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Friction Data for Tube Bundles of Different Size
and Tube Bundle to Shell Wall Spacing

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SUMMARY

An analytical method of correlating fluid friction data for flow parallel to square array cylindrical tubes in rectangular cross-section flow channels has been devised. In this technique, the experimental values for friction factors and Reynolds moduli are multiplied by correlation terms which yield new friction factors and Reynolds moduli that pertain to the core region of the tube bundle excluding the boundary effects; hence, model configurations with different numbers of tubes and different tube bundle to shell wall spacing can be correlated.

Experimental fluid friction data have been obtained on a 25-tube tube bundle by the ARE Division (Ref. 1) and on a 100-tube tube bundle by the REE Division (Ref. 2) with some disagreement of results and it was thought that by this technique this disagreement could be explained. However, it is shown that for either of these two configurations application of this technique changes the original data by less than five percent and does not bring these data into better agreement.

INTRODUCTION

Fluid friction data for the fuel side of model ART fuel-to-Nak heat exchangers have been obtained using a 25-tube heat exchanger by the ARE Division (Ref. 1) and using a 100-tube heat exchanger by the REE Division (Ref. 2). The results obtained on these two heat exchangers are in some disagreement and it was thought that this disagreement was due to the different numbers of tubes and tube bundle to shell wall spacing used. An analytical investigation was undertaken in order that these variations in configuration could be considered in the correlation of experimental data.

By making use of the continuity equation and the hydraulic diameter correlation concept for flow channels of different configurations, two correlation factors are evolved which yield new friction factors and Reynolds moduli that pertain to the core region of the tube bundle and exclude the effects caused by flow adjacent to the shell wall of the heat exchanger.

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NOMENCLATURE

- a = cross sectional flow area of channel, ft.^2
 B = constant in $f = B N_{\text{Re}}^{-C}$
 C = constant in $f = B N_{\text{Re}}^{-C}$
 d = hydraulic diameter ($4a/p$), ft.^2
 D = tube diameter, ft.
 f = friction factor defined by $f = \frac{\Delta P}{L/d} \frac{\rho u^2}{2g_c}$
 g_c = gravitational conversion factor = $4.170 \times 10^8 \text{ lb}_m\text{-ft}/\text{lb}_f\text{-hr}^2$
 L = channel length, ft.
 n = number of channels
 N_x = number of tubes in a horizontal row
 N_y = number of tubes in a vertical row
 N_{Re} = Reynolds modulus defined by $N_{\text{Re}} = \frac{ud}{\nu}$
 s = tube spacing, wall to wall, ft. (assumed to be the same in both x and y directions)
 s' = tube to shell wall spacing, ft. (assumed to be the same in both x and y directions)
 S_1 = s/D
 S_2 = s'/D
 u = mean fluid velocity, ft./hr.
 p = wetted perimeter, ft.
 ΔP = pressure drop through channels, lb_f/ft^2
 ν = kinematic viscosity, ft^2/hr
 ρ = fluid density, lb_m/ft^3
 q = volumetric flow rate, ft^3/hr

Subscripts

- 1: central channels
 2: side channels
 3: corner channels
 e: total or experimental

ANALYSIS OF SYSTEM

In obtaining experimental fluid friction data on this type of configuration, it is customary to use a mean velocity determined by dividing the total volumetric flow by the total cross-sectional flow area and to use the hydraulic diameter which is based on the total wetted perimeter. However, in the heat exchanger shell, there are side and corner flow channels which will contribute to the data thus obtained. Since the effect of these peripheral channels will be different for different types of model heat exchangers, this effect should be considered in the general correlation of data.

The fluid actually flows through three distinct types of channels as shown in Figure 1: (1) those in the core of the tube bundle, (2) those adjacent to the shell wall, and (3) those at the corners of the tube bundle. If the hydraulic diameters of these channels are not the same, for a given pressure drop, each will have a different mean fluid velocity and hence a different Reynolds modulus and friction factor. This analysis is based upon the assumption that fully developed turbulent flow is established, hence the equation $f = B N_{Re}^{-C}$ applies to all three types of channels with the same values for the constants B and C. It is then possible to interpret the experimental data in terms of the fluid friction characteristics for a single central flow channel or, the so-called infinite array.

The following equations for the flow cross-sectional area, wetted perimeter and hydraulic diameter apply to the three types of flow channels.

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Fig. 1. Cross Section of Tube Bundle Showing the Three Types of Flow Channels.

$$a_1 = D^2 \left[(S_1 + 1)^2 - \frac{\pi}{4} \right] \text{-----} \quad (1)$$

$$p_1 = \pi D \text{-----} \quad (2)$$

$$d_1 = D \left[\frac{4(S_1 + 1)^2 - \pi}{\pi} \right] \text{-----} \quad (3)$$

$$a_2 = D^2 \left[(S_1 + 1) \left(S_2 + \frac{1}{2} \right) - \frac{\pi}{8} \right] \text{-----} \quad (4)$$

$$p_2 = D(S_1 + 1 + \frac{\pi}{2}) \text{-----} \quad (5)$$

$$d_2 = D \left[\frac{4(S_1 + 1)(S_2 + \frac{1}{2}) - \frac{\pi}{2}}{S_1 + 1 + \frac{\pi}{2}} \right] \text{-----} \quad (6)$$

$$a_3 = D^2(S_2 + \frac{1}{2})^2 - \frac{\pi}{16} \text{-----} \quad (7)$$

$$p_3 = D(2 S_2 + 1 + \frac{\pi}{4}) \text{-----} \quad (8)$$

$$d_3 = D \left[\frac{4(S_2 + \frac{1}{2})^2 - \frac{\pi}{4}}{2 S_2 + 1 + \frac{\pi}{4}} \right] \text{-----} \quad (9)$$

For the experimental data, which are taken for all channels as a group,
the following equations apply:

$$a_e = D^2 \left[(N_x + N_x S_1 - S_1 + 2S_2)(N_y + N_y S_1 - S_1 + 2S_2) - \frac{N_x N_y \pi}{4} \right] \text{----} \quad (10)$$

$$p_e = D \left[2(N_x + N_y + N_x S_1 + N_y S_1 - 2S_1 + 4S_2) + N_x N_y \pi \right] \text{-----} \quad (11)$$

$$d_e = D \left[\frac{4(N_x + N_x S_1 - S_1 + 2S_2)(N_y + N_y S_1 - S_1 + 2S_2) - N_x N_y \pi}{2(N_x + N_y + N_x S_1 + N_y S_1 - 2S_1 + 4S_2) + N_x N_y \pi} \right] \text{-----} \quad (12)$$

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By summing the quantities for the individual channels, it can be demonstrated that

$$a_e = n_1 a_1 + n_2 a_2 + n_3 a_3 \quad (13)$$

$$p_e = n_1 p_1 + n_2 p_2 + n_3 p_3 \quad (14)$$

$$d_e = \frac{4(n_1 a_1 + n_2 a_2 + n_3 a_3)}{n_1 p_1 + n_2 p_2 + n_3 p_3} \quad (15)$$

and by definition

$$n_1 = (N_x - 1)(N_y - 1) \quad (16)$$

$$n_2 = 2(N_x + N_y - 2) \quad (17)$$

$$n_3 = 4 \quad (18)$$

The experimental friction factor and Reynolds modulus are obtained from

$$f_e = \frac{\Delta P}{L/d_e} \frac{\rho u_e^2}{2g_c} \quad (19)$$

and

$$N_{Re_e} = \frac{u_e d_e}{\nu} \quad (20)$$

and for a central flow channel

$$f_1 = \frac{\Delta P}{L/d_1} \frac{\rho u_1^2}{2g_c} \quad (21)$$

and

$$N_{Re_1} = \frac{u_1 d_1}{\nu} \quad (22)$$

From these four equations,

$$\frac{f_1}{f_e} = \frac{d_1}{d_e} \left(\frac{u_e}{u_1} \right)^2 \quad (23)$$

and

$$\frac{N_{Re_1}}{N_{Re_e}} = \frac{d_1/d_e}{u_e/u_1} \quad \text{-----} \quad (24)$$

These are the factors by which the experimental friction factors and Reynolds moduli may be multiplied to obtain the friction characteristics of a central flow channel. The hydraulic diameters d_1 and d_e are obtained from equations (5) and (12) and the relationship between u_e and u_1 is obtained by the following method. The experimental mean fluid velocity, u_e , is obtained by dividing the total quantity flow, q_e , by the total cross-sectional flow area, then

$$q_e = n_1 q_1 + n_2 q_2 + n_3 q_3 \quad \text{or}$$

$$u_e a_e = n_1 u_1 a_1 + n_2 u_2 a_2 + n_3 u_3 a_3$$

Therefore

$$\frac{u_e}{u_1} = \frac{n_1 a_1}{a_e} + \frac{n_2 a_2}{a_e} \left(\frac{u_2}{u_1} \right) + \frac{n_3 a_3}{a_e} \left(\frac{u_3}{u_1} \right) \quad \text{-----} \quad (25)$$

$$\text{Since } f \sim N_{Re}^{-C} = \frac{\nu^C}{u^C d^C}$$

$$\Delta P = \frac{L}{d_1} \frac{\nu^C}{u_1^C d_1^C} \frac{\rho u_1^2}{2g_c} = \frac{L}{d_2} \frac{\nu^C}{u_2^C d_2^C} = \frac{L}{d_3} \frac{\nu^C}{u_3^C d_3^C} \frac{\rho u_3^2}{2g_c} \quad \text{or}$$

$$\frac{u_2}{u_1} = \left(\frac{d_2}{d_1} \right)^{\frac{1+C}{2-C}} \quad \text{-----} \quad (26)$$

and

$$\frac{u_3}{u_1} = \left(\frac{d_3}{d_1} \right)^{\frac{1+C}{2-C}} \quad \text{-----} \quad (27)$$

By using these equations, the correlation factors f_1/f_e and N_{Re1}/N_{Ree} can be determined for a given configuration and assumption for C . It is possible to express these correlation factors in terms of N_x , N_y , S_1 and S_2 ; but as these equations are long and complicated it is simpler to work in steps with the individual equations as stated.

The method of calculation for the ARED configuration with $C = 1/4$ is demonstrated in the Appendix.

APPLICATION TO EXPERIMENTAL DATA

These correlation factors have been calculated for the configurations of References 1 and 2 for two cases: (1) assuming $C = 1/4$ and (2) assuming $C = 1/2$, and the results are shown in Table I.

In order to determine the extent to which application of these factors will shift the original fluid friction data for these two configurations, they have been applied to a point ($N_{Re} = 4,000$) on the Blasius equation ($f = 0.316 N_{Re}^{-1/4}$) in Figure 2. Referring to Figure 2, point 0 is assumed to be an original data point. Point 1 shows the extent to which this point would be shifted if it were obtained from the ARED configuration with the assumption that $C = 1/4$. In similar manner, points 2, 3 and 4 are established. The change in position from point 0 to any other point is the data shift for that particular configuration and that assumption for C . It should be noted that Figure 2 does not present actual experimental data but is intended to show the degree of data shift that would be expected by applying these correlation factors to the two configurations under consideration.

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TABLE I

PHYSICAL CHARACTERISTICS AND CORRELATION
FACTORS FOR TWO MODEL HEAT EXCHANGERS

	<u>ARED</u> <u>Ref.(1)</u>	<u>REED</u> <u>Ref.(2)</u>
D, ft.	0.01592	0.01575
s, ft.	0.00225	0.00167
s', ft.	0.00258	0.00442
S ₁	0.1414	0.1058
S ₂	0.1623	0.2804
N _x = N _y	5	10
n ₁	16	81
n ₂	16	36
n ₃	4	4
d ₁ /d _e	1.116	0.929
Assuming C = 1/4:		
u _e /u ₁	0.932	1.062
f ₁ /f _e	0.97	1.05
N _{Re1} /N _{Ree}	1.20	0.88
Assuming C = 1/2		
u _e /u ₁	0.908	1.090
f ₁ /f _e	0.92	1.10
N _{Re1} /N _{Ree}	1.23	0.85

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Fig. 2. Application of Correlation Factors to an Arbitrarily Assumed Data Point.

DISCUSSION

As can be seen from Figure 2, application of these correlation factors will not cause a significant change in fluid friction data obtained from the two configurations under consideration. The assumption that $C = 1/2$ is probably more valid than that of $C = 1/4$ for the Reynolds modulus range used and with this assumption the ARED data will be lowered slightly and the REED data raised slightly which will bring them into closer agreement, but will not cause them to coincide.

One possible explanation for this disagreement of data is that most of these data are taken in the transitional region from laminar to turbulent flow. It is then possible that laminar flow exists in some channels with turbulent flow in others and/or unstable flow randomly changing from laminar to turbulent. Unless turbulent flow exists, the results of this analysis are invalid since for laminar flow, the friction characteristics are dependent upon the shape of the cross-sectional flow area and the use of the hydraulic diameter as a correlating term for different shapes is not correct.

The major disagreement seems to be in the effect of the individual spacers on the total friction factor: the REED data indicate that the effect of spacer density is rather small while those from ARED indicate that the effect of the spacers is quite significant. Assuming that an individual spacer can be considered to be a cylindrical strut in a free stream, an analysis has been conducted which gives "spacer friction factors" of the same order of magnitude as those predicted by the REED data. These are also substantiated by a rough contraction-expansion analysis.

It should be mentioned that from both configurations, the "spacer friction factor" is determined by subtracting sets of data of the same magnitude and dividing by the total number of spacer sets so that at best the accuracy of these data is rather poor.

CONCLUSIONS

This analysis should be useful in correlating data from different configurations under the following conditions:

1. For a given configuration, the hydraulic diameters of the three different types of channels should not have widely differing values.
2. Fully developed turbulent flow exists at all times.
3. The value for C should be judiciously chosen, i.e., it should correspond to the slope of the f vs. N_{Re} curve for that configuration when plotted logarithmically.

It is recommended that in the design of model configurations, the shell be designed so that the hydraulic diameter of the side flow channels be the same as that of the central flow channels. Under this condition the effect of these correlation factors will be a minimum being only controlled by the small effect of the four corner flow channels.

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APPENDIX

CALCULATIONS FOR THE ARED
HEAT EXCHANGER WITH $C = 1/4$

From Reference (1), $D = 0.191 \text{ in} = 0.01592 \text{ ft}$ and $s = 0.027 \text{ in} = 0.00225 \text{ ft}$
hence,

$$S_1 = \frac{0.027}{0.191} = 0.1414$$

The inside shell dimension is 1.125 in, hence

$$s' = \frac{1}{2} [1.125 - 5(0.191) - 4(0.027)] = 0.031 \text{ in} = 0.00258 \text{ ft}$$

$$S_2 = \frac{0.031}{0.191} = 0.1623$$

Since $N_x = N_y$, $n_1 = (N_x - 1)(N_y - 1) = 16$, $n_2 = 2(N_x + N_y - 2) = 16$, and $n_3 = 4$.

From equations (1) through (12) the following values are obtained:

$$a_1 = 0.01887 \text{ in}^2, p_1 = 0.600 \text{ in}, d_1 = 0.1258 \text{ in},$$

$$a_2 = 0.01325 \text{ in}^2, p_2 = 0.518 \text{ in}, d_2 = 0.1023 \text{ in},$$

$$a_3 = 0.00884 \text{ in}^2, p_3 = 0.403 \text{ in}, d_3 = 0.0877 \text{ in},$$

$$a_e = 0.5493 \text{ in}^2, p_e = 19.50 \text{ in}, d_e = 0.1127 \text{ in}.$$

$$\frac{d_1}{d_e} = \frac{0.1258}{0.1127} = 1.116$$

From equations (26) and (27) using $C = 1/4$,

$$\frac{u_2}{u_1} = 0.863 \text{ and } \frac{u_3}{u_1} = 0.773, \text{ hence, from equation (25), } u_e/u_1 = 0.932.$$

From these values and equations (23) and (24), $f_1/f_e = 0.97$ and

$$N_{Re1}/N_{Ree} = 1.20.$$