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SHELL-SIDE PRESSURE DROP AND HEAT TRANSFER WITH TURBULENT FLOW IN SEGMENTALLY BAFFLED SHELL-AND-TUBE HEAT EXCHANGERS

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Abstract—Empirical correlations of shell-side data from baffled exchangers are discussed. Methods for estimating the magnitude of the non-effective fluid streams in the shell are considered so that the effective flow through the tube bundle may be determined: this is used in conjunction with correlated data from experimental tube banks in cross-flow.

NOMENCLATURE

A, identification symbol for leakage stream between tubes and baffles;
A_B, heat-transfer surface area in cross-flow zone, ft^2;
A_T, total shell-side heat-transfer surface area, ft^2;
A_W, heat-transfer surface area in window zone, ft^2;
B, identification symbol for cross-flow stream through tube bank;
B_1, identification symbol for the part of the stream of fluid passing through baffle windows which contributes to stream B;
b, constant to be determined experimentally, dimensionless;
C, identification symbol for by-pass stream around tube bank;
C', experimentally determined constant, dimensionless;
C'', experimentally determined constant, dimensionless;
C_p, specific heat of shell-side fluid, Btu/lb degF;
C_1, proportionality factor, h^2/lb;
C_{10}, ratio of friction factors in ideal tube bank at Reynolds numbers Re_B and Re_C, dimensionless;
D, identification symbol for leakage stream between shell and edge of one baffle;
D_c, distances between planes through centres of area of the baffle windows parallel to the baffle edges, ft;
D_e, hydraulic mean diameter for cross-flow = mean value of 4 × flow area/wetted perimeter, ft;
D_{eK}, hydraulic mean diameter for parallel flow, ft;
D_1, shell inside diameter, ft;
D_3, tube bundle diameter, ft;
d, outside diameter of tubes, ft;
E, identification symbol for leakage stream between baffle and shell;
F, Tinker's [18] end space factor =

$$\frac{L_2 + (L_1 - L_2)\left(\frac{2L_3}{L_1 - L_2}\right)^{0\cdot6}}{L_1},$$

dimensionless;
F_{BP}, fraction of total minimum cross-flow area that is in the by-pass channel, dimensionless;
f, friction factor, defined in equation (9), dimensionless;
$f_{Re, B}$ friction factor in tube bundle at Re_B;

2T—H.M.

$f_{Re,\,C}$, friction factor in tube bundle at Re_C;
G, distant mass velocity, lb/h ft²;
G_{av}, $\frac{1}{3}\left(\frac{Q}{S_W} + \frac{Q}{S_M} + \frac{Q}{S_u}\right)$, lb/h ft²;
G_{max}, mean effective mass velocity at minimum free area for cross-flow through rows of tubes, lb/h ft²;
$G_{r,\,max}$, mass velocity at minimum free area for cross-flow through rth row of tubes, lb/h ft²;
G_K, mass velocity calculated for S_K, lb/h ft²;
G_M, mass velocity through S_M, lb/h ft²;
G_W, mass velocity through S_W, lb/h ft;
G_Z, $\sqrt{(G_M G_W)}$ = the geometric mean of the cross-flow and window mass velocities, lb/h ft²;
g_c, dimensional factor, ft/h²;
H, proportionality factor = $\left(1 - \frac{h_L}{h_{NL}}\right)$, dimensionless;
ΔH, head loss in cross-flow zone, ft;
h, mean shell-side heat-transfer coefficient in heat exchanger, Btu/h ft² degF;
h_B, mean shell-side heat-transfer coefficient in cross-flow zone, Btu/h ft² degF;
h_{BP}, mean shell-side heat-transfer coefficient in tube bank with by-pass, Btu/h ft² degF;
h_I, mean shell-side heat-transfer coefficient in ideal tube bank without by-pass, Btu/h ft² degF;
h_L, mean shell-side heat-transfer coefficient in tube bank with leakage, Btu/h ft² degF;
h_m, mean shell-side heat-transfer coefficient around single isolated tube in cross-flow, Btu/h ft² degF;
h_{NL}, mean shell-side heat-transfer coefficient in tube bank without leakage, Btu/h ft² degF;
h_W, mean shell-side heat-transfer coefficient in window zone, Btu/h ft² degF;
k, thermal conductivity at bulk-fluid temperature, Btu/h ft degF;
k_f, thermal conductivity at film-fluid temperature, Btu/h ft degF;
L_1, total length between tube sheets, ft;
L_2, length between end baffles, ft;
L_3, length between adjacent baffles, ft;
m, index in Jakob's correlation of friction factors [equation (10)] dimensionless;
N, number of rows of tubes in cross-flow, rows;
N_b, number of baffles in the exchanger, baffles;
N_e, number of rows of tubes between baffle edges, rows;
N_S, number of sealing strips in by-pass channel, strips;
Nu, Nusselt number, dimensionless;
N_W, estimated number of tube rows passed by fluid in window zone, rows;
n, experimentally determined constant, dimensionless;
n_r, number of tubes in rth row of tube bank, rows;
P, tube pitch, ft;
Pr, Prandtl number, dimensionless;
p, pressure, lb/ft²;
Δp_a, pressure drop due to turning of fluid, lb/ft²;
Δp_B, pressure drop in tube bundle between adjacent baffles, lb/ft²;
Δp_{BP}, pressure drop in rectangular tube bank with by-pass, lb/ft²;
Δp_c, pressure drop due to contraction, lb/ft²;
Δp_e, pressure drop due to expansion, lb/ft²;
Δp_f, pressure drop due to friction, lb/ft²;
Δp_I, pressure drop in ideal tube bank without by-pass, lb/ft²;
Δp_L, pressure drop with leakage, lb/ft²;
Δp_{NL}, pressure drop without leakage, lb/ft²;
Δp_u, total pressure drop between adjacent baffles, lb/ft²;

Δp_W, pressure drop in one baffle window, lb/ft^2;
Q, identification symbol for total stream passing through the heat exchanger;
Q, total mass rate of flow of fluid through shell of heat exchanger, lb/h;
Q_A, part of total fluid flow associated with tube hole leakage, lb/h;
Q_B, part of total fluid flow in cross-flow through tube bundle, lb/h;
Q_C, part of total fluid flow by-passing between bundle and shell, lb/h;
Q_E, part of total flow associated with leakage between baffles and shell, lb/h;
Re_B, Reynolds number in 'B' stream, dimensionless;
Re_C, Reynolds number in 'C' stream, dimensionless;
r, $\frac{A_W}{A_T}$, dimensionless;
S_B, average of minimum cross-sectional areas for rows of tubes in cross-flow zone, ft^2;
S_{Bh}, effective minimum free area for flow in cross-flow zone (in computing heat transfer), ft^2;
S_{Bp}, effective minimum free area for flow in cross-flow zone (in computing pressure drop), ft^2;
S_C, area for flow in by-pass space, ft^2;
S_E, leakage area between baffle and shell, ft^2;
S_K, $D_1\left(1-\frac{1}{x_T}\right)L_3$, ft^2;
S_L, total area for flow in leakage path, ft^2;
S_M, minimum free area for flow in cross-flow zone, ft^2;
S_p, maximum free area for flow in space between adjacent rows in cross-flow zone, ft^2;
S_S, net cross-sectional area of shell, ft^2;
S_{SB}, area for flow in leakage path between shell and baffles, ft^2;
S_{TB}, area for flow in leakage path between tubes and baffles, ft^2;
S_W, free area for flow in plane of baffle window, ft^2;
V_{AR} V_{BR} V_{CR} V_{ER}, relative velocities proportional to the velocities of the streams 'A', 'B', 'C', 'E', respectively, ft/h;
V_B, velocity through S_B, ft/h;
V_M, velocity through S_M, ft/h;
V_W, velocity through S_W, ft/h;
V_Z, geometric mean of the cross-flow and window velocities, ft/h;
X, longitudinal pitch of tubes in tube bank, ft;
x, ratio of pressure drop in window to pressure drop in flow across tubes, dimensionless;
x_L, ratio of longitudinal pitch to tube diameter, dimensionless;
x_T, ratio of transverse pitch to tube diameter, dimensionless;
Y, transverse pitch of tubes in tube bank, ft;
α, constant to be determined experimentally, $Btu/h^{0.4}\ ft^{2.6}\ degF$;
β, constant to be determined experimentally, $Btu/h^{0.4}\ ft^{2.6}\ degF$;
μ, viscosity at bulk-fluid temperature, lb/h ft;
μ_f, viscosity at film-fluid temperature, lb/h ft;
μ_w, viscosity at tube wall temperature, lb/h ft;
ρ, density, lb/ft^3;
ϕ, constant, dimensionless.

1. INTRODUCTION

THE ACCURATE rating of a shell-and-tube heat exchanger at the drawing-board stage is very difficult to accomplish on the basis of past experience. It was primarily for this reason that a survey of the field of shell-side design from the thermal and fluid-flow standpoints was made; the review shows that, although there are still many uncertainties, various methods for rating exchangers are available which may be expected to give results of reasonable accuracy.

The shell-side heat-transfer coefficient may be increased, in the absence of a change of phase, by raising the velocity of the shell-side fluid relative to the tubes, and this is most commonly accomplished by placing segmental disc baffles across the shell, normal to the tubes, with the segmental openings (or "windows") on alternate sides of the centre line. The length of the fluid path, in making a number of passes across the tubes, is increased, with a consequent increase in the fluid velocity and heat-transfer coefficient. A further consequence of the baffles is an increased pressure drop which limits the number of baffles that may be used. This review is limited to shell-and-tube exchangers of this type, in which the shell-side fluid is turbulent and undergoes no change of phase.

The correlation of the performance of industrial and experimental heat exchangers has been approached in two ways, which may be termed the integral and the analytic approaches. In the integral method an attempt is made to correlate the heat-transfer coefficient or pressure drop of the shell-side, taken as a whole (hence "integral"), by means of dimensionless groups containing as many as possible of the parameters. The analytic method attempts to distinguish between the various internal flow paths and to evaluate their individual effects.

The first approach is very successful with simple heat-exchanger models, but leads to poorer correlations as the models become more complex. Thus, as shown by McAdams [1], p. 259, the heat transfer to or from air flowing normal to a single isolated tube may be correlated very well if the Nusselt number is plotted against Reynolds number, provided these dimensionless groups are defined as

$$Nu = \frac{h_m d}{k_f}, \quad Re = \frac{Gd}{\mu_f} \tag{1}$$

where h_m is the mean heat-transfer coefficient around the tube, d is the tube diameter, G is the distant mass velocity and k_f, μ_f are the film values of the thermal conductivity and viscosity evaluated at the film temperature (the mean of the surface and bulk fluid temperatures).

The heat transfer to or from air flowing normal to banks of tubes of various geometries may be correlated on the basis of the mean heat-transfer coefficient, using functions containing the appropriate characteristic dimensions, though a distinction has to be made between "staggered" and "in-line" tube arrangements: but the correlations so far obtained for tube banks are notably inferior in precision to the correlations for isolated single tubes, and this lack of precision becomes more marked when further variables, such as departures from rectangularity of the tube bank, baffle spacing, size of baffle window and dimensions of internal-leakage passages, are brought into correlations. It is for this reason efforts have been made to approach the correlation of heat-transfer coefficients and pressure drops analytically. The effects of variations in internal dimensions on the local pattern of flow may thus be used to build up a composite picture of the exchanger operating as a whole.

The flow paths in a segmentally baffled shell are illustrated in Fig. 1. It is seen that, in addition to the cross-flow stream B through the tube bundle from one baffle window to the next, there is a by-pass stream C which evades the tube bundle and passes between the bundle and the shell, making no contribution to heat transfer. There is a further by-pass stream E which leaks through the clearance space between the baffles and the shell, and a leakage path A through the clearance spaces between the tubes and baffles interacts with the main cross-flow stream. Fig. 1 also shows how the flow through the tube bundle changes from parallel flow in the window to almost normal flow and then back to parallel flow in the next window.

2. INTEGRAL CORRELATION OF PRESSURE DROP

Short [2, 3, 4] reported a large number of tests on heat exchangers in which the dimensions, baffle height, baffle spacing, tube diameter, tube pitch and baffle-to-shell clearance, were varied one at a time. He considered the total pressure drop Δp_u between two adjacent baffles as comprising pressure drops due to contractions on entry to the window and to the tube rows, expansions on leaving the window and tube rows, turning of the fluid in the window and around the tubes, and friction, i.e.

$$\Delta p_u = (\Delta p_c + \Delta p_e + \Delta p_a)_W + (\Delta p_c + \Delta p_e + \Delta p_a)_B + \Delta p_f. \tag{2}$$

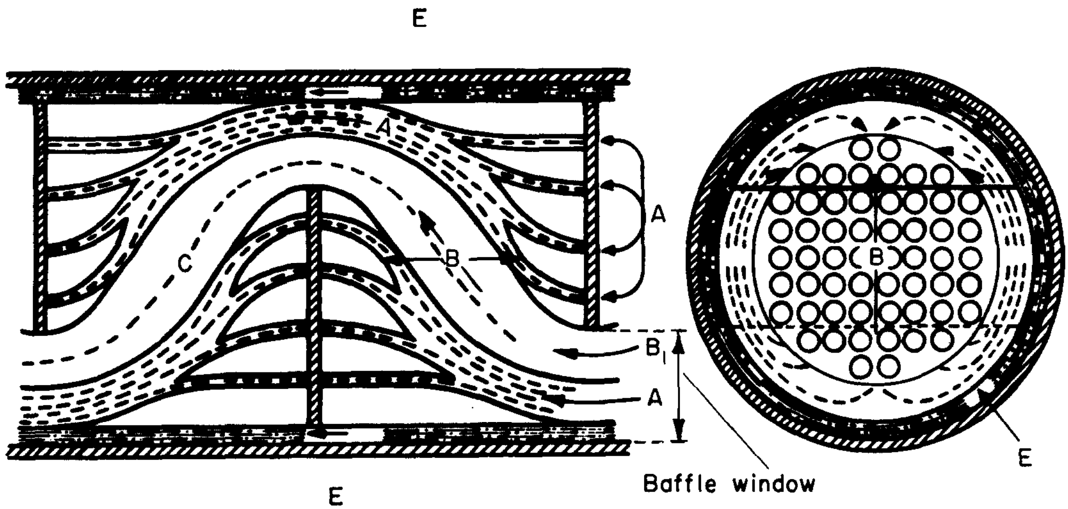

FIG. 1. Diagram of fluid streams through the shell side of a heat exchanger (from Tinker [18]).

where subscripts c, e, a, refer to contraction, expansion, turning, subscripts W and B refer to the window and the tube bundle, and subscript f to friction. Short decided that the contraction loss in the tube rows was negligible and that the turning loss in the tube rows could be included in the Δp_{eB} term: the friction term Δp_f was also omitted. Equation (2) then became:

$$\Delta p_u = (\Delta p_c + \Delta p_e + \Delta p_a)_W + \Delta p_{eB} \qquad (3)$$

and each term on the right of equation (3) was evaluated thus:

$$\Delta p_{cW} = \rho \left(0{\cdot}42 - 0{\cdot}45 \frac{S_W}{S_S}\right)^2 \frac{V_W^2}{2g_c}, \qquad (4)$$

$$\Delta p_{eW} = \rho \left(1 - \frac{S_W}{S_S}\right)^2 \frac{V_W^2}{2g_c}, \qquad (5)$$

$$\Delta p_{aW} = \rho \frac{D_c}{\sqrt{(D_c^2 + 0{\cdot}038)}} \frac{V_W^2}{2g_c} \qquad (6)$$

and

$$\Delta p_{eB} = \rho \left[2{\cdot}5\left(\frac{P-d}{d}\right)\left(1 - \frac{S_M}{S_p}\right)^2 N_e\right] \left(\frac{S_W}{S_M}\right)^2 \frac{V_W^2}{2g_c}. \qquad (7)$$

The total pressure drop for the exchanger thus became:

$$\Delta p = \left\{\left[\frac{D_c}{\sqrt{(D_c^2 + 0{\cdot}038)}} + \left(0{\cdot}42 - 0{\cdot}45 \frac{S_W}{S_S}\right)^2 + \left(1 - \frac{S_W}{S_S}\right)^2\right] N_b + \left[2{\cdot}5\left(\frac{P-d}{d}\right)\left(1 - \frac{S_M}{S_p}\right)^2 \times N_e\left(N_b + 1\right)\left(\frac{S_W}{S_M}\right)^2\right]\right\} \frac{\rho V_W^2}{2g_c}. \qquad (8)$$

It will be seen that the correlation takes no account of by-pass between tube bundle and shell or of internal leakage.

3. COMPONENTS OF PRESSURE DROP

3.1 *Rectangular banks of tubes in cross-flow*

There is a large volume of data in the literature reporting pressure drop in isothermal flow normal to experimental tube banks; these are compared and reviewed by Boucher and Lapple [5]. The tube diameters range from 0·197 to 3·94 in, the transverse pitch ratios [= (transverse pitch)/(tube diameter)] from 1·135 to 5, and the longitudinal pitch ratios from 1·00 to 6·05. The boundaries of these rectangular tube banks vary considerably. The tube lengths range from 1·26 to 84 diameters, the number of tubes per row from one to twenty-five, and the number of rows from two to thirty-two. The channel walls pass either through the centres of the outside tubes or at various distances from them. Reynolds numbers up to 1 500 000 have been investigated. It is not surprising, therefore, that no one has succeeded in correlating the data onto a single line.

The most commonly used data are those given in 1937 by Huge [6] and by Pierson [7], in the range of Reynolds numbers 2000 to 40 000, correlated by Grimison [8]. Huge and Pierson both used ten-row tube banks, so the correlation is strictly applicable only to banks having that number of rows. However, where the number of rows differ from ten, the data of Pierson [7] may be used to apply a correction. The friction factor is defined as:

$$f = \frac{2g_c\ \rho\ \Delta_p}{4N\ G^2_{\max}}, \tag{9}$$

where $G_{\max}$ is the mass velocity at the smallest cross section for flow.

Grimison's correlation is in graphical form, consisting of charts in which parameters are Reynolds number (in which the tube diameter is the characteristic dimension), the pitch ratios and the friction factor. From Pierson's results Jakob [9] has derived the dimensionless equations below.

For tubes in line and $1{\cdot}5 < x_T < 4{\cdot}0$,

$$f = \left[0{\cdot}044 + \frac{0{\cdot}08\ x_L}{(x_T - 1)^m}\right]\left(\frac{G_{\max} d}{\mu_f}\right)^{-0{\cdot}15} \tag{10}$$

where $m = 0{\cdot}43 + (1{\cdot}13/x_L)$; and for staggered tubes, and $1{\cdot}5 < x_T < 4{\cdot}0$,

$$f = \left[0{\cdot}25 + \frac{0{\cdot}1175}{(x_T - 1)^{1{\cdot}08}}\right]\left(\frac{G_{\max} d}{\mu_f}\right)^{-0{\cdot}16}. \tag{11}$$

Equation (11) is not applicable to staggered banks in which the minimum flow area is through the diagonal opening between succeeding rows of tubes.

Boucher and Lapple [5] found that the Grimison correlation represented all the available data with an average deviation of 13 per cent in both in-line and staggered arrangements; the Jakob formulae were shown to be good approximations within the prescribed limits. The correlation in 1945 by Gunter and Shaw [10] was less satisfactory.

The Grimison correlation appears in some textbooks (e.g. Fishenden and Saunders [11]).

Since Boucher and Lapple reviewed the field in 1948, other groups of investigators have published results. Bergelin, Brown and Doberstein [12] in 1952 obtained pressure-drop data for five tube arrangements with oil as the shell-side fluid. The upper limit of the Reynolds number was 6000 and the runs were non-isothermal. Their curves are reproduced by McAdams [1].

Diehl and Unruh [13], in 1958, reported tests on four tube arrangements with air, methane vapour and pentane vapour and Reynolds numbers up to 500 000. Where Diehl and Unruh tested a tube arrangement identical to one used by Bergelin, Brown and Doberstein, there is very precise agreement in the Reynolds number range common to both tests.

Gram, Mackey and Monroe [14] in 1958 presented new data for in-line tube arrangements and correlated them graphically; their results are in reasonable agreement with those of Huge and Pierson.

Bressler [15, 16] has studied tube arrangements in which the lateral displacement of the staggered rows varies from zero to half a transverse pitch length. Where comparable with Grimison's correlation, Bressler's friction factors for 10-row tube banks are up to 20 per cent lower. He suggests that the differences may in some cases be due to the effect of the duct walls, which was not deducted in the Grimison correlation.

In determining the friction factor for non-isothermal flows, allowance must be made for a viscosity gradient in the boundary layer. Sieder and Tate [17] found in 1936 that, for non-isothermal flow in pipes, the pressure drops could be correlated by applying the factor $(\mu/\mu_w)^{0{\cdot}14}$ to the friction factor calculated for isothermal flow at the same mixed mean temperature. Bergelin, Brown and Doberstein [12] found that, for non-isothermal flow of oils normal to the tube banks, the friction factors could be roughly correlated with the isothermal factors of Grimison [8] by application of the same factor, i.e. $(\mu/\mu_w)^{0{\cdot}14}$.

3.2 *Cylindrical banks of tubes in cross-flow*

Where the tube bank is in the form of a bundle contained within a cylindrical shell, the free area for flow normal to the tubes varies from one tube row to the next, and a mean effective value must be adopted in applying the basic pressure-drop data obtained from rectangular tube banks. Most writers appear to adopt the arithmetic mean value. The author considers that the value developed below is more rational.

Let S_{Bp} be the effective flow area for pressure drop; then the effective velocity is proportional to $1/S_{Bp}$ and the mean pressure drop per row to $(1/S_{Bp})^2$, assuming a constant friction factor in the tube bundle. Similarly the pressure drop in the rth row of tubes is proportional to $(1/S_r)^2$, so that

$$N\left(\frac{1}{S_{Bp}}\right)^2 = \sum_1^N \left(\frac{1}{S_r}\right)^2 \tag{12}$$

and

$$S_{Bp} = \sqrt{\left[\frac{N}{\sum_1^N (S_r)^{-2}}\right]}, \tag{13}$$

where N is the number of rows in cross-flow and

$$S_r = (n_r - 1)(Y - d)L_3 \tag{14}$$

or

$$S_r = n_r[\sqrt{(4X^2 + Y^2)} - 2d]L_3, \tag{15}$$

whichever is the smaller.

Where there is a space between the tube bundle and the shell there will be some interchange of fluid between the bundle and the by-pass space due to the varying flow resistance per row of the tube bank. This interchange is generally ignored, though it is not always trivial.

3.3 *The by-pass space between tube bundle and shell*

Tinker [18] states, "The co-ordinations of Grimison for in-line flow for a fixed longitudinal tube spacing indicate that, for a given tube diameter and mass flow rate, f varies almost inversely as the transverse space between the tubes. On this basis, f_c for a given Re, would equal

$$\frac{X - d}{0{\cdot}5(D_1 - D_3)} f_B$$

if another row of tubes were substituted for the shell boundary (where f_B, f_C, are the friction factors appropriate to the cross-flow and by-pass streams respectively). Since the shell boundary is smooth instead of irregular, it is assumed that the overall friction factor of the by-pass route will be 75 per cent of the foregoing expression. The term C_{10} is introduced (as a factor) to adjust the friction factor for the by-pass route for its mass velocity (or Re) which is higher than the mass velocity for the flow through the tube nest", where:

$$C_{10} = \frac{f_{Re,\,C}}{f_{Re,\,B}} \tag{16}$$

and

$$Re_C = \left(\frac{V_C}{V_B}\right) Re_B.$$

As pointed out in 1951 by Fritzsche [19], the Reynolds numbers in the by-pass and tube-bundle streams can only be found by successive approximations.

In his contribution [19] to the discussion on Tinker's [18] paper, and in a later monograph [20], Fritzsche gave some of his own test results in an experiment to discover the effect of by-pass. Fig. 2a shows a test bank of tubes of the type normally used for obtaining pure cross-flow

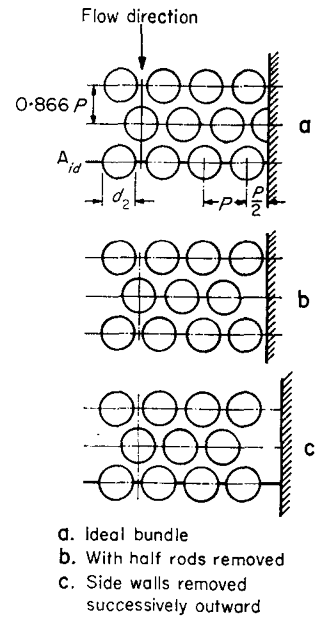

FIG. 2. Fluid-flow models (from Fritzsche [19]).

data. In Fritzsche's tests the half tubes were first removed (Fig. 2b) and the side walls gradually moved outwards, increasing the by-pass area (Fig. 2c). With pressure difference across the bank kept constant, measurements were made of the total flow. The cross-flow area in the tube bank he called A_{id} (ideal) and in tube banks with by-pass it was called A_{eff} (effective) as defined in Figs. 2a and 2c. The results were applied to cylindrical tube banks on the basis of Fig. 3 in which $(A_{eff})_i$ is averaged over all the rows. The ratio of Reynolds number in the arrangement without by-pass to the apparent Reynolds number in the arrangement with by-pass, under the same pressure difference, is plotted against the ratio of A_{eff} to A_{id} for various tube pitches and Reynolds numbers. The apparent Reynolds number is calculated on the assumption that all flow penetrates the tube bundle. The larger pitch ratios (x_T = 1·35 and 1·50) gave Reynolds number ratios independent of absolute Reynolds number, but the closely pitched tubes (x_T = 1·20) gave an increasing proportion of fluid flowing through the tube bank as the Reynolds number increased. The maximum Reynolds number in the tests was 5000 and the tube layout was in the form of equilateral triangles.

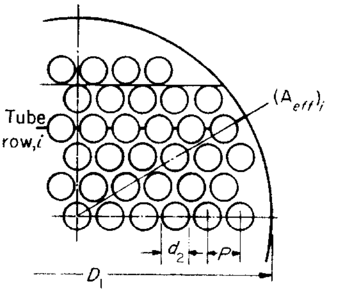

FIG. 3. Tube nest in a cylindrical shell (from Fritzsche [19]).

Bergelin, Bell and Leighton [21] performed tests very similar to those of Fritzsche in the Reynolds number range 3000 to 18 000, using equilateral pitch tube banks in two arrangements. In correlating the results it was again assumed that, for a given pressure difference, the flow through the tube bank was independent of the flow in the by-pass. With both tube arrangements (pitch ratios 1·25 and 1·5) it was found that the proportion of fluid flowing in the by-pass decreased with increasing Reynolds number, though this effect was more marked with the more closely pitched tube bank. The authors plotted friction factors for the by-pass channel against the by-pass Reynolds number; the friction factor was defined by equation (9) in which N was given a value equal to the number of "close approaches" of the tubes to the shell (in this case, half the number of tube rows). The results for a given tube arrangement correlated well on this basis and showed little variation in friction factor with Reynolds number, but the by-pass friction factor for the 1·5 pitch ratio was approximately 30 per cent lower than that for the 1·25 pitch ratio.

In a later paper Bell [22] expressed these results with the empirical equation

$$\frac{\Delta p_{BP}}{\Delta p_I} = \exp\left[-3{\cdot}8\,F_{BP}\left(1-\frac{2N_S}{N}\right)\right]. \quad (17)$$

The term $2N_S/N$ is included to represent the results obtained when sealing strips were placed in the by-pass space to restrict the flow there.

3.4 *The baffle window*

Donohue [23], in 1949, using the test data given by Short [3], obtained values for the pressure drop through the baffle window by assuming that, in those tests where the baffle spacing was very large, there was negligible pressure drop between the baffles as compared to the pressure drop through the windows. He plotted these values of pressure drop per baffle, against mass velocity in the opening, on a log–log scale and found that the values could be expressed within ±50 per cent by the equation

$$\Delta p_W = 4{\cdot}17 \times 10^{-11}\frac{G_W^2}{(\textit{Specific gravity})} \quad (18)$$

or

$$V_W = 0{\cdot}697\sqrt{\left(2g_c\frac{\Delta p_W}{\rho}\right)}. \quad (19)$$

The average deviation from equation (19) is ±36 per cent. Unfortunately the experimental data on which these data are based were

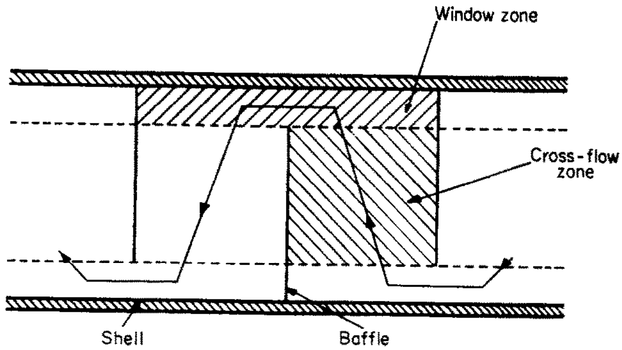

FIG. 4. Delineation of cross-flow and window zones.

obtained from unbored shells (in which there is frequently a high leakage rate around the baffles) and no allowance was made for leakage, so the curve tends to be optimistic.

Drew and Genereaux [24] had earlier suggested a coefficient of 0·7 in equation (19).

Bergelin, Brown and Colburn [25] conducted tests in 1954 to discover the effects of variations in baffle window size and baffle spacing in exchangers without internal leakage. The by-pass flow was also reduced to a minimum by using a tube bundle which fitted closely to the internal bore of the shell and by inserting spacer rods in the larger gaps around the periphery of the bundle. Pressure tappings were placed in the shell-side fluid at points on either side of a window, level with the baffle edge (the distance from the edge is not given), to measure the window pressure drop. The cross-flow zone was defined arbitrarily as the zone between baffle edges, and the remainder of the fluid path (in the turn-around region) as the window zone (see Fig. 4).

When the friction factors for the cross-flow zone are plotted against Reynolds number there is some scatter due to baffle configuration, but the line for simple cross-flow is fairly representative in the turbulent region. If the pressure drop through the window zone (as defined) is plotted against window velocity (based on the free area in the window in the plane of the baffle) there is a spread of data, the pressure drop being greater where there is a high ratio of cross-flow to window velocity. If the pressure drop in the window zone is plotted against V_Z, the geometric mean of the cross-flow and window velocities, the correlation is much improved.

For turbulent flow the head loss in the cross-flow zone was found to be fairly well represented by the relationship

$$\frac{\Delta H}{N} = 0{\cdot}6 \frac{V_M^2}{2g_c} \tag{20}$$

where dimensions are consistent and V_M is the average maximum velocity through a tube row in the cross-flow zone.

Bergelin, Brown and Colburn [25] state, "In the baffle window there is . . . flow across the tubes, plus a complete reversal of direction, and also flow along the tubes. If, for turbulent flow, the friction for the flow along the tubes is neglected, and the loss, owing to reversal of direction, is estimated at two velocity heads, the total pressure drop can be estimated as

$$\Delta p_W = (2 + 0{\cdot}6 N_W) \frac{\rho V_Z^2}{2g_c} \tag{21}$$

where the velocity head $V_Z^2/2g_c$ is determined from the geometric mean velocity and N_W is an estimated number of [tube rows] passed in the window." N_W was estimated as half the number of tube rows in the window less the outside one. The window zone pressure drop calculated on this basis was within ± 23 per cent of the measured loss.

3.5 *Leakage paths through and round baffles*

Clearance between the baffle and shell and between tube and baffle provide leakage paths in the form of fine annular orifices: the coefficients for these orifices depend on whether the annulus is concentric or tangential, but it would seem safer to assume that the majority are tangential. Curves of coefficients for both tangential and concentric orifices of fine clearances were plotted by Bergelin, Bell and Leighton [26], in 1958, for various Reynolds numbers up to 20 000 against "shape factor" (length/width). These authors tested a number of experimental heat exchangers with known internal clearance and a by-pass area which had been made as small as possible. It was estimated that the flow area of the by-pass was about 15 per cent of the cross-flow area. The pressure drop was determined for various leakage areas and the ratio of pressure drop with leakage to pressure drop without leakage was plotted against the ratio of total leakage area to cross-flow area. The correlation is necessarily crude, but gives an indication of the order of pressure drop that may be expected.

It was observed that the effect on pressure drop of a given area of shell-to-baffle leakage is about twice that of the same area of tube-to-baffle leakage. Bell [22] illustrates these results with plots of $\Delta p_L/\Delta p_{NL}$ against S_L/S_B for two series of tests. In the first there was no leakage between baffles and tubes, but the leakage between the baffles and the shell was progressively increased; in the second the leakage between the baffles and tubes was progressively increased while the other leakage was held at zero. The results of the second test may be expressed as

$$\frac{\Delta p_L}{\Delta p_{NL}} = 1 - 2{\cdot}7\left(\frac{S_{TB}}{S_B}\right)^{0{\cdot}35} \tag{22}$$

If it is assumed that the effects of the leakages are additive if each leakage is weighted by its corresponding area, then

$$\left(1 - \frac{\Delta p_L}{\Delta p_{NL}}\right)_{\text{exchanger}} = 2{\cdot}7\left(\frac{S_{TB}}{S_B}\right)^{0{\cdot}35} \frac{S_{TB} + 2S_{SB}}{S_{TB} + S_{SB}}. \tag{23}$$

3.6 *Stream velocities*

Tinker [18], in 1951, attempted a correlation of heat-exchanger performances by dividing the total flow through the shell into the various flow paths in Fig. 1, and calculating the velocities and quantities in each stream. His nomenclature is formidable and includes constants numbered C_1 to C_{13} and an additional constant x. Some of these are simple functions of the physical dimensions of the heat exchanger, but at least half are "estimates" of various values necessary to the calculation. Tinker did not explain how he arrived at some of these estimates, which makes his method, as it stands, difficult to apply. However, the notes on the components of pressure drop in Sections 3.1–3.5 should assist in the application of a Tinker-type analysis.

Tinker defines the various streams as below (see Fig. 1).

Stream — Description

A — "Leakage stream through the annular space between tubes and baffle holes of one baffle."

B — "Cross-flow stream through the heat-transfer surface between successive baffle windows. It will be noted that this stream is made up of B_1 (a portion of fluid passing through baffle windows) plus portions of the A streams." For usual heat-exchanger proportions, Tinker says equation (24) represents a close approximation of the average cross-flow stream:

$$B = B_1 + 0{\cdot}5A. \tag{24}$$

C — "By-pass on one side of tube bundle flowing between successive baffle windows."

D — "Leakage stream between shell and edge of one baffle."

Q — "The total fluid passing through the heat exchanger:

$$Q = 2A + B_1 + 2C + 2E." \tag{25}$$

From equations (24) and (25):

$$Q = 1{\cdot}5A + B + 2C + 2E \tag{26}$$

$$Q = Q_A + Q_B + Q_C + Q_E. \tag{27}$$

The relative pressure differentials across the flow streams are given below

Flow stream identification	Description	Average pressure differential producing flow	Relative pressure differential producing flow
A	Flow through holes of one baffle	$\Delta p_W + \Delta p_B = \Delta p_B(1 + x)$	$1 + x$
B	Flow across tubes	Δp_B	1
C	Flow around tube bank	Δp_B	1
E	Flow past one baffle	$\Delta p_B + \Delta p_W = \Delta p_B(1 + x)$	$1 + x$

Clearly (though Tinker does not mention it) the value of x will depend on the velocities through the window and the tube bundle. Since these are not known, a first approximation must be assumed, to be improved later by successive approximations.

The "flow resistance" in velocity heads is then determined for each component of flow. (By "flow resistance" is meant the ratio of the pressure drop to the velocity head $V^2/2g_c$, e.g. the flow resistance of a sharp-edged orifice of coefficient 0·6 is $0{\cdot}6^{-2}\rho$ since the pressure drop is $0{\cdot}6^{-2}\rho\, V^2/2g_c$.) The ratio of the relative pressure differential to the flow resistance gives the square of the relative velocity in each stream; the product of the relative velocity and flow area gives the relative magnitude of each stream, expressed as Q_{AR}, Q_{BR}, Q_{CR} and Q_{ER}.

Then from equation (27):

$$Q_B = \frac{Q_{BR}}{Q_{AR} + Q_{BR} + Q_{CR} + Q_{ER}} \times Q \qquad (28)$$

and the magnitudes of the other stream quantities may be similarly determined.

It is in determining the components of flow resistance that the chief difficulty arises; Tinker gives little help with this. Since they are usually functions of Reynolds numbers, which are not known, an iterative method is called for, which is tedious and time consuming; yet with the increasing use of fast automatic computers in design calculations, iterative processes lose their disagreeable qualities and lead to greater accuracy in design.

3.7 *Recommendations*

In calculating the flow resistance of the cross-flow zone between one pair of baffles, end limits for the zone must be arbitrarily assumed, and the choice will largely be determined by the treatment to be accorded to the window region. If equation (19) by Donohue [23]:

$$V_W = 0{\cdot}697 \sqrt{\left(2g_c \frac{\Delta p_W}{\rho}\right)} \qquad (19)$$

is to be used for calculating the flow resistance in the plane of the window, then some allowance must be made for the change in velocity (in both magnitude and sense) in the fluid as it approaches and leaves the window, and it will probably be assumed that the cross-flow zone ends in the plane through the centres of area of the windows parallel to the baffle edge. Such a course would, at the present state of the art, present many uncertainties. It seems preferable to regard the cross-flow zone as being bounded by the planes through the baffle edges, and the remainder of the fluid path as the window zone (see Fig. 4).

In the cross-flow zone thus defined, the flow throughout will be very nearly normal to the tubes, and the correlations of Grimison [8] (see Section 3.1) or the equations of Jakob [9], equations (10) and (11), may be applied to determine the friction factor, and thence the flow resistance, for the B stream. The term $G_{\max}$ used in determining the Reynolds number should be the effective mass velocity based on S_{Bp} in equation (13).

For the calculation of the flow resistance in the by-pass or C stream, use should be made of of the data of Fritzsche [19, 20] for Reynolds numbers up to 8000 or of the data of Bergelin, Bell and Leighton [21] for Reynolds numbers up to 18 000. For Reynolds numbers beyond

18 000 it seems preferable to extrapolate the data of Bergelin, Bell and Leighton [21] rather than to use the method of Tinker [18], (see Section 3.3), which has no experimental justification.

The flow resistances in the internal clearances (the A and E streams) may be obtained by reference to the curves of Bergelin, Bell and Leighton [26] for friction factors in fine annular orifices.

To determine the factor x the window pressure drop should be computed from equation (21):

$$\Delta p_W = (2 + 0{\cdot}6N_W)\frac{\rho V_Z^2}{g_c} \tag{21}$$

where

$$V_Z^2 = \frac{Q_B + Q_C}{S_W} V_B. \tag{29}$$

From equation (9) the pressure drop in the cross-flow zone is

$$\Delta p_B = \frac{4f\,N\,V_B^2\rho}{2g_c} \tag{30}$$

and

$$x = \frac{\Delta p_W}{\Delta p_B} = \frac{(2 + 0{\cdot}6N_W)[S_{Bp} + (V_C/V_B)Sc]}{4f\,N\,S_W}. \tag{31}$$

Values of V_C/V_B and f must be assumed for a first approximation.

The total pressure drop in the baffled section of the heat exchanger, i.e. between the end baffles, will be

$$\Delta p_W \times N_b + \Delta p_B \times (N - 1).$$

The pressure drop in the end zones (between the nozzles and the end baffles) must be determined separately by an analysis similar to that for the baffled section, and to this must be added the pressure drop in the nozzles.

Alternatively, the pressure drop in the cross-flow zone may be calculated first for an ideal tube bank without by-pass or leakage and assuming that it is penetrated by the whole fluid flow. The pressure drop may then be corrected for by-pass and leakage using equations (17) and (23). Bell [22] gives a step-by-step procedure for calculating the pressure drop for the entire exchanger, using this method. Whiteley [27] has used Bell's method to calculate the pressure drop for nine exchangers in which the pressure drop was measured, and has compared the results with the predictions from other correlations. Of these correlations, Bell's is the only one which takes into account all the variables of baffle cut, by-pass and leakage, and the error was between −10 per cent and +47 per cent of the measured pressure drop, with an average error of 15 per cent. On the other hand Kern's [28] correlation gave errors between −57 per cent and +885 per cent, with an average error of 388 per cent. Bell's method is much simpler to use than the Tinker analysis, but the latter is fundamentally more sound and capable of piecemeal improvement as better data become available.

4. HEAT TRANSFER

4.1 *Rectangular banks of tubes in cross-flow*

Colburn [29] published in 1933 a correlation of the existing data for the flow of gases normal to staggered tube banks, in which $Nu(Pr)_f^{-0{\cdot}33}$ was plotted against Reynolds number. The viscosity and thermal conductivity of the gas were taken at the film temperature (arithmetic mean of the surface and bulk-fluid temperatures); the characteristic dimension was the tube diameter and the velocity was determined by the minimum free area for flow. Between Reynolds numbers of 2000 and 32 000 the curve was well represented by the equation

$$\frac{hd}{k_f} = 0{\cdot}33\left(\frac{C_p\mu}{k}\right)_f^{0{\cdot}33}\left(\frac{G_{max}\,d}{\mu_f}\right)^{0{\cdot}6}. \tag{32}$$

For in-line arrangements Colburn suggested a similar equation with a different numerical coefficient:

$$\frac{hd}{k_f} = 0{\cdot}26\left(\frac{C_p\mu}{k}\right)_f^{0{\cdot}33}\left(\frac{G_{max}\,d}{\mu_f}\right)^{0{\cdot}6}. \tag{33}$$

Tucker [30] showed in 1936 that, in staggered tube banks five rows deep, equation (34) was valid with an average deviation of ±5 per cent for air for wide variations in both longitudinal and transverse tube spacing with values b and n equal to 0·30 and 0·6 respectively.

Grimison [8] has correlated the extensive data of Huge [6] and Pierson [7] in the Reynolds number range 2000–40 000, for both staggered and in-line arrangements with air, in the form of equation (34):

$$\frac{hd}{k_f} = b\left(\frac{G_{\max} d}{\mu_f}\right)^n, \tag{34}$$

with tabulated values of b and n for different tube arrangements. These values are reproduced by McAdams [1].

Grimison's values were largely confirmed by Bressler [15] in the Reynolds number range 5000 to 15 000. Bressler also obtained results from arrangements of tubes in which the lateral displacement of the staggered rows was less than half the lateral pitch. His study of the effect of turbulence in the incident air stream suggests that this may give rise to different results from different experimental arrangements.

Gardner and Siller [31], in 1947, obtained extensive data for heating and cooling of oils and of water. They stated that they could be correlated with a viscosity gradient factor of $(\mu/\mu_w)^{0\cdot19}$, though the scatter was such that it is doubtful whether the index 0·19 represents the data any more accurately than the Sieder and Tate [17] index of 0·14.

For turbulent flow ($Re > 2000$) in liquids, McAdams [1] recommends use of equation (35):

$$\frac{hd}{k} = C'\left(\frac{C_p\mu}{k}\right)^{0\cdot33}\left(\frac{G_{\max} d}{\mu}\right)^{0\cdot6}\left(\frac{\mu}{\mu_w}\right)^{0\cdot14} \tag{35}$$

where $C' = 0{\cdot}33$ for staggered arrangements,
$C' = 0{\cdot}26$ for in-line arrangements.

This is an extension of the Colburn formula. A similar extension of the Grimison correlation may be made by application of the Sieder and Tate viscosity gradient factor $(\mu/\mu_w)^{0\cdot14}$.

There is evidence to show that neither the Grimison nor the Colburn correlations may be safely extrapolated beyond a Reynolds number of 70 000. Sheehan, Schomer and Dwyer [33] found in 1954 that, in a tube bank of equilateral triangular pitch (pitch ratio 1·58), the Colburn curve fitted well with water flow up to $Re = 70\,000$. In this range the heat-transfer coefficient was proportional to $Re^{0\cdot6}$; between $Re = 70\,000$ and $Re = 1\,000\,000$ the heat-transfer coefficient was proportional to $Re^{0\cdot8}$ and the value of C' was 0·033. Thus for $Re > 70\,000$ equation (35) becomes

$$\frac{hd}{k} = 0{\cdot}033\left(\frac{C_p\mu}{k}\right)^{0\cdot33}\left(\frac{G_{\max} d}{\mu}\right)^{0\cdot8}\left(\frac{\mu}{\mu_w}\right)^{0\cdot14} \tag{36}$$

Pierson [7] in 1937, Kays and Lo [32] in 1952 and Sheehan, Schomer and Dwyer [33] in 1954 and others have shown that the first few rows of a tube bank have a lower heat-transfer coefficient than the tubes of later rows; for staggered tube arrangements the coefficient in the first row is approximately 0·63 times the mean coefficient in an infinite number of rows. The factor for the fifth row is 0·99. A ten-row tube bank has a mean heat-transfer coefficient equal to 0·93 times the mean coefficient of an infinite tube bank. The mean coefficients for one to ten rows are tabulated by McAdams [1] as ratios of the mean coefficient for ten rows for both staggered and in-line arrangements; the table may be used in conjunction with the Grimison [8] correlation for tube banks ten rows deep.

The higher coefficients of the rows in the rear of the bank are attributed to increased turbulence.

4.2 *Cylindrical banks of tubes in cross-flow*

For the application of cross-flow heat-transfer data obtained in the laboratory from rectangular laboratory tube banks to cylindrical tube banks, an effective value for the area for flow must be estimated to determine the effective Reynolds number. Most writers appear to adopt the arithmetic mean minimum area for all the tube rows, but reasoning similar to that developed in Section 3.2 appears to give a more rational value. The effective flow area S_{Bh} for calculating heat transfer differs from the effective flow area S_{BP} for calculating pressure drop (see Section 3.2).

The effective Reynolds number is proportional to $1/S_{Bh}$, and the effective heat-transfer coefficient is proportional to $(1/S_{Bh})^{0\cdot6}$. Similarly the heat-transfer coefficient in the rth row is proportional to $(1/S_r)^{0\cdot6}$, so that

$$N\left(\frac{1}{S_{Bh}}\right)^{0\cdot6} = \sum_{1}^{N}\left(\frac{1}{S_r}\right)^{0\cdot6} \tag{37}$$

and therefore

$$S_{Bh} = \left[\frac{N}{\sum_{1}^{N}(S_r)^{-0.6}}\right]^{1.67} \tag{38}$$

4.3 *Cross-baffled heat exchangers*

Kern [28], correlating "industrial data" in 1950 for baffled exchangers with "acceptable" internal clearances and a 25 per cent window cut, employed the hydraulic mean diameter, D_{eK}, of the shell for flow parallel to the tubes as the characteristic dimension in the Reynolds and Nusselt numbers; the mass velocity was calculated from a nominal free flow area computed as

$$S_K = D_1\left(1 - \frac{1}{X_T}\right)L_3.$$

He presented a curve representing

$$\left(\frac{hD_{eK}}{k}\right)\left(\frac{C_p\mu}{k}\right)^{-1/3}\left(\frac{\mu}{\mu_w}\right)^{-0.14}$$

as a function of Reynolds number $G_K D_{eK}/\mu$. The curve was drawn so that the deviation of the test points (not shown) ranges from 0 to approximately 20 per cent high. For Reynolds numbers between 2000 and 1 000 000 the data was said to be represented by the equation

$$\frac{hD_{eK}}{k} = 0.36\left(\frac{G_K\ D_{eK}}{\mu}\right)^{0.55}\left(\frac{C_p\mu}{k}\right)^{1/3}\left(\frac{\mu}{\mu_w}\right)^{0.14}. \tag{39}$$

This equation and the curve to which it approximates are commonly used in design offices, sometimes to the exclusion of all other methods for calculating the shell-side heat-transfer coefficient. While it undoubtedly has the merit of simplicity, it must be recognized that equation (39) takes no account of effects due to variations in window cut, baffle spacing, leakage between the baffles and the shell and tubes and in the space between the tube bundle and the shell. For example, Tinker [18] has estimated that the flow through the tube bundle varies between 12 and 60 per cent of the total shell-side flow in liquid coolers, which are very sensitive to changes in leakage or by-pass streams; Kern's correlation gives no help in predicting the results of such changes.

Donohue [23], in 1949, correlated the data of Short [3], Heinrich and Stückle [34], Bowman [35], Gardner and Siller [31] and Tinker [36], for both bored and unbored shells. He found that, in any of Short's heat-exchanger models, variations in baffle spacing could be correlated by an equation of the form:

$$\frac{hd}{k} = C''\left(\frac{G_Z\, d}{\mu}\right)^{0.6}\left(\frac{C_p\mu}{k}\right)^{0.33} \tag{40}$$

The cross-flow velocity was taken as the velocity in the tube row at or near the widest section of the shell, i.e. in the centre-line plane parallel to the baffle edges. Variations in window opening could be correlated with a similar equation provided that the window area was not less than 15 per cent of the baffle area. Where there is an appreciable viscosity gradient in the shell-side fluid the Sieder and Tate [17] viscosity gradient factor $(\mu/\mu_w)^{0.14}$ must be applied to equation (40) to give

$$\frac{hd}{k} = C''\left(\frac{G_Z\, d}{\mu}\right)^{0.6}\left(\frac{C_p\mu}{k}\right)^{0.3}\left(\frac{\mu}{\mu_w}\right)^{0.14} \tag{41}$$

Bowman's [35] tests on a line of commercial heat exchangers with unbored shells had values of C'' in equation (41) of 0·31, 0·21, 0·23, 0·26, 0·20, 0·20, for the six exchangers tested. Tinker's tests [36] on ten different industrial heat exchangers were correlated by equation (41) with a value of $C'' = 0{\cdot}25$, virtually all the points being within ± 35 per cent of that line. Short's [3] tests on a wider variety of heat exchangers showed more variation in the value of C''. When C'' was plotted against D_e, the hydraulic mean diameter for cross-flow, Donohue found that

$$C'' = 0{\cdot}84(D_e)^{0.6} \tag{42}$$

where D_e is in feet. Correlated by equations (41) and (42), all the points from Short's tests fell within 25 per cent of the correlation line and the average deviation was 9 per cent. Donohue does not claim that equation (42) is of universal application and says that C'' is fairly constant for a given type of heat exchanger. It is clear that C'' must account for variations in by-pass and leakage area as well as tube arrangement.

The correlation of the data of Short [4], Tinker [37], and Gardner and Siller [31], prepared by the City and Guilds College for the British Shipbuilding Research Association

[38], gives C'' as a function of a "clearance ratio" and a tube arrangement factor with the Reynolds number as re-defined by Short [3] and ranging up to 44 000. The full correlating equation is

$$\frac{hd}{k} = 1{\cdot}9\left(\frac{P-d}{P}\frac{d}{D_1}\right)^{0\cdot4}\left(\frac{S_W}{S_W+S_E}\right)^2 \left(\frac{G_{\text{av}}\,d}{\mu}\right)^{0\cdot6}\left(\frac{C_p\mu}{k}\right)^{0\cdot3}\left(\frac{\mu}{\mu_w}\right)^{0\cdot14}F \quad (43)$$

where

$$G_{\text{av}} = \frac{1}{3}\left(\frac{Q}{S_W}+\frac{Q}{S_M}+\frac{Q}{S_p}\right); \quad (44)$$

$(P - d)/P$ is the ratio of the minimum distance between tube surfaces to the tube pitch, ranging from 0·167 to 0·29;
d/D_1 is the ratio of the tube diameter to the shell diameter in the range 0·0364 to 0·106;
S_W is the free area for flow in the plane of the window;
S_E is the leakage area between the baffles and shell;
$S_W/(S_W + S_E)$ ranges from 0·76 to 0·98; and

$$F = \frac{L_2 + (L_1 - L_2)[2L_3/(L_1 - L_2)]^{0\cdot6}}{L_1}. \quad (45)$$

Tinker's [18] end space factor, F, ranges from 0·67 to 0·95 and Prandtl numbers range from 2·3 to 2000.

A novel feature of equation (43) is the factor $(d/D_1)^{0\cdot4}$ which shows that, for the small exchangers which yielded the data for this correlation, the heat-transfer coefficient diminished with increasing shell size when the tube diameter was constant. There does not appear to be a ready explanation for this and it would be interesting to compare data from larger exchangers.

The model heat exchanger of Bergelin, Brown and Colburn [25] has already been described in the section on pressure drop in the baffle window. It will be recalled that they divided the fluid flow into a cross-flow zone between planes through the baffle edges, and a window zone in the remainder of the fluid path (see Fig. 4). Owing to the different flow patterns in these two zones the coefficients of heat transfer, h_B and h_W respectively, are different; but the total heat transfer for the whole exchanger will be the area sum of these two components, thus:

$$h\,A_T = h_B\,A_B + h_W\,A_W \quad (46)$$

where $A_T = A_B + A_W$ and subscripts B and W refer to the cross-flow zone and the window zone respectively, or:

$$h_B = h \div \left[1 - r + r\left(\frac{h_W}{h_B}\right)\right] \quad (47)$$

where $r = A_W/A_T$.

The cross-flow heat transfer could be represented by:

$$h_B = aV_M^{0\cdot6} \quad (48)$$

and, as a first approximation, it was assumed that the same type of relationship held in the baffle window when the geometric mean velocity was used, or:

$$h_W = aV_Z^{0\cdot6} = aV_M^{0\cdot3}\,V_W^{0\cdot3}. \quad (49)$$

From equations (48) and (49):

$$\frac{h_W}{h_B} = \left(\frac{V_W}{V_M}\right)^{0\cdot3} = \left(\frac{S_B}{S_W}\right)^{0\cdot3}, \quad (50)$$

and substituting in equation (47):

$$h_B = h \div \left[1 - r + r\left(\frac{S_B}{S_W}\right)^{0\cdot3}\right]$$

or

$$h = h_B\left[1 - r + r\left(\frac{S_B}{S_W}\right)^{0\cdot3}\right]. \quad (51)$$

When h was correlated by equation (51), using simple cross-flow values for h_B, the correlation was much improved.

G. A. Brown is quoted by Bell [22] as having advanced this argument by assuming, as before that

$$h_B = a\,V_M^{0\cdot6} \quad (52)$$

but that

$$h_W = \beta V_Z^{0\cdot6} = \beta V_M^{0\cdot3}\,V_W^{0\cdot3}. \quad (53)$$

This leads to the equation

$$h = h_B\left[1 - r + \frac{r\beta}{a}\left(\frac{S_B}{S_W}\right)^{0\cdot3}\right]. \quad (54)$$

Brown found for his experimental exchanger

$$\beta = 108 \left(\frac{S_B}{S_W}\right)^{-0.27} r^{-0.68}$$

and α was 205. Substituting these values in equation (54) gives

$$h = h_B \left[1 - r + 0.524\, r^{0.32} \left(\frac{S_B}{S_W}\right)^{0.03}\right] \quad (55)$$

or

$$h = \phi\, h_B \quad (56)$$

where

$$\phi = 1 - r + 0.524\, r^{0.32} \left(\frac{S_B}{S_W}\right)^{0.03}. \quad (57)$$

Because ϕ is such a weak function of S_B/S_W the latter may be taken as unity and

$$\phi = 1 - r + 0.524\, r^{0.32}. \quad (58)$$

A plot of ϕ against r is given by Bell [22].

Bergelin, Bell and Leighton [26], in 1958, obtained heat-transfer and pressure-drop data in the Reynolds number range 1000 to 10 000 from baffled exchangers with known clearance between baffles and shell and between baffles and tubes. They found that internal leakage had a greater effect on pressure drop than on heat transfer, and that a leakage area between the baffle and shell had more effect than an equal area between baffle and tube. They correlated their data by plotting the ratio of heat-transfer coefficients with and without leakage against a complicated resistance factor suggested by Sullivan and Bergelin [39], but the correlation is only slightly better than that obtained by plotting the ratio of heat-transfer coefficients against the ratio of leakage area to cross-flow area.

Bell [22] has suggested an equation for heat transfer similar to that for pressure drop, equation (23), to determine the effect of leakage. It is

$$1 - \frac{h_L}{h_{NL}} = H \left(\frac{S_{TB} + 2S_{SB}}{S_{TB} + S_{SB}}\right), \quad (59)$$

where H is to be determined experimentally.

He has plotted H against the ratio of leakage to cross-flow areas for all the leakage data obtained at the University of Delaware and he shows a rough correlation.

In another paper in 1959, Bergelin, Bell and Leighton [21] reported tests on a rectangular tube bank with a variable by-pass between the tubes and the walls of the test duct. By varying the by-pass area and measuring the pressure drop and mass flow rate they were able to estimate the amount of fluid passing through the by-pass and tube bundle respectively. The shell-side coefficient, based on the tube-bundle mass velocity, correlated about 10 per cent above that for the ideal tube bundle without by-pass. The values of heat-transfer coefficient were calculated on the basis of the mixed stream outlet temperature on the assumption that there is some exchange of fluid between the bundle and the by-pass stream. The authors cite Cernik as having shown by photographic studies that this exchange is substantial, and this view was confirmed in 1960 by Short [40] for the turbulent region. Bell [22] has expressed the results in the equation

$$\frac{h_{BP}}{h_I} = \exp\left[-1.25\, F_{BP}\left(1 - \frac{2N_S}{N}\right)\right]. \quad (60)$$

Studies of the local heat-transfer coefficients in the tube bundle by Ambrose and Knudsen [41], in 1958, and Gurushankariah and Knudsen [42], in 1959, indicated that:

(a) The local heat-transfer coefficient where the tubes passed through a baffle was two to four times that at the centre of the cross-flow region, (i.e. mid-way between baffles) in the same tubes.

(b) In those tubes which passed through the baffle windows, the local heat-transfer coefficient in the plane of the window was higher than in the between-baffle sections of the same tubes. (In the model tested, the window velocity was 3·12 times the centre cross-flow velocity.)

(c) There were indications of a large eddy zone in the lee of each baffle, with a slightly higher heat-transfer coefficient.

These findings were from an exchanger having a 6-in diameter shell with $\frac{1}{8}$-in clearance on the diameter between baffles and shell. The baffle-tube clearance was $\frac{1}{16}$-in on the diameter. Two tube bundles were tested, one of four and one of fourteen 1-in diameter tubes. Ambrose and Knudsen found lower heat-transfer coefficients in the fourteen-tube bundle, even though the velocity through the exchanger was higher, and they attributed this to turbulent effects in the tube bundle. It is noticeable however, that in the four-tube bundle, the minimum space between the tube surfaces is $1\frac{3}{16}$ in, and the clearance between the outermost tubes and the shell is $\frac{1}{2}$ in. With the fourteen-tube bundle, this clearance is still $\frac{1}{2}$ in, but the space between the larger number of tubes is reduced to $\frac{1}{4}$ in. Thus the by-pass flow in the fourteen-tube bundle may be expected to be much greater than in the four-tube bundle, with a consequent effect on the heat-transfer coefficient similar to that observed.

An eddy zone in the lee, or downstream side of the baffle, was also reported in 1957 by Gupta and Katz [43] who experimented with a glass-shell heat exchanger. Gurushankariah and Knudsen [42] found that the heat-transfer coefficient in the eddy zone was between 1 and 28 per cent higher than the average of the true cross-flow zone. It can be suggested that this increase is due, in part at least, to the high velocity jets issuing through the annular spaces in the baffle holes. The heat-transfer coefficients in the parallel-flow zone were more nearly equal to the cross-flow zone values, being lower for the close spacing and higher where there were fewer baffles. Stachiewicz and Short [44] found values of heat-transfer coefficient in the eddy zones in the lee of the baffles which were lower than the average for the cross-flow zone. There was no internal leakage in their exchanger.

Tinker [18], in papers discussed in Section 3.6, correlated the results from a number of commercial heat exchangers by plotting the product of $Nu(Pr)^{-0.33}$ against the "apparent" Reynolds number (based on the assumption that the whole fluid flow penetrates the bundle and there is no leakage). The result was a wide scatter of data, all falling short of the line recommended by McAdams [1] for liquids in cross-flow. Tinker then correlated the same results by plotting $Nu(Pr)^{-0.33}$ against the "effective" Reynolds number, based on Q_B, the mass flow rate through the tube bundle as determined by equation (28). The result was a very much better correlation with the points straddling the McAdams' line.

Since this method takes no account of the difference in heat-transfer coefficient between the cross-flow and parallel-flow zones, a correction is applied based on the window height. Tinker's recommended window height and correction factors are listed below:

Ratio of diameter shell to length between baffles	Window height as fraction of diameter	Correction factor
8·0 to 2·0	0·20	1·00
1·9 to 1·0	0·35	0·90
0·9 to 0·7	0·45	0·80

The correlation of results presented by Tinker relies, to a large extent, on the correct estimation of certain "constants" which, he says, are obtained from the designer's experience, including the correction factor.

4.4 *Recommendations*

For a quick rating of a heat exchanger, equation (43) is recommended. It may be found that, for a particular line of similar exchangers, a numerical coefficient different from the 1·9 given in equation (43) may be appropriate, but it must be remarked that equation (43) takes no account of the by-pass space between the bundle and the shell. For a less rough-and-ready method the step-by-step procedure given by Bell [22], which incorporates equations (56), (58) and (59), is recommended. A more accurate rating may be expected from the type of analysis suggested by Tinker, which is more fundamental in concept.

It may be possible to eliminate Tinker's correction factor by differentiating between the cross-flow and window zones, according to the definitions of these zones given in Section 3.7 and Fig. 4, and using the geometric mean of the cross-flow and window velocities to determine the heat-transfer coefficient in the window zone, after the style of Bergelin, Brown and Colburn [25] and equations (56) and (58).

For the calculation of the heat-transfer coefficient in the cross-flow zone, the value of

G_{max} in the Reynolds number should be computed from Q_B, obtained from equation (28) and S_{Bh}, obtained from equation (38). In the Reynolds number range, 2000 to 40 000, the Grimison [8] correlation is recommended, with the application of the viscosity gradient factor $(\mu/\mu_w)^{0.14}$ where this is significant. Equation (35) is recommended as an alternative in the Reynolds number range 2000 to 40 000 and may be extended up to a Reynolds number of 70 000. For Reynolds numbers between 70 000 and 1 000 000, equation (36) is recommended for staggered tube banks; for in-line tube arrangements the author suggests that the coefficient 0·033 in equation (36) might be replaced by one of 0·026, based on the ratio of heat-transfer coefficients in in-line and staggered tube arrangements observed by Colburn at lower Reynolds numbers.

Suggestions for dealing with the remaining "estimates" in Tinker's synthesis were made in Section 3.7.

Tinker [18] assumes that the fluid which leaks through the baffle holes is lost from the cross-flow stream and is therefore lost for heat-transfer purposes. Yet the high values of local heat-transfer coefficient at the baffle holes and in the eddy zone, shown by Ambrose and Knudsen [41], and Gurushankariah and Knudsen [42], indicate that Tinker takes a pessimistic view of the effect of tube-hole leakage on heat transfer. Indeed Bergelin, Bell and Leighton [26] found that introducing leakage between the tubes and the baffles in one of their experiments increased the shell-side heat-transfer coefficient over that obtained with baffle-to-shell leakage only.

5. CONCLUSIONS

As a result of the work of many investigators a fairly detailed picture of the internal-flow pattern and heat-transfer characteristics of segmentally baffled shell-and-tube exchangers is beginning to emerge. The data are not yet sufficient to enable precise predictions of performance to be made, but they do permit tolerable estimates and indicate the fields in which more progress is necessary. The effect of tube bundle by-pass has been shown to be marked, and one to which the floating head type of removable bundle is particularly vulnerable. Efforts should therefore be directed towards both reducing this by-pass and estimating more accurately its effect. The by-pass space should be made as small as is practicable and should be blocked against flow by means of sealing strips or similar devices. More experiments are needed to help in estimating resistance to flow in the by-pass space.

The degree of mixing which takes place between the by-pass stream and the stream through the tube bundle is also a profitable field of study in which a start was made in 1960 by Short [40].

The flow pattern and heat-transfer coefficients in the baffle window and its environs are complicated and have not yet been fully described. More study is needed of this important part of the heat exchanger.

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Résumé—On étudie les relations empiriques des données relatives aux dimensions des enceintes d'échangeurs à tubes. On utilise des méthodes qui permettent d'estimer l'importance des courants fluides inefficaces dans l'enceinte et par suite de connaître l'écoulement efficace à travers le faisceau de tubes, les données expérimentales sont obtenues à partir de séries de tubes placés dans un écoulement transversal.

Zusammenfassung—Die am Mantelrohr eines Wärmeübertragers mit Umlenkblechen gemachten Erfahrungen werden diskutiert. Mit Hilfe von korrelierten Versuchsdaten an quer angeströmten Rohrbündeln liessen sich Methoden finden, die Grösse des nicht wirksamen Flüssigkeitsstroms an der Wand zu bestimmen und damit den durch das Rohrbündel tretenden wirksamen Strom zu errechnen.

Аннотация—Рассмотрены эмпирические корреляции опытных данных по теплообмену и перепаду давления в межтрубном пространстве кожухотрубных теплообменников с перегородками. Описаны методы определения величины неэффективных потоков, с помощью которых можно рассчитать эффективное течение сквозь пучок труб, при чем эти методы используются в связи с корреляцией данных, полученных на экспериментальном пучке труб при поперечном обтекании.