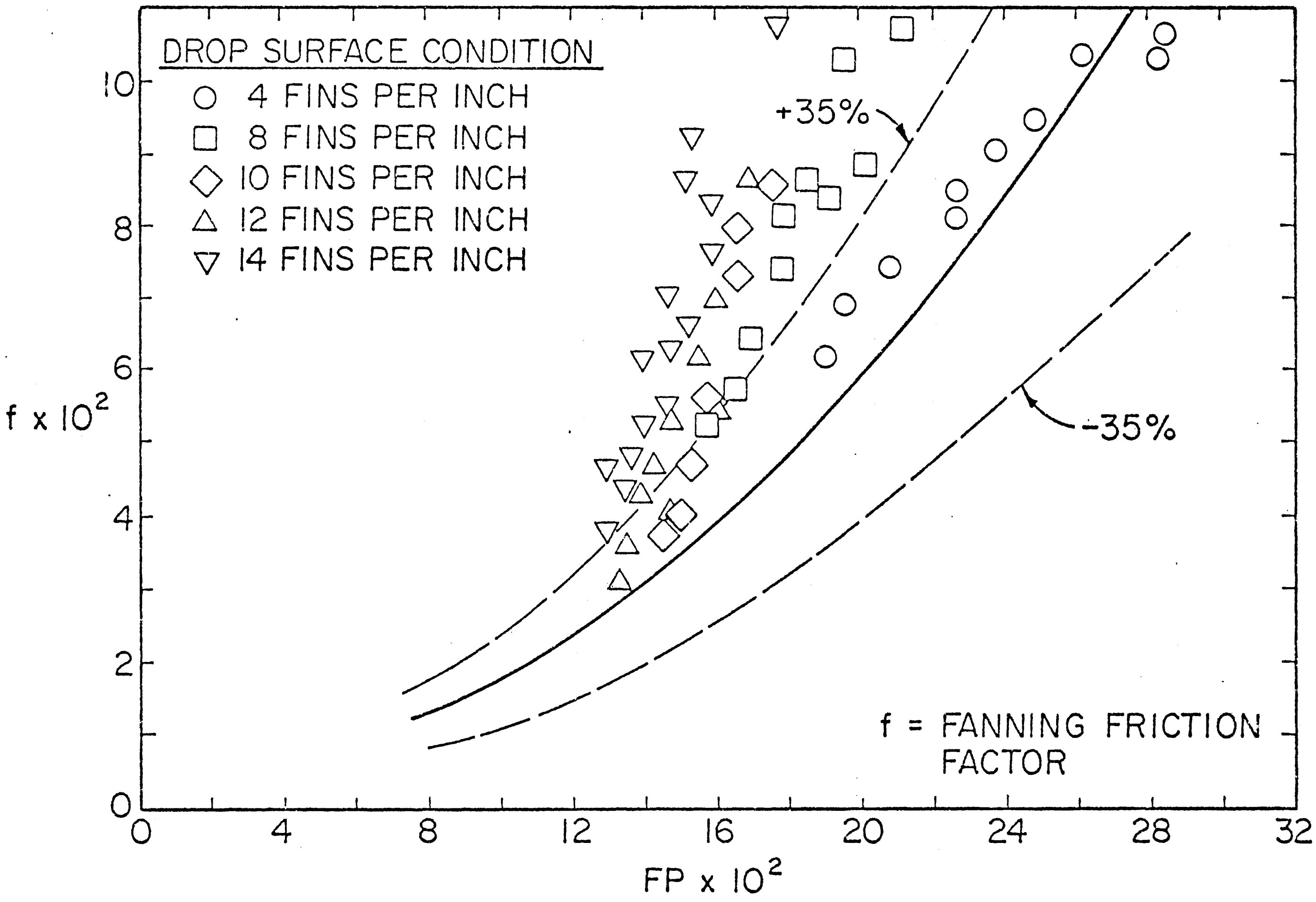

Figure 28. Effect of Drop Condensation on Friction Factors

where $\quad J(s) = (a + b\,Re_s^{\,n})F_s^{\,m}$ (5.18)

in which a, b, n, and m are constants. Similarly for the friction data

$$f_{wet} = (f_{dry})\ \{F(s)\} \tag{5.19}$$

where $\quad F(s) = (c + d\,Re_s^{\,p})F_s^{\,q}$ (5.20)

and c, d, p, and q are constants. Correlations of this form were selected so that J(s) and F(s) represent the relation between wet and dry surface factors. This facilitates the prediction of the wet surface factors from dry surface data. Results of these correlations are discussed in the following section.

Correlation of Wet Surface Data

Correlation of the sensible j factor with film condensation is shown in Figure 29. The correlating factor, J(s), is

$$J(s) = \left[0.84 + 4.0 \times 10^{-5}(Re_s)^{1.25}\right] \tag{5.21}$$

in which the exponent of F_s is equal to zero. Figure 29 shows that all but a few points are within ±10 percent and only the 14 fins per inch data at the highest Reynolds numbers fall outside this range.

The total j factors with film condensation are correlated in Figure 30 using the following correlation factor

$$J(s) = \left[0.95 + 4.0 \times 10^{-5}(Re_s)^{1.25}\right](F_s)^2 \tag{5.22}$$

Once again nearly all the data is bracketed by ±10 percent except the 14 fins per inch data at the higher Reynolds numbers.

When the form of condensation is dropwise the sensible j factors may be correlated using

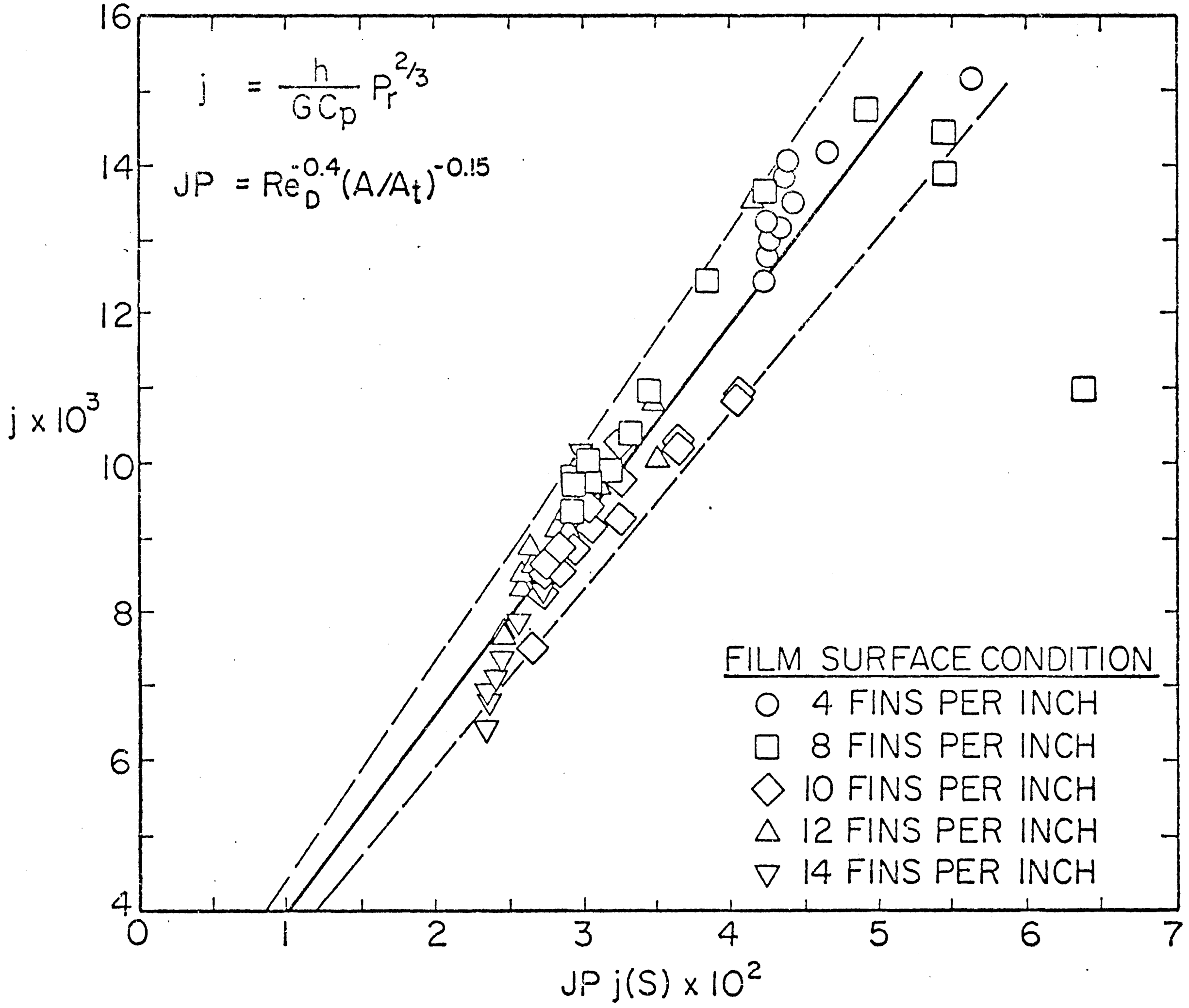

Figure 29. Mass Transfer (Film) Correlations for Sensible j Factors

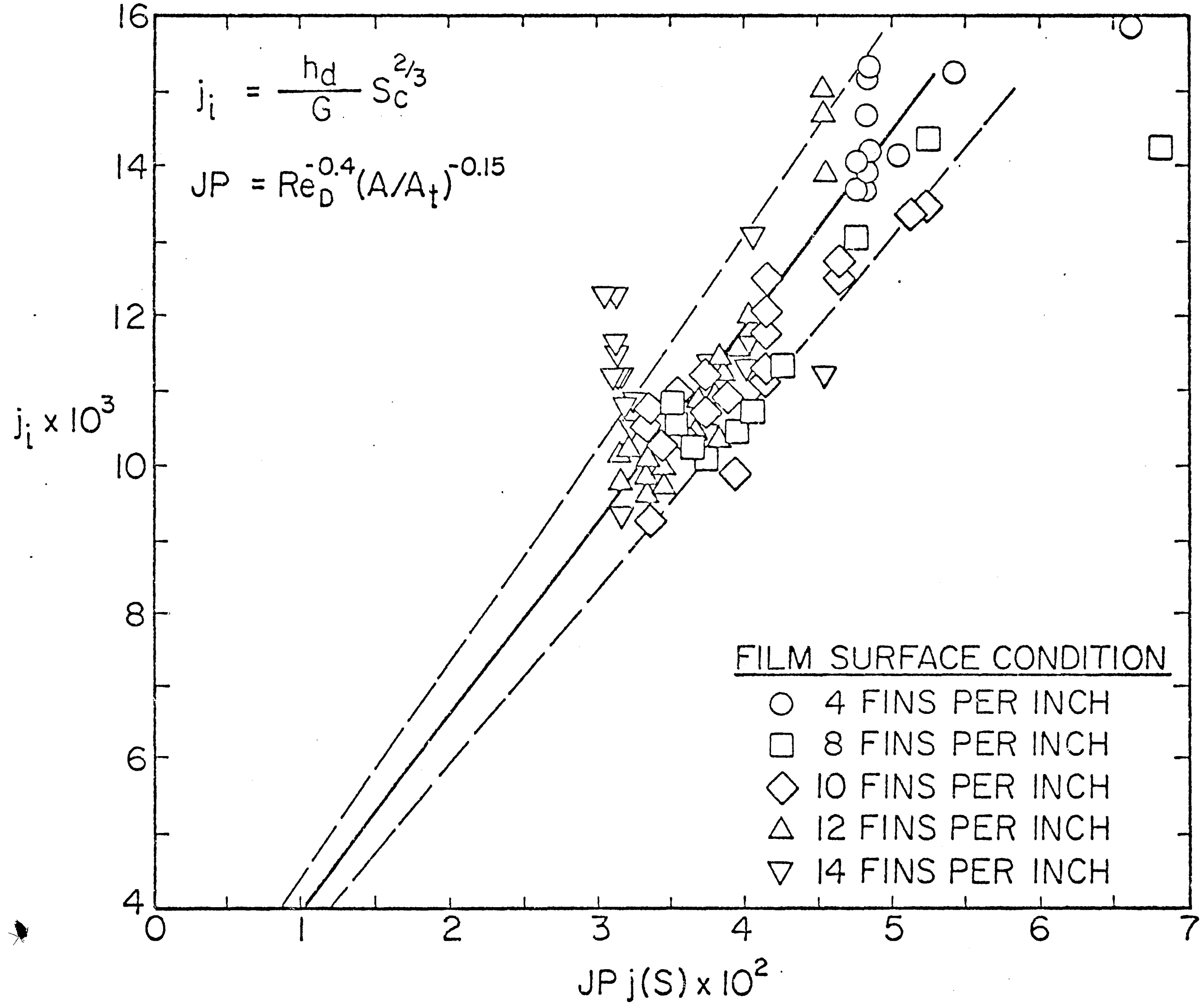

Figure 30. Mass Transfer (Film) Correlations for Total j Factors

$$J(s) = \left[0.9 + 4.3 \times 10^{-5}(Re_s)^{1.25}\right] F_s^{-1.0} \tag{5.23}$$

with the results shown in Figure 31. Agreement is not as good as for film condensation yet the bulk of the data is still within ±10 percent.

Total j factors with drop condensation are correlated using

$$J(s) = \left[0.8 + 4 \times 10^{-5}(Re_s)^{1.25}\right] F_s^{4.0} \tag{5.24}$$

and are plotted in Figure 32. As in the case of film condensation, agreement is good except for the 14 fins per inch data at high Reynolds numbers.

Only minor differences were noted in the friction data between drop and film condensation. This is reflected in the following correlations.

Correlation of the film condensation friction data, Figure 33, was obtained using

$$F(s) = \left[0.6 + Re_s^{-0.15}\right] F_s^{-3.0} \tag{5.25}$$

Only a few points at very low Reynolds numbers were not bracketed by ±35 percent.

Drop condensation friction data was correlated using

$$F(s) = \left[0.325 + Re_s^{-0.05}\right] F_s^{-3.0} \tag{5.26}$$

and is shown in Figure 34. Once again the data is bracketed by ±35 percent. However, the slope of the correlated data does not follow the dry correlation as desired.

The total j factor correlations are troublesom due to the increased heat transfer coefficient at the higher flow rates. It may be reasoned that within the range of practical application the correlations are quite acceptable since lower flow rates are used with higher density

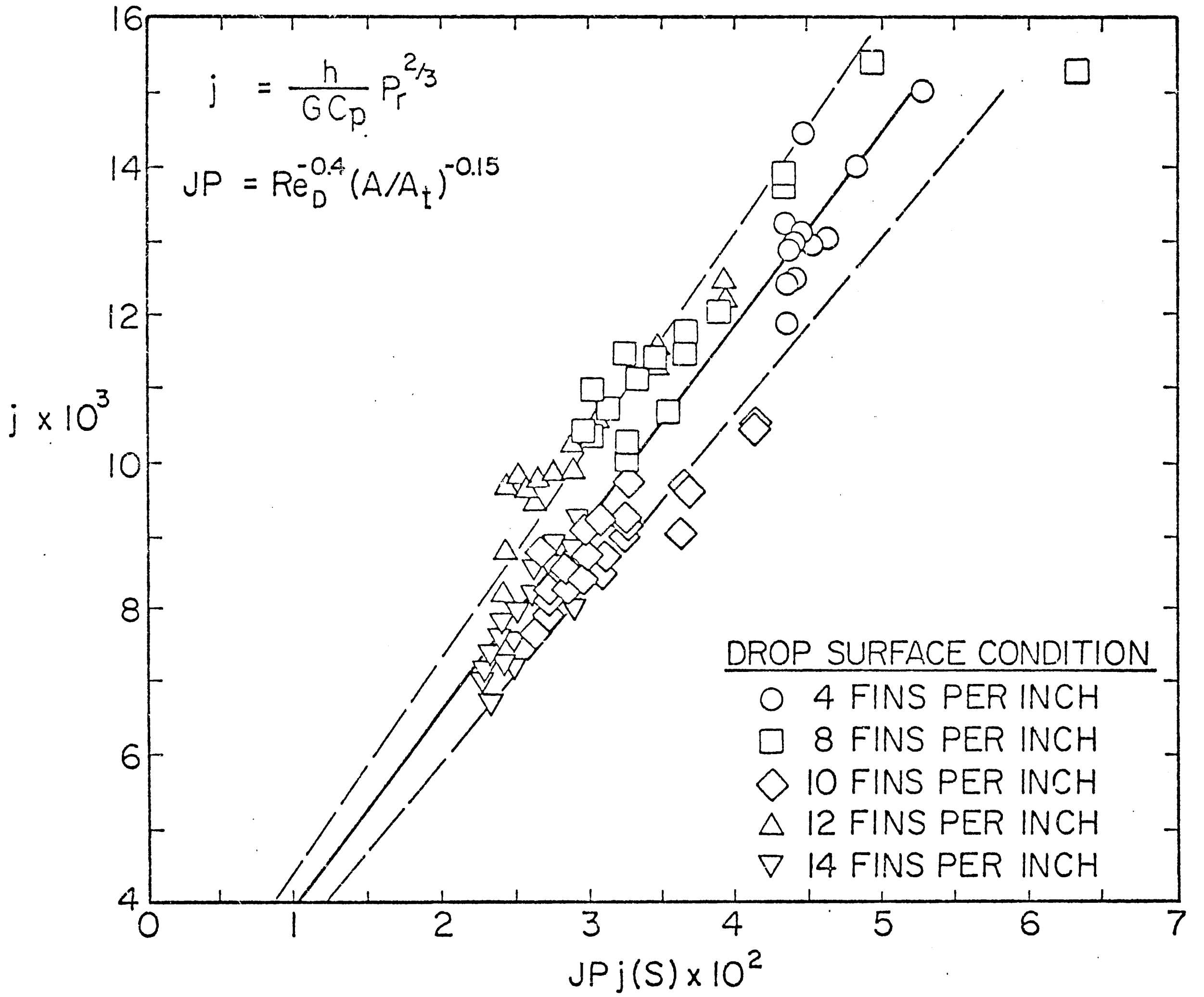

Figure 31. Mass Transfer (Drop) Correlations for Sensible j Factors

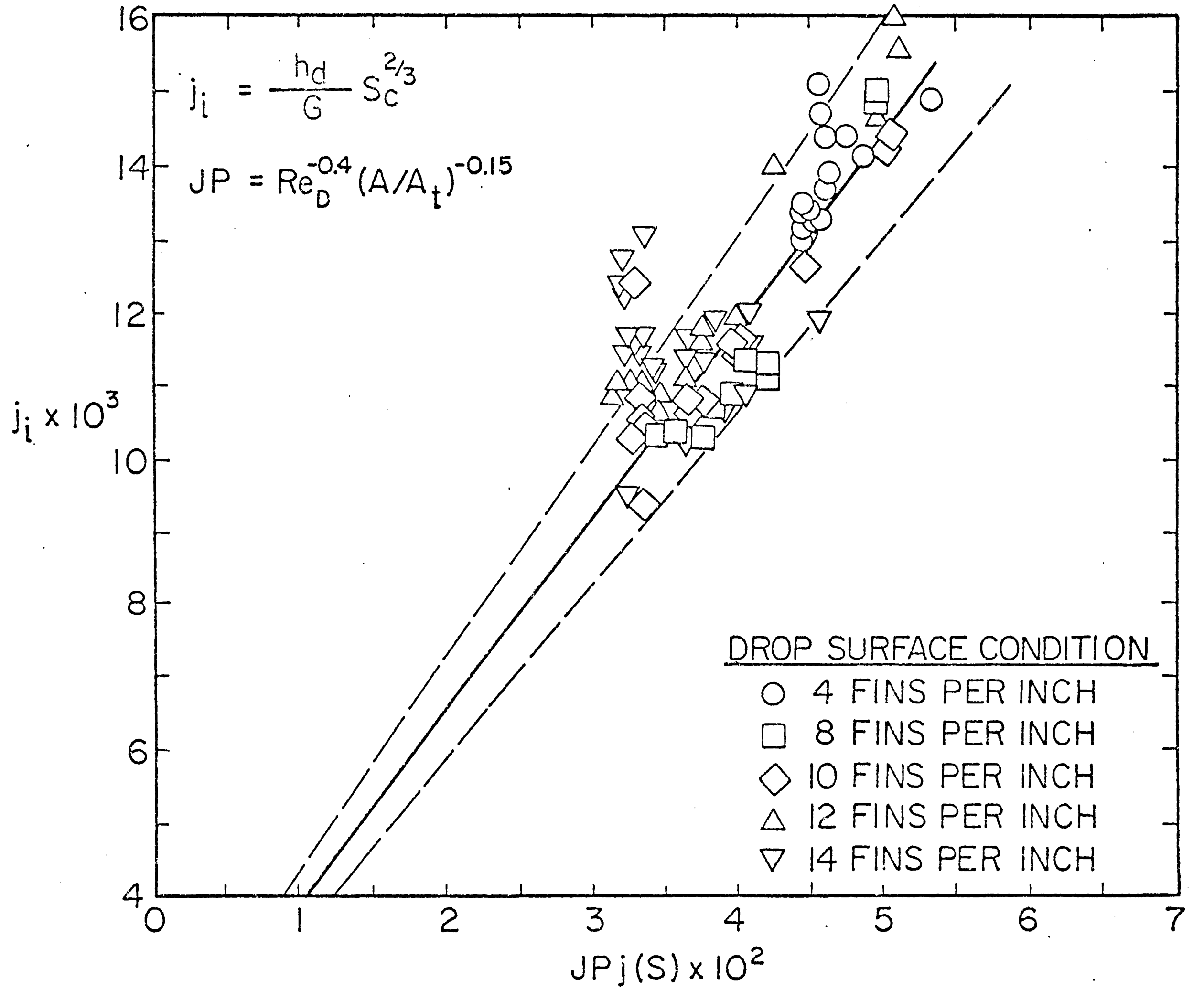

Figure 32. Mass Transfer (Drop) Correlations for Total j Factors

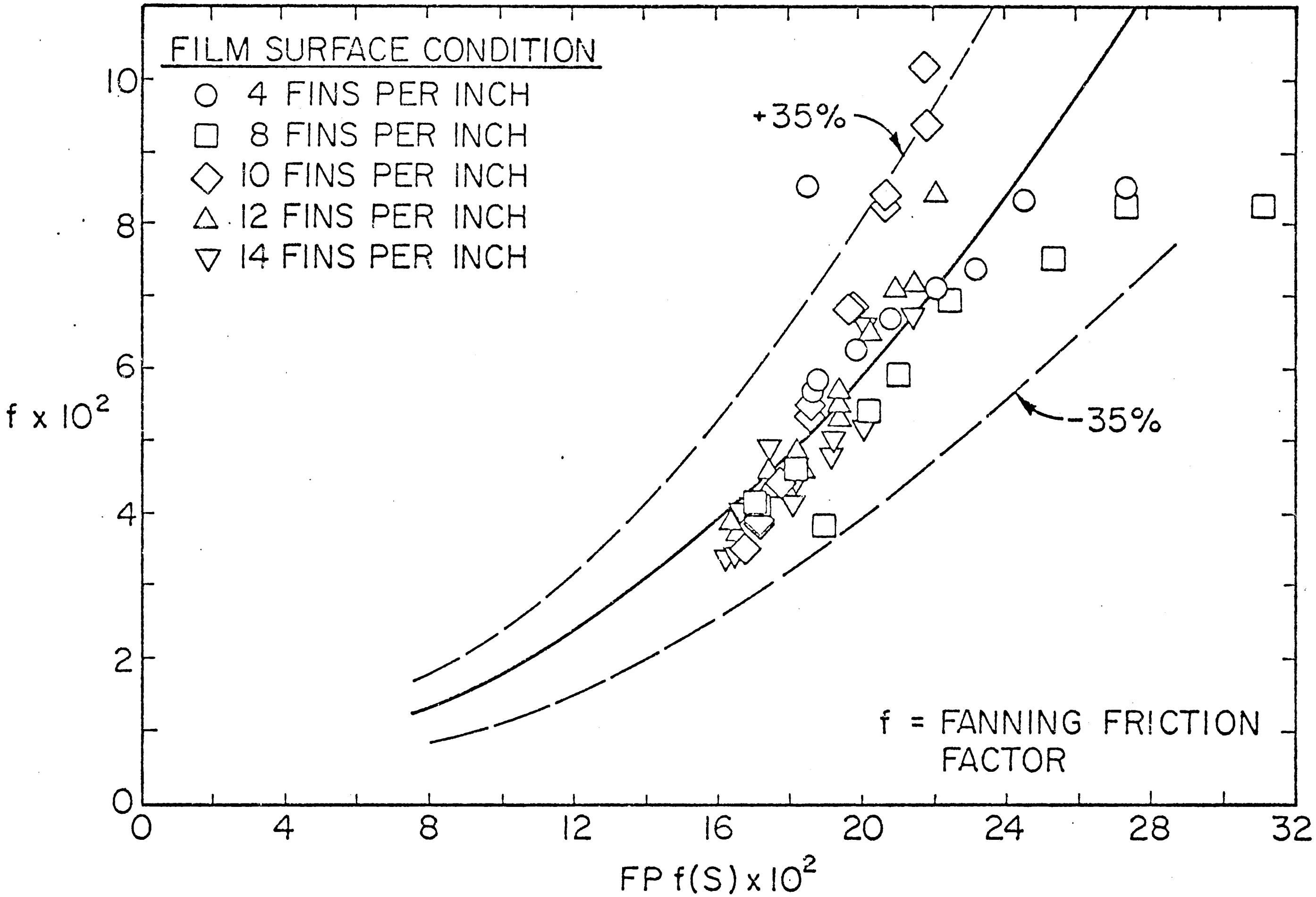

Figure 33. Mass Transfer (Film) Correlations for Friction Factors

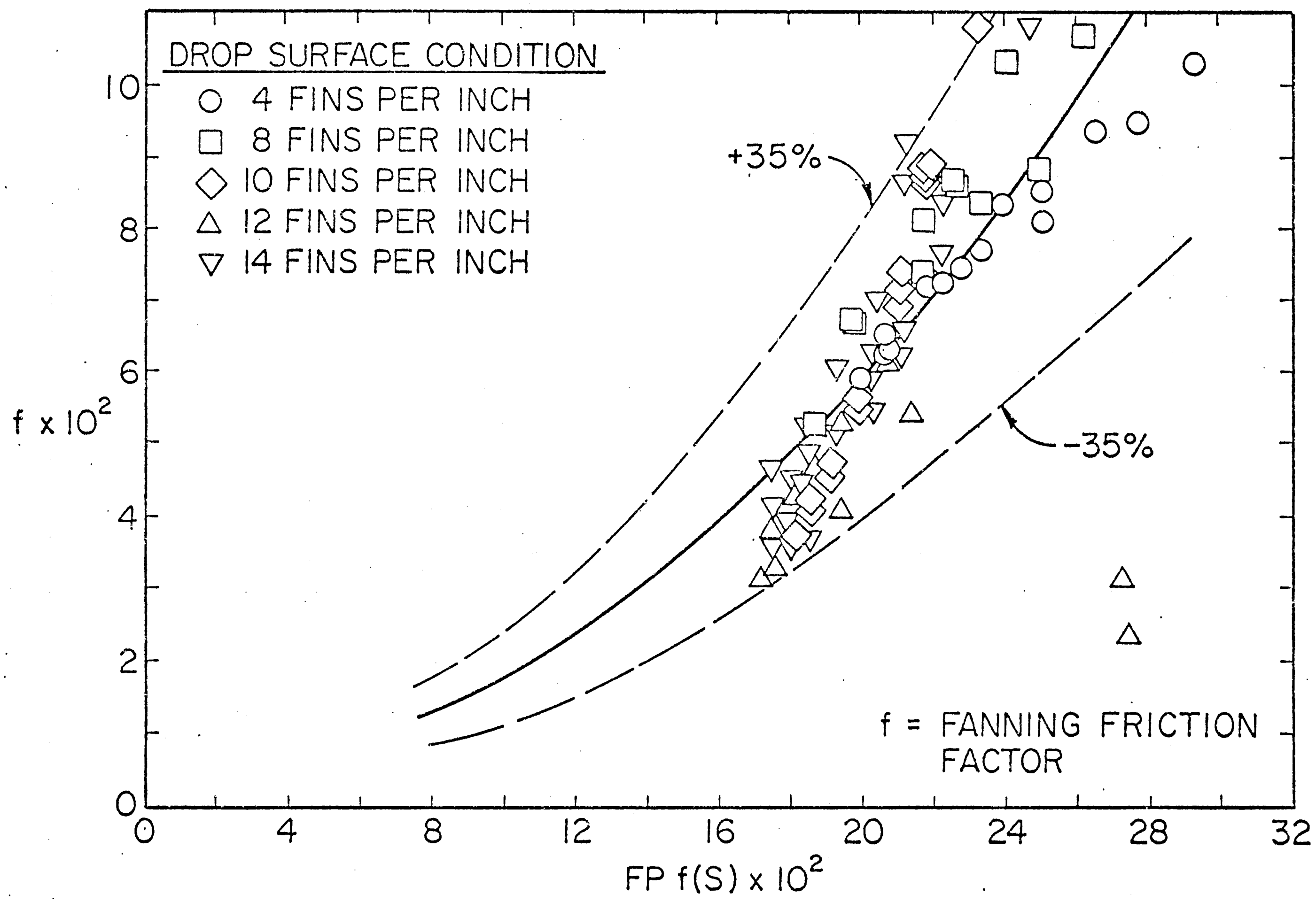

Figure 34. Mass Transfer (Drop) Correlations for Friction Factors

fins to avoid condensate blowing off the coil. It would seem, however, that some additional parameter becomes important as the fin pitch increases beyond 12 fins per inch. Further investigation of this effect is indicated.

CHAPTER VI

SUMMARY AND CONCLUSIONS

Correlations have been developed for heat, mass, and momentum transfer in plate-fin-tube heat exchangers. Almost all of the heat and mass transfer j factors were correlated to within ±10 percent of the observed values with the only significant deviations occurring with the total j factors for the 14 fins per inch heat exchanger at the high Reynolds numbers.

Correlations for the sensible j factors with no mass transfer, dry surface, were excellent for a four tube row coil. Using the results of Rich (1), the correlation was extended to a multi-row coil resulting in equation (5.12) which expresses the j factor as a function of tube rows and Reynolds number based on longitudinal tube spacing.

Correlations sought for mass transfer j factors were of the form

$$\frac{j_{wet}}{j_{dry}} = J(s) \qquad (5.17)$$

in which J(s) was a strong function of fin spacing.

To summarize:

Sensible Heat Transfer Factors

$$J(s)_{Film} = \left[0.84 + 4.0 \times 10^{-5}(Re_s)^{1.25}\right] \qquad (5.21)$$

$$J(s)_{Drop} = \left[0.9 + 4.3 \times 10^{-5}(Re_s)^{1.25}\right] F_s^{-1.0} \qquad (5.23)$$

Total Heat Transfer Factors

$$J(s)_{Film} = \left[0.95 + 4.0 \times 10^{-5}(Re_s)^{1.25}\right] F_s^{2.0} \tag{5.22}$$

$$J(s)_{Drop} = \left[0.8 + 4.0 \times 10^{-5}(Re_s)^{1.25}\right] F_s^{4.0} \tag{5.24}$$

where Re_s and F_s are defined by equations (5.15) and (5.16) respectively.

Dry surface friction data for the plate-fin-tube heat exchanger was correlated with success to within ±35 percent of the observed values. This correlation, good for three or more tube row exchangers, is given by equation (5.14).

Friction factors with mass transfer have also been correlated to within ±35 percent. The form of the correlation is the same as that for heat and mass transfer

$$\frac{f_{wet}}{f_{dry}} = F(s) \tag{5.19}$$

in which F(s) is also a strong function of fin spacing. The correlating factor is

$$F(s)_{Film} = \left[0.6 + Re_s^{-0.15}\right] F_s^{-3.0} \tag{5.25}$$

$$F(s)_{Dry} = \left[0.325 + Re_s^{-0.05}\right] F_s^{-3.0} \tag{5.26}$$

where Re_s and F_s are again defined by equations (5.15) and (5.16) respectively.

Improvement to these correlations may be obtained if a purely statistical approach were taken. Further testing would be required to isolate the appropriate parameter to account for the deviation of the 14 fins per inch data. This parameter may be a property of the coil geometry yet is more likely to be hydrodynamic in nature. It is recommended that further investigation be undertaken as suggested above.

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APPENDIXES

TABLE I

NOMENCLATURE FOR REDUCED DATA

Date	Month/Day/Year
Barometric Pressure	Inches of mercury
Ambient Temperature	Degrees Fahrenheit, ^oF
TDB1	Entering air dry bulb temperature, ^oF
TWB1	Entering air wet bulb temperature, ^oF
TDB2	Exiting air dry bulb temperature, ^oF
TWB2	Exiting air wet bulb temperature, ^oF
TWA1	Entering water temperature, ^oF
TWA2	Exiting water temperature, ^oF
QDOT	Total heat transfer, BTU/hour
XJ	Sensible j factor
XJI	Total j factor
F	Fanning friction factor
RE	Reynolds number based on hydraulic diameter
RED	Reynolds number based on tube diameter
RES	Reynolds number based on fin spacing
REB	Reynolds number based on tube spacing
JP	Heat transfer correlation parameter
FP	Friction factor correlation parameter

TABLE II

DROP TEST DATA - 10 FINS PER INCH

TUBE O. D. =0.3920 (IN) FIN PITCH = 10 (FINS/IN) HYDPAULIC DIA. =0.01027 (FT) FIN THICKNESS =0.0060 (IN)

FREE FLOW/FRONTAL AREA =0.572 TOTAL AREA/VOLUME = 222.6 (SQ. FT/CU. FT) FIN AREA/TOTAL AREA = 0.928 TUBES IN FACE = 5

TRANS. TUBE SPACING =1.000 (IN) LONG. TUBE SPACING =0.866 (IN) NUMBER OF ROWS = 4 FINNED LENGTH = 12 (IN)

RUN SERIES 40001 CATE 6/29/77 BAROMETRIC PRESSURE 29.13 AMBIENT TEMPERATURE 88.0

RUN	TDB1	TWB1	TDB2	TWB2	TWA1	TWA2	QDOT	XJ	XJI	F	RE	RED	RES	REB	JP	FP
1	79.1	71.2	52.2	50.9	36.3	37.9	8633.3	0.00880	0.01074	0.08559	605.8	1927.0	491.6	4257.1	0.0330	0.1724
2	79.7	72.5	51.1	49.9	36.3	37.9	7897.1	0.00932	0.01157	0.11216	496.4	1579.0	402.8	3488.3	0.0358	0.1812
3	79.2	75.6	49.3	48.5	35.9	37.3	6914.7	0.00975	0.01286	0.15479	352.9	1122.4	286.3	2479.7	0.0410	0.1973
4	79.7	77.9	47.4	46.4	35.4	36.6	5732.2	0.01053	0.01419	0.20260	250.5	796.8	203.3	1760.3	0.0470	0.2150
5	79.5	75.8	49.9	48.1	35.3	36.7	7041.7	0.00912	0.01273	0.16010	352.4	1121.0	286.0	2476.5	0.0410	0.1974
6	79.7	73.7	51.1	50.1	35.6	37.0	8478.2	0.00901	0.01162	0.11137	502.5	1598.4	407.8	3531.1	0.0356	0.1806
7	79.8	72.6	52.7	51.3	36.2	37.7	9227.1	0.00854	0.01099	0.08646	604.9	1924.2	490.9	4251.0	0.0330	0.1724
8	79.8	72.2	53.6	52.6	36.7	38.4	9913.4	0.00846	0.01057	0.07294	700.2	2227.4	568.2	4920.6	0.0312	0.1662
9	80.4	72.4	55.3	54.5	37.7	39.6	11220.1	0.00830	0.01046	0.05614	849.5	2702.1	689.3	5969.3	0.0288	0.1584
10	80.5	72.3	56.1	55.8	38.6	40.5	12002.3	0.00844	0.01051	0.04719	971.4	3090.0	788.3	6826.4	0.0273	0.1532
11	80.6	72.6	57.2	57.0	39.4	41.4	12312.5	0.00836	0.01059	0.04060	1084.9	3450.9	880.3	7623.6	0.0262	0.1490
12	80.6	73.2	57.8	57.6	38.5	39.8	14222.7	0.00769	0.01031	0.03768	1186.8	3775.2	963.1	8340.1	0.0252	0.1457
13	80.5	72.8	57.4	57.2	39.4	41.4	12863.7	0.00817	0.01048	0.04008	1084.7	3450.3	880.2	7622.3	0.0262	0.1490
14	80.1	72.6	56.9	56.7	39.8	41.9	11658.7	0.00830	0.01034	0.04555	973.1	3095.4	789.7	6838.4	0.0273	0.1531
15	79.9	72.5	56.1	55.9	40.2	42.0	10565.1	0.00865	0.01055	0.05520	851.8	2709.5	691.2	5985.8	0.0288	0.1583
16	79.7	72.5	54.9	54.7	40.3	42.0	9202.7	0.00914	0.01090	0.06986	698.8	2222.7	567.0	4910.5	0.0312	0.1663

TABLE II (Continued)

RUN SERIES 40002 CATE 6/30/77 BAROMETRIC PPESSURE 28.94 AMBIENT TEMPERATURE 88.0

RUN	TDB1	TWB1	TDB2	TWB2	TWA1	TWA2	QDOT	XJ	XJI	F	RE	RED	RES	REB	JP	FP
1	80.5	72.6	53.8	52.4	37.7	39.5	8811.1	0.00871	0.01081	0.08585	601.5	1913.4	488.1	4227.0	0.0331	0.1727
2	80.5	73.4	52.6	51.3	37.8	39.3	7871.7	0.00918	0.01152	0.10865	493.1	1568.6	400.1	3465.3	0.0358	0.1815
3	79.7	76.2	50.9	50.0	37.7	39.1	6790.4	0.00964	0.01231	0.15012	350.7	1115.4	284.5	2464.1	0.0411	0.1976
4	80.3	78.7	48.8	47.9	37.2	38.3	5736.0	0.01060	0.01442	0.17778	249.0	791.9	202.0	1749.5	0.0471	0.2153
5	80.1	77.0	49.7	49.0	35.9	37.3	7267.0	0.00958	0.01301	0.14733	351.2	1117.2	285.0	2468.2	0.0411	0.1975
6	79.5	72.5	50.2	49.7	35.9	37.5	7952.9	0.00977	0.01173	0.11732	495.6	1576.5	402.2	3482.8	0.0358	0.1812
7	79.5	71.7	52.0	51.4	36.7	38.5	8760.0	0.00928	0.01100	0.08817	608.8	1936.7	494.0	4278.4	0.0330	0.1722
8	79.4	71.7	52.9	52.6	37.3	38.9	9478.3	0.00914	0.01080	0.07483	694.8	2210.2	563.8	4882.7	0.0313	0.1666
9	79.5	71.7	54.5	54.3	37.9	39.8	10631.9	0.00876	0.01050	0.05687	845.1	2688.0	685.7	5938.3	0.0289	0.1586
10	79.8	72.0	56.2	56.0	38.9	40.9	11621.0	0.00838	0.01027	0.04687	974.6	3100.1	790.9	6848.8	0.0273	0.1530
11	80.4	73.0	57.2	57.2	39.4	41.4	13064.9	0.00833	0.01098	0.04207	1081.4	3439.9	877.5	7599.3	0.0262	0.1491
12	80.5	72.7	57.3	57.2	40.3	42.3	13957.7	0.00888	0.01237	0.03829	1179.0	3750.2	956.7	8284.8	0.0253	0.1459
13	80.1	72.5	56.4	57.0	37.0	39.4	12791.4	0.00793	0.00939	0.04190	1093.8	3479.2	887.5	7686.1	0.0261	0.1487
14	80.0	72.4	56.1	55.8	38.2	40.4	12006.2	0.00822	0.01033	0.04761	971.5	3090.3	788.3	6827.0	0.0273	0.1532
15	80.1	72.4	55.5	55.3	39.2	41.2	10742.4	0.00877	0.01068	0.05500	846.8	2693.4	687.1	5950.3	0.0289	0.1585
16	79.9	72.5	54.6	54.4	39.6	41.4	9217.5	0.00912	0.01080	0.07109	692.6	2203.1	562.0	4867.0	0.0313	0.1667

TABLE III

DRY TEST DATA - 10 FINS PER INCH

TUBE O. D. =0.3920 (IN)   FIN PITCH = 10 (FINS/IN)   HYDRAULIC DIA. =0.01027 (FT)   FIN THICKNESS =0.0060 (IN)

FREE FLOW/FRONTAL AREA =0.572   TOTAL AREA/VOLUME = 222.6 (SQ. FT/CU. FT)   FIN AREA/TOTAL AREA = 0.928   TUBES IN FACE = 5

TRANS. TUBE SPACING =1.000 (IN)   LONG. TUBE SPACING =0.866 (IN)   NUMBER OF ROWS = 4   FINNED LENGTH = 12 (IN)

RUN SERIES 40100   CATE 6/27/77   BAROMETRIC PRESSURE 28.94   AMBIENT TEMPERATURE 80.0

RUN	TDB1	TWB1	TDB2	TWB2	TWA1	TWA2	QDOT	XJ	XJI	F	RE	RED	RES	REB	JP	FP
1	80.8	71.6	136.8	86.9	158.4	156.8	6411.9	0.01087	0.00000	0.05452	430.5	1369.3	349.3	3025.1	0.0379	0.1877
2	79.9	71.8	140.6	88.7	158.8	157.7	4917.5	0.01204	0.00000	0.06108	303.4	965.2	246.2	2132.2	0.0435	0.2049
3	80.1	72.0	143.1	89.6	159.3	158.8	3604.1	0.01258	0.00000	0.07122	213.9	680.5	173.6	1503.2	0.0501	0.2236
4	80.3	71.1	142.2	89.1	160.2	158.8	4999.8	0.01227	0.00000	0.06119	302.6	962.5	245.5	2126.4	0.0436	0.2050
5	81.9	70.7	139.0	87.5	160.6	158.8	6524.0	0.01102	0.00000	0.05326	428.7	1363.6	347.9	3012.4	0.0379	0.1879
6	82.4	70.7	137.0	87.5	160.6	158.8	7644.5	0.01043	0.00000	0.04989	525.5	1671.4	426.4	3692.4	0.0350	0.1786
7	83.0	70.9	135.0	87.1	160.9	158.8	8387.0	0.00971	0.00000	0.04660	607.0	1930.8	492.5	4265.5	0.0330	0.1723
8	83.7	70.9	132.1	85.9	160.9	158.6	9585.2	0.00898	0.00000	0.04250	746.3	2373.9	605.6	5244.5	0.0304	0.1636
9	84.6	71.3	130.0	84.6	160.6	158.4	10281.6	0.00831	0.00000	0.04030	854.2	2717.1	693.1	6002.5	0.0288	0.1582
10	86.2	72.0	123.9	84.4	160.6	157.9	10864.5	0.00792	0.00000	0.03812	959.2	3050.9	778.3	6740.1	0.0275	0.1537
11	86.9	72.7	127.6	84.2	160.6	157.7	11258.7	0.00751	0.00000	0.03773	1043.4	3319.0	846.7	7332.2	0.0266	0.1505
12	86.2	72.5	128.9	84.4	160.2	157.5	10769.7	0.00801	0.00000	0.03773	950.5	3023.3	771.2	6679.0	0.0276	0.1540
13	85.3	71.6	129.6	84.8	160.0	157.5	10063.2	0.00823	0.00000	0.04062	856.8	2725.5	695.3	6021.1	0.0287	0.1581
14	84.2	71.1	131.6	85.5	159.7	157.5	9360.0	0.00892	0.00000	0.04252	743.5	2365.0	603.3	5224.8	0.0304	0.1638
15	82.8	70.9	134.1	86.2	159.7	157.7	8278.6	0.00970	0.00000	0.04724	607.9	1933.8	493.3	4272.0	0.0330	0.1722
16	82.1	70.7	135.9	86.9	159.7	157.9	7459.2	0.01024	0.00000	0.04919	522.0	1660.4	423.6	3668.1	0.0350	0.1789

TABLE IV

FILM TEST DATA - 10 FINS PER INCH

TUBE O. D. =0.3920 (IN)   FIN PITCH = 10 (FINS/IN)   HYDRAULIC DIA. =0.01027 (FT)   FIN THICKNESS =0.0060 (IN)

FREE FLOW/FRONTAL AREA =0.572   TOTAL AREA/VOLUME = 222.6 (SQ. FT/CU. FT)   FIN AREA/TOTAL AREA = 0.928   TUBES IN FACE = 5

TRANS. TUBE SPACING =1.000 (IN)   LONG. TUBE SPACING =0.866 (IN)   NUMBER OF ROWS = 4   FINNED LENGTH = 12 (IN)

RUN SERIES 41001   DATE 7/ 6/77   BAROMETRIC PRESSURE 29.13   AMBIENT TEMPERATURE 88.0

RUN	TDB1	TWB1	TDB2	TWB2	TWA1	TWA2	QDOT	XJ	XJI	F	RE	RED	RES	REB	JP	FP
1	80.5	71.8	51.2	49.7	36.0	37.4	7638.4	0.00927	0.01112	0.09429	496.0	1577.7	402.5	3485.4	0.0358	0.1812
2	80.5	71.6	52.1	50.9	36.2	37.6	8804.6	0.00911	0.01084	0.08210	605.3	1925.4	491.2	4253.5	0.0330	0.1724
3	81.2	71.9	53.5	52.3	36.7	33.3	9875.1	0.00884	0.01067	0.06924	704.0	2239.2	571.2	4946.8	0.0311	0.1660
4	81.5	72.6	55.5	54.7	37.9	39.7	11236.9	0.00853	0.01044	0.05355	848.6	2699.1	688.5	5962.8	0.0289	0.1584
5	81.4	72.6	56.3	55.9	38.7	40.6	12157.1	0.00852	0.01063	0.04422	970.6	3087.5	787.6	6820.8	0.0273	0.1532
6	81.9	73.3	57.8	57.6	39.9	42.0	13042.0	0.00847	0.01074	0.03909	1077.4	3427.0	874.2	7570.9	0.0262	0.1493
7	82.1	73.1	58.0	57.7	36.8	39.2	14059.1	0.00757	0.00924	0.03561	1189.0	3781.9	964.8	8354.9	0.0252	0.1456
8	82.0	73.2	57.1	56.8	37.7	39.9	13659.5	0.00320	0.01033	0.03893	1094.6	3481.8	888.2	7692.0	0.0261	0.1487
9	81.8	73.1	56.5	56.2	38.5	40.4	12430.9	0.00854	0.01059	0.04445	975.8	3104.0	791.8	6857.2	0.0273	0.1530
10	81.4	73.2	55.8	55.6	39.3	41.1	11201.1	0.00893	0.01094	0.05458	844.8	2687.2	685.5	5936.6	0.0289	0.1586
11	81.3	73.3	54.9	54.7	40.0	41.5	9633.6	0.00939	0.01118	0.06897	693.8	2206.8	563.0	4875.2	0.0313	0.1666
12	81.2	73.3	54.1	53.8	40.3	41.6	8550.6	0.00977	0.01147	0.08405	592.9	1885.9	481.1	4166.4	0.0333	0.1733
13	81.1	73.3	52.9	52.7	40.3	41.5	7438.8	0.01027	0.01177	0.10209	494.4	1572.5	401.1	3473.9	0.0358	0.1814
14	80.6	76.1	52.0	51.8	40.1	41.1	6383.6	0.01019	0.01251	0.19568	350.9	1116.3	284.8	2466.0	0.0411	0.1976
15	81.2	77.7	50.1	49.9	39.4	40.2	5202.8	0.01095	0.01348	0.21816	249.1	792.3	202.1	1750.3	0.0471	0.2153
16	80.6	75.6	51.1	50.9	39.2	40.2	6434.1	0.01039	0.01272	0.16760	351.5	1118.0	285.2	2469.8	0.0411	0.1975

TABLE IV (Continued)

RUN SERIES 41002 DATE 7/ 7/77 BAROMETRIC PRESSURE 29.03 AMBIENT TEMPERATURE 93.0

RUN	TDB1	TWB1	TDB2	TWB2	TWA1	TWA2	QDOT	XJ	XJI	F	RE	RED	RES	REB	JP	FP
1	82.1	74.2	52.1	51.4	36.9	38.4	8124.4	0.00939	0.01128	0.10390	487.7	1551.2	395.7	3426.9	0.0360	0.1820
2	81.7	74.1	54.6	54.4	37.8	39.5	8930.2	0.00846	0.00990	0.08743	601.2	1912.2	487.8	4224.4	0.0331	0.1727
3	82.1	74.5	56.7	55.4	39.3	41.1	10088.0	0.00806	0.01042	0.07176	690.1	2195.0	559.9	4849.1	0.0313	0.1668
4	82.2	74.9	58.2	57.5	40.4	42.5	11492.2	0.00797	0.01041	0.05305	839.4	2670.1	681.2	5898.8	0.0290	0.1589
5	82.6	74.9	57.5	57.1	37.5	39.5	13520.1	0.00775	0.01025	0.04489	971.8	3091.2	788.6	6829.1	0.0273	0.1532
6	83.0	75.1	58.9	58.7	38.9	41.2	14233.1	0.00777	0.01040	0.04006	1086.5	3456.1	881.7	7635.3	0.0261	0.1489
7	83.0	75.5	60.2	59.9	40.0	42.3	14827.9	0.00753	0.01044	0.03718	1175.8	3740.0	954.1	8262.4	0.0253	0.1460
8	82.2	74.2	58.1	57.8	38.1	40.3	13855.8	0.00774	0.01012	0.03965	1080.0	3435.3	876.3	7589.1	0.0262	0.1492
9	82.2	74.3	57.9	57.6	39.4	41.2	12545.8	0.00798	0.01020	0.04374	962.9	3062.8	781.3	6766.2	0.0274	0.1535
10	81.8	73.9	57.1	57.0	40.3	42.2	11009.9	0.00852	0.01041	0.05429	841.2	2675.7	682.6	5911.1	0.0290	0.1588
11	81.7	74.0	56.2	55.8	41.0	42.5	9604.3	0.00900	0.01092	0.06787	690.8	2197.4	560.6	4854.5	0.0313	0.1668
12	81.7	74.2	55.4	55.3	41.2	42.8	8783.8	0.00942	0.01119	0.08159	610.1	1940.7	495.1	4287.3	0.0329	0.1721
13	81.9	75.5	54.6	54.4	41.4	42.8	8075.4	0.00978	0.01207	0.10053	497.8	1583.4	403.9	3498.1	0.0357	0.1810
14	81.9	76.9	53.4	53.1	41.5	42.7	6383.5	0.01021	0.01252	0.15527	349.2	1110.7	283.3	2453.7	0.0412	0.1978
15	82.2	78.5	51.9	51.7	41.2	42.0	5387.4	0.01090	0.01339	0.20940	259.8	826.4	210.8	1825.6	0.0463	0.2130
16	81.9	76.9	53.0	52.9	41.2	42.1	6440.9	0.01027	0.01254	0.16029	349.4	1111.4	283.5	2455.2	0.0411	0.1978

TABLE V

DROP TEST DATA - 14 FINS PER INCH

TUBE O. D. =0.3920 (IN) FIN PITCH = 14 (FINS/IN) HYDRAULIC DIA. =0.00731 (FT) FIN THICKNESS =0.0060 (IN)

FREE FLOW/FRONTAL AREA =0.557 TOTAL AREA/VOLUME = 304.8 (SQ. FT/CU. FT) FIN AREA/TOTAL AREA = 0.949 TUBES IN FACE = 5

TRANS. TUBE SPACING =1.000 (IN) LONG. TUBE SPACING =0.866 (IN) NUMBER OF ROWS = 4 FINNED LENGTH = 12 (IN)

RUN SERIES 50002 DATE 6/15/77 BAROMETRIC PRESSURE 29.12 AMBIENT TEMPERATURE 95.0

RUN	TDB1	TWB1	TDB2	TWB2	TWA1	TWA2	QDOT	XJ	XJI	F	RE	RED	RES	REB	JP	FP
1	78.8	71.4	49.0	47.3	38.2	38.4	8340.9	0.00809	0.01086	0.08353	373.1	1667.8	303.9	3684.5	0.0334	0.1594
2	78.9	70.8	49.7	48.5	38.3	39.9	9227.5	0.00817	0.01067	0.06633	444.3	1986.2	361.9	4387.8	0.0311	0.1526
3	79.4	70.5	50.9	49.7	38.8	40.4	10038.6	0.00802	0.01034	0.06329	514.0	2297.4	418.6	5075.3	0.0294	0.1471
4	79.7	70.8	53.5	52.1	39.4	41.7	11171.0	0.00724	0.01053	0.05262	620.1	2771.5	505.0	6122.8	0.0272	0.1404
5	80.7	72.5	55.2	54.7	40.8	43.5	12845.6	0.00731	0.01309	0.04870	710.4	3175.1	578.6	7014.5	0.0258	0.1357
7	78.9	76.0	48.2	47.9	39.3	40.5	7167.0	0.00892	0.01190	0.10790	258.3	1154.4	210.3	2550.3	0.0387	0.1747
8	79.4	72.7	49.3	48.9	39.4	41.1	8594.0	0.00893	0.01146	0.07655	376.9	1684.7	307.0	3721.8	0.0332	0.1590
9	79.4	72.2	50.1	49.8	39.5	41.1	9767.8	0.00866	0.01136	0.06308	454.7	2032.2	370.3	4489.5	0.0308	0.1517
10	79.8	72.3	51.6	51.0	40.1	41.9	10602.1	0.00826	0.01119	0.05475	513.2	2294.0	418.0	5067.8	0.0294	0.1472
11	79.9	71.9	54.4	54.3	39.2	41.6	15154.0	0.00723	0.01135	0.03844	861.2	3849.5	701.4	8504.3	0.0239	0.1293
12	80.3	72.4	54.5	54.4	40.2	42.4	14454.8	0.00745	0.01149	0.03995	799.9	3575.2	651.5	7898.4	0.0246	0.1317
13	80.4	72.7	55.0	54.6	41.1	43.5	13452.9	0.00738	0.01103	0.04455	732.2	3272.7	596.3	7229.9	0.0255	0.1347
14	79.6	72.0	52.5	51.9	39.3	41.2	12204.9	0.00763	0.01073	0.04464	621.4	2777.5	506.1	6136.0	0.0272	0.1403
15	79.9	71.8	54.3	54.2	38.8	41.1	15254.7	0.00711	0.01109	0.03681	865.0	3866.5	704.5	8541.8	0.0238	0.1292
16	80.1	72.3	55.3	55.1	39.8	42.2	13871.0	0.00676	0.00948	0.03997	794.9	3553.3	647.5	7849.9	0.0247	0.1319

TABLE V (Continued)

RUN SERIES 50003 DATE 6/17/77 BAROMETRIC PRESSURE 28.98 AMBIENT TEMPERATURE 90.0

RUN	TDB1	TWB1	TDB2	TWB2	TWA1	TWA2	QDOT	XJ	XJI	F	RE	RED	RES	REB	JP	FP
1	80.6	72.8	55.0	54.8	39.4	42.0	15825.7	0.00717	0.01279	0.04690	867.0	3875.3	706.1	8561.2	0.0238	0.1291
2	79.6	72.3	54.0	53.7	39.5	42.0	14697.2	0.00740	0.01234	0.03746	791.1	3536.0	644.3	7811.6	0.0247	0.1321
3	79.5	72.2	52.6	52.4	38.6	41.0	13951.3	0.00767	0.01172	0.03754	714.2	3192.3	581.7	7052.3	0.0257	0.1355
4	79.6	72.2	52.3	52.1	39.7	41.9	12232.1	0.00801	0.01134	0.05296	619.5	2769.2	504.6	6117.6	0.0272	0.1404
5	79.7	72.9	51.6	51.2	40.3	42.1	10781.1	0.00836	0.01167	0.07020	505.5	2259.6	411.7	4992.0	0.0296	0.1477
6	79.5	72.8	50.8	50.6	40.9	42.5	9587.4	0.00895	0.01194	0.09242	442.1	1976.2	360.1	4365.7	0.0312	0.1528
7	79.2	72.5	47.0	46.7	37.3	38.7	8777.5	0.00928	0.01200	0.12617	364.2	1628.0	296.6	3596.6	0.0337	0.1603
8	79.3	72.0	49.0	48.8	37.6	39.3	9639.6	0.00840	0.01056	0.08641	440.0	1966.5	358.3	4344.4	0.0312	0.1529
9	79.3	71.6	49.5	49.3	38.0	40.1	10800.5	0.00861	0.01144	0.05992	510.6	2282.4	415.9	5042.1	0.0294	0.1473
10	79.4	71.3	51.2	51.0	38.6	40.7	12142.2	0.00818	0.01123	0.06194	620.9	2775.1	505.7	6130.8	0.0272	0.1403
11	79.5	71.5	52.8	52.7	39.5	41.9	13210.7	0.00789	0.01142	0.05236	716.0	3200.3	583.1	7070.0	0.0257	0.1354
12	79.7	72.1	53.4	53.2	38.4	41.0	14877.3	0.00736	0.01167	0.04545	791.9	3539.5	645.0	7819.4	0.0247	0.1320
13	80.0	72.2	53.7	53.6	37.8	40.5	15979.8	0.00721	0.01239	0.04212	862.1	3853.3	702.1	8512.7	0.0239	0.1293

TABLE VI

DRY TEST DATA - 14 FINS PER INCH

TUBE O. D. =0.3920 (IN)   FIN PITCH = 14 (FINS/IN)   HYDRAULIC DIA. =0.00731 (FT)   FIN THICKNESS =0.0060 (IN)

FREE FLOW/FRONTAL AREA =0.557   TOTAL AREA/VOLUME = 304.8 (SQ. FT/CU. FT)   FIN AREA/TOTAL AREA = 0.949   TUBES IN FACE = 5

TRANS. TUBE SPACING =1.000 (IN)   LONG. TUBE SPACING =0.865 (IN)   NUMBER OF ROWS = 4   FINNED LENGTH = 12 (IN)

RUN SERIES 50100   CATE 6/10/77   BARCMETRIC PRESSURE 29.50   AMBIENT TEMPERATURE 88.0

RUN	TDB1	TWB1	TDB2	TWB2	TWA1	TWA2	CDOT	XJ	XJI	F	RE	RED	RES	REB	JP	FP
1	79.3	68.8	131.2	85.2	145.6	143.3	8520.7	0.01008	0.00000	0.04067	454.3	2030.5	370.0	4485.7	0.0308	0.1517
2	81.1	69.0	129.4	84.5	145.6	143.3	9664.6	0.00938	0.00000	0.03765	553.9	2475.7	451.1	5469.3	0.0285	0.1444
3	82.2	69.4	127.6	84.0	145.6	143.1	10414.9	0.00869	0.00000	0.03370	635.6	2840.9	517.7	6276.0	0.0270	0.1395
4	83.4	69.9	126.5	83.8	145.6	143.1	11083.7	0.00829	0.00000	0.03126	711.5	3180.4	579.5	7026.2	0.0258	0.1356
5	84.0	70.6	125.4	83.4	145.3	142.9	11549.9	0.00795	0.00000	0.03043	773.8	3458.9	630.3	7641.3	0.0249	0.1326
6	84.9	70.8	124.7	83.4	145.1	142.6	11595.9	0.00768	0.00000	0.02928	807.4	3609.0	657.6	7973.0	0.0245	0.1314
7	84.7	70.3	125.1	83.8	144.9	142.4	11306.6	0.00789	0.00000	0.03010	773.7	3458.5	630.2	7640.4	0.0249	0.1328
8	84.0	70.3	126.0	83.8	144.9	142.6	10748.5	0.00817	0.00000	0.03148	708.2	3165.4	576.8	6992.9	0.0258	0.1358
9	83.6	70.3	127.4	84.0	144.7	142.6	10031.0	0.00867	0.00000	0.03394	633.4	2831.1	515.9	6254.3	0.0270	0.1396
10	82.2	69.4	128.9	84.3	144.7	142.6	9314.3	0.00933	0.00000	0.03647	551.5	2465.2	449.2	5446.2	0.0285	0.1445
11	80.7	68.5	131.0	84.7	144.7	142.9	8206.0	0.01014	0.00000	0.04038	451.2	2016.7	367.5	4455.3	0.0309	0.1520
12	80.0	68.1	132.5	85.6	144.9	143.1	7432.0	0.01072	0.00000	0.04318	390.8	1746.6	318.3	3858.7	0.0328	0.1575
13	79.1	67.6	134.3	86.3	144.9	143.8	6383.7	0.01145	0.00000	0.04943	319.0	1426.0	259.8	3150.2	0.0355	0.1657
14	78.0	65.8	136.6	87.0	145.3	144.7	4788.8	0.01213	0.00000	0.05927	225.4	1007.6	183.6	2225.9	0.0408	0.1808
15	78.0	66.8	136.1	87.2	145.8	145.1	3365.6	0.01100	0.00000	0.06949	159.6	713.6	130.0	1576.4	0.0469	0.1971
16	78.2	66.8	137.5	87.2	146.7	145.6	4840.2	0.01194	0.00000	0.05932	225.1	1006.0	183.3	2222.5	0.0408	0.1808
17	79.3	67.0	136.4	86.7	147.1	145.6	6577.1	0.01154	0.00000	0.04954	318.0	1421.3	259.0	3140.0	0.0356	0.1659
18	79.8	67.2	135.2	86.5	147.6	146.0	7761.4	0.01097	0.00000	0.04303	386.0	1725.5	314.4	3811.9	0.0329	0.1580

TABLE VII

FILM TEST DATA - 14 FINS PER INCH

TUBE O. D. =0.3920 (IN) FIN PITCH = 14 (FINS/IN) HYDRAULIC DIA. =0.00731 (FT) FIN THICKNESS =0.0060 (IN)

FREE FLOW/FRONTAL AREA =0.557 TOTAL AREA/VOLUME = 304.8 (SQ. FT/CU. FT) FIN AREA/TOTAL AREA = 0.949 TUBES IN FACE = 5

TRANS. TUBE SPACING =1.000 (IN) LONG. TUBE SPACING =0.866 (IN) NUMBER OF ROWS = 4 FINNED LENGTH = 12 (IN)

RUN SERIES 51000 CATE 6/20/77 BAROMETRIC PRESSURE 29.04 AMBIENT TEMPERATURE 94.0

RUN	TDB1	TWB1	TDB2	TWB2	TWA1	TWA2	QDOT	XJ	XJI	F	RE	RED	RES	REB	JP	FP
1	80.1	77.2	48.5	47.1	37.0	38.4	7719.6	0.00782	0.01123	0.13013	257.5	1151.1	209.7	2542.9	0.0387	0.1749
2	79.5	73.7	49.6	48.4	38.5	40.1	8969.2	0.00819	0.01138	0.07016	367.4	1642.1	299.2	3627.8	0.0336	0.1600
3	79.9	73.2	51.0	50.0	39.2	40.9	10026.6	0.00795	0.01105	0.05375	442.4	1977.6	360.4	4368.9	0.0312	0.1527
4	80.1	73.1	52.3	51.4	39.8	41.5	10856.9	0.00766	0.01070	0.05178	508.5	2272.8	414.1	5021.0	0.0295	0.1475
5	80.4	72.7	53.8	53.2	40.6	42.6	12106.0	0.00757	0.01061	0.04517	621.0	2775.9	505.8	6132.5	0.0272	0.1403
6	80.9	73.0	53.6	53.4	38.8	41.1	13990.4	0.00737	0.01066	0.03937	711.0	3178.1	579.1	7021.0	0.0258	0.1356
7	81.3	73.2	54.7	54.7	39.6	42.0	14895.5	0.00732	0.01121	0.04024	788.9	3526.4	642.6	7790.5	0.0247	0.1322
8	81.9	73.9	55.5	55.4	38.9	41.5	16561.6	0.00693	0.01229	0.03557	862.7	3856.2	702.7	8519.1	0.0239	0.1292
9	82.2	73.8	57.2	57.1	39.6	42.2	15870.3	0.00642	0.00937	0.03410	900.2	4023.9	733.2	8889.5	0.0235	0.1279
10	81.9	73.8	55.2	55.0	38.1	40.6	16731.0	0.00684	0.01152	0.03556	863.3	3858.6	703.1	8524.3	0.0239	0.1292
11	81.8	73.6	54.9	54.7	39.0	41.5	15412.0	0.00710	0.01106	0.03653	792.3	3541.5	645.3	7823.9	0.0247	0.1320
12	80.2	72.2	52.7	52.5	37.8	40.2	13813.7	0.00739	0.01043	0.04994	712.5	3184.6	580.3	7035.4	0.0258	0.1356
13	80.0	72.1	52.1	51.8	38.9	41.0	12307.5	0.00787	0.01078	0.04148	620.3	2772.8	505.3	6125.7	0.0272	0.1404
14	79.8	72.2	51.0	50.8	39.5	41.2	10560.1	0.00836	0.01090	0.04815	510.1	2280.0	415.5	5037.0	0.0294	0.1474
15	79.7	72.3	50.3	50.1	40.1	41.6	9348.7	0.00885	0.01129	0.06677	435.6	1947.1	354.8	4301.6	0.0314	0.1533
16	79.7	72.5	48.6	48.3	40.4	41.9	8359.7	0.01012	0.01311	0.06705	363.5	1624.8	296.1	3589.6	0.0337	0.1604

TABLE VII (Continued)

RUN SERIES 51001   DATE 6/21/77   BAROMETRIC PRESSURE 29.13   AMBIENT TEMPERATURE 92.0

RUN	TDB1	TWB1	TDB2	TWB2	TWA1	TWA2	QDOT	XJ	XJI	F	RE	RED	RES	REB	JP	FP
1	78.7	70.9	48.1	47.3	36.0	37.7	9753.6	0.00813	0.01043	0.05189	449.3	2008.3	366.0	4436.8	0.0310	0.1521
2	80.0	71.4	49.4	48.5	36.6	38.5	11019.2	0.00811	0.01076	0.04636	511.9	2288.2	417.0	5055.1	0.0294	0.1473
3	80.5	71.8	50.9	51.3	37.8	40.0	12321.1	0.00828	0.01052	0.04048	622.5	2782.6	507.0	6147.4	0.0272	0.1402
4	81.3	72.3	53.2	52.9	39.0	41.1	13769.1	0.00776	0.01089	0.03889	716.8	3203.9	583.8	7077.9	0.0257	0.1354
5	80.4	72.1	54.2	54.0	39.4	41.9	14577.7	0.00742	0.01120	0.03610	804.3	3595.2	655.1	7942.5	0.0245	0.1315
6	80.4	72.0	53.5	53.3	36.9	39.5	16069.3	0.00710	0.01120	0.03499	864.3	3863.4	704.0	8534.9	0.0238	0.1292
7	80.6	72.3	54.5	54.4	38.0	40.7	16424.3	0.00704	0.01233	0.03448	909.8	4066.7	741.0	8984.2	0.0234	0.1275
8	80.6	72.4	54.7	54.5	39.0	41.5	15591.4	0.00716	0.01163	0.03470	865.4	3868.3	704.9	8545.8	0.0238	0.1291
9	80.2	72.1	53.1	52.9	37.4	39.7	15076.5	0.00720	0.01080	0.03661	794.1	3549.4	646.8	7841.3	0.0247	0.1319
10	80.1	72.1	52.7	52.6	38.2	40.3	13751.8	0.00745	0.01061	0.03795	713.7	3190.0	581.3	7047.3	0.0257	0.1355
11	79.8	71.9	51.9	51.8	38.8	41.0	12231.5	0.00786	0.01071	0.04028	621.6	2778.3	506.2	6137.7	0.0272	0.1403
12	79.4	71.8	50.8	50.5	39.3	41.1	10431.9	0.00833	0.01082	0.04558	510.9	2283.5	416.1	5044.7	0.0294	0.1473
13	79.3	71.6	49.9	49.6	39.4	41.0	9268.6	0.00869	0.01089	0.05020	444.1	1985.1	361.7	4385.4	0.0311	0.1526
14	79.2	72.0	48.5	48.2	39.3	40.7	8179.9	0.00933	0.01164	0.06358	364.3	1628.2	296.7	3597.0	0.0337	0.1603

VITA

Kevin Francis O'Brien

Candidate for the Degree of

Master of Science

Thesis: CORRELATION OF HEAT, MASS, AND MOMENTUM COEFFICIENTS IN PLATE-FIN-TUBE HEAT EXCHANGERS

Major Field: Mechanical Engineering

Biographical:

Personal Data: Born in Tulsa, Oklahoma, August 14, 1954, the son of Mr. and Mrs. Paul H. O'Brien.

Education: Graduated from Thomas Edison High School, Tulsa, Oklahoma, in May, 1972; received Bachelor of Science degree in Mechanical Engineering from Oklahoma State University in 1976; completed requirements for the Master of Science degree at Oklahoma State University, Stillwater, Oklahoma, in December, 1977.

Professional Experience: Summer employment, U.S. Army Corps of Engineers, 1974; summer employment, International Business Machines, 1975; graduate research assistant, School of Mechanical and Aerospace Engineering, Oklahoma State University, 1976-77.

Professional Associations: American Society of Mechanical Engineers; American Society of Heating, Refrigerating, and Air Conditioning Engineers.