Exchanger Design . . .

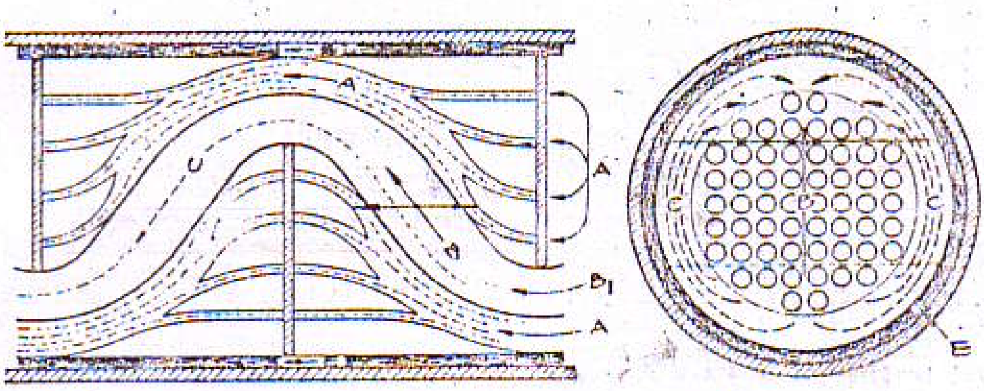

FIG. 1. Schematic diagram of streams on shell side of baffled heat exchangers.

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THE COOPERATIVE RESEARCH PROGRAM on baffled shell and tube heat exchangers at the University of Delaware carried out experimental and analytical studies on the shell-side phenomena in heat exchangers over a course of a dozen years. The chief result of this program was the accumulation of a large amount of carefully obtained experimental data whose final interpretation is perhaps yet years away due to the sheer bulk of information obtained and the complexity of the shell-side problem. From time to time various workers on the project have suggested how these data could be applied to improve the design and design methods of exchangers. Surveying these various design contributions, the author has selected a few pieces and assembled them in a coherent form which appears reasonable for the design of exchangers at this time. This is the first time that the Delaware project has suggested a design procedure encompassing all of the usually encountered shell-side features of an exchanger.

This procedure is derived from generally accepted models of the various fluid flow and heat transfer mechanisms on the shell-side supported by data obtained to specifically illuminate each mechanism individually. At almost every step, it is possible to suggest a more rigorous (hence presumably more accurate) treatment than that employed here; the author's choices of where to draw the line are certainly challengeable and the engineer is encouraged to choose and modify as the problem demands.

Dr. Kenneth J. Bell

FLOW PATTERNS and temperature profiles on the shell side of baffled shell and tube heat exchangers have long been recognized as extremely complex, and the thermal-hydraulic design of such exchangers is more a matter of experience and intuition than science.

To permit the mechanical assembly and disassembly of such a heat exchanger requires that there be small clearances between the tubes and the baffles and between the baffles and the shell. For some exchanger designs there is also a substantial clearance between the outermost tubes and the inside of the shell. Since there are pressure drops across these various alternate flow paths, there is a substantial flow through these paths in addition to the prescribed ideal flow through the tube bundle and through the baffle turnaround (window) from one crossflow section to the next. The presence of these alternate streams adversely influences the local velocities in the exchanger and therefore decreases the local heat transfer coefficients and the local pressure gradient. Additionally, these alternate streams adversely distort the ideal temperature pattern in the exchanger.

Fig. 1[1] shows the complex flow problem on the shell side of the exchanger. Stream B is the desired crossflow stream, Stream A is the leakage stream through the clearance between the tubes and the baffle, Stream E is the leakage stream through the clearance between the baffles and the shell, and Stream C is the bypass flow stream along the outside of the tube bundle between the outermost tubes and the shell.

A rigorous approach to the design of shell and tube exchangers would require that the relative resistances of all of these streams be known and that the interaction and exchange of fluid be known. Such information is not presently available, although a start has been made toward obtaining it, and a design procedure based on such a rigorous analysis would probably be too complex and time-consuming to be of interest in the vast majority of exchanger design calculations.

There are two different approaches to a practical handling of the exchanger design problem that are well established in the literature and in practice. The first is by means of dimensional analysis. The pressure drop and the heat transfer rate may be empirically related to a small number of dimensionless groups embodying the characteristics of the fluid in question and the geometrical features of the shell side. If all of the important geometrical features of the shell side (i.e., tube diameter, arrangement, and pitch; baffle spacing and cutdown; shell diameter; tube-to-baffle, shell-to-baffle, and bundle-to-shell clearances) were to be considered, a complete dimensional analysis would require a vast amount of data which is simply not available and a difficult computation problem in reducing these data to determine the empirical constants. Dimensional analysis in any case lends no great confidence to designs in which it is necessary to extrapolate the original analysis. The second approach attempts to establish the nature of the physical mechanisms existing in the exchanger, to evaluate the effect of these mechanisms on pressure drop and heat transfer, and to sum the individual mechanisms into a total exchanger effect.

The second approach has two important advantages over the dimensional analysis approach. It requires less data, and there is greater confidence in the extrapolation to new and untested designs. On the other hand, the mechanism approach presupposes a clear picture of the controlling processes and how they may be combined into a final analysis. Often the best method is to combine both methods, using the mechanism approach to suggest the proper form of the dimensionless correlations. *The American Society of Mechanical Engineers — University of Delaware Cooperative Research Program on Heat Exchangers* was established to explore at length the mechanism approach to the design of heat exchangers and to use the best of both methods in the development of a design procedure.

The philosophy for this study was to regard the baffled shell and heat tube exchanger as a series of ideal tube banks connected by flow turnaround (window) regions with short-circuiting of the flow by the three alternate leakage flow paths described above. Given this approach to the problem, it follows that one studies first pressure drop and heat transfer in ideal tube banks, then the phenomena occurring during flow through the baffle window or turnaround, and finally the individual leakage phenomena. The study at Delaware covered a period of a dozen years of active experimental work and a continuing period of analysis of the data obtained. The "final interpretation" of the Delaware data, if indeed there is to be any such thing, has not yet been made. But it is now possible to present a picture of the processes occurring on the shell side and a preliminary design method based upon that picture.

Flow across banks of tubes

First task of the Delaware project was to examine the state of knowledge concerning flow normal to rectangular banks of tubes. In the turbulent flow regime many good studies have been published in the literature and correlations proposed for these data. The data and correlation of Pierson, Huge and Grimison[2,3,4] were particularly complete for Reynolds numbers between 4000 and 40,000, and Grimison's correlation had much to recommend it.

In the laminar and transition flow regimes,* however,

*The following arbitrary division of flow regimes is used throughout this paper:

Laminar	$D_o G_m/\mu < 100$
Transition	$100 < D_o G_m/\mu < 4000$
Turbulent	$4000 < D_o G_m/\mu$

No.	Arrangement	Rows	OD, in.	Pitch Ratio
1	Triangular	10	⅜	1.25
2	Square, in line	10	⅜	1.25
3	Square, staggered	14	⅜	1.25
4	Triangular	10	⅜	1.50
5	Square, in line	10	⅜	1.50
6	Square, in line	10	¾	1.25
7	Square, staggered	14	¾	1.25

Models 6 and 7 gave results identical to those of Models 2 and 3, respectively.

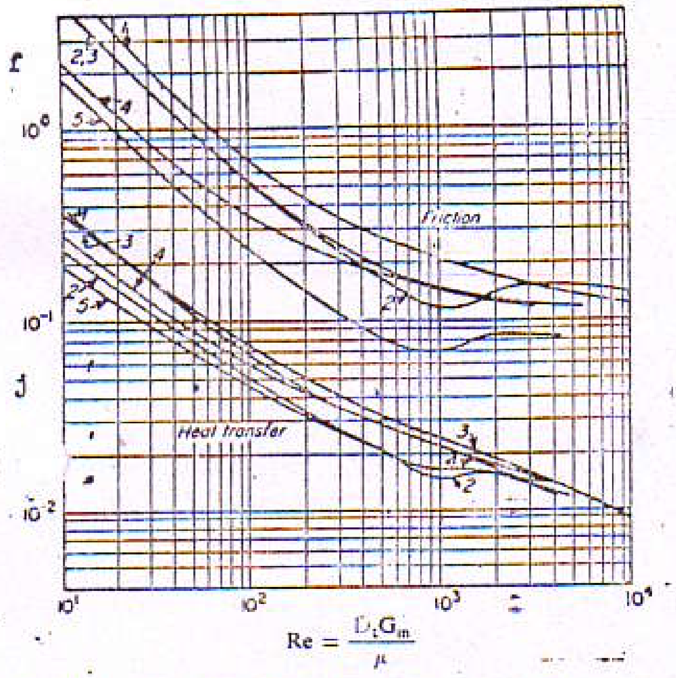


FIG. 2. f and j factors for seven ideal tube banks in laminar, transition, and early turbulent regimes.

there were very few published data, and those were not very consistent. Therefore, using seven tube banks, the Delaware project studied friction factors and j-factors in the Reynolds number region from 1 to 10,000. Later studies extended the curves to even lower Reynolds numbers and showed that they could be safely extrapolated in that direction.

The Delaware data[12] are summarized in Fig. 2, which friction factor and j-factor versus the Reynolds number defined on the basis of the tube diameter and the velocity through minimum crossflow area in the tube bank. Several features of the curves should be noticed. At low Reynolds numbers the slope of the friction factor curve is -1 and the slope of the j-factor curve is $-⅔$ as required by theory. Hence one may confidently extrapolate these lines to arbitrarily low Reynolds numbers as required. In the transition flow regime the staggered tube banks show a smooth, always-decreasing curve with increasing Reynolds numbers, whereas the inline tube banks show a marked dip in both friction factor and heat transfer. Visual experiments here indicated that, for the staggered tube banks, turbulence begins at the downstream tube row and slowly builds up through the bank with increasing Reynolds numbers until the entire tube bank is in turbulent flow. The inline tube banks, however, behave more like rough-walled conduits, and the transition from laminar to turbulent flow is much more definite and corresponds to the region of positive slope of f and j.

It is possible to predict curves of f and j as a function of Reynolds numbers for tube banks of geometries different from those tested. First, the rules of geometric and hydrodynamic similarity hold; i.e., the curves of f and j versus Re for all equilateral triangular tube banks of 10 rows with a 1.25 pitch ratio are identical.* In the turbulent regime Grimison's correlation involves the use of geometrical configuration factors covering the entire range of interest of longitudinal and transverse pitch ratios. In the laminar regime the friction factor curve can be predicted by a new method,[5] though no corresponding method has been developed for predicting the j-factor curve. Bergelin, Colburn, and Hull[6] suggest for the laminar regime

$$j = 1.4\left(\frac{D_v G_m}{\mu}\right)^{-2/3} \pm 20\% \text{ for seven tube banks tested, } \frac{D_v G_m}{\mu} < 100 \quad (1)$$

In the transition region one may fair in the curve between the predicted laminar and fully turbulent curves, making use of the difference in behavior observed between inline and staggered arrangements.

The correction for non-isothermal effects on f and j has been studied, and for most cases it is found that the factor $(\mu/\mu_s)^{0.14}$ is quite satisfactory over the entire Reynolds number range. Hull[6] does recommend the use of $\left(\frac{\mu}{\mu_s}\right)^m$ where $m = 1.57\ Re^{-0.25}$ for non-isothermal corrections on f in the laminar regime, but for most purposes the additional refinement seems to be unnecessary.

There are data from several sources (e.g.[6,7]) that indicate that f (in the transition and turbulent regimes) and j (all flow regimes) are dependent upon the number of tube rows in the direction of flow. In the laminar regime, $j \sim N_c^{-n}$ where by limited data, $n = 0.18$; this is attributed to the building of an adverse temperature gradient which is not completely dissipated by expansion-contraction induced mixing. This effect is important in the design of baffled units. In the turbulent regime, as shown in Table 2, f and j vary markedly with N_c for the first few rows and become essentially identical to asymptotic values for $N_c \cong 6$; the mean f and j for banks of 7 or more rows are within 10% of the asymptotic value. The probable explanation of this effect is that it requires a few rows for fully developed turbulence to be established. Since the baffle turnarounds are very effective in inducing and maintaining turbulence, baffled exchangers operate at essentially the asymptotic values of f and j. In general, it is not necessary to consider the effect of number of tube rows on f. However, the effect on j is very important in the laminar regime and worth considering in the turbulent regime.

The results in Fig. 2 apply only to tube banks in which the tubes extend to or are imbedded into the side walls. If there is any spacing between the outer tubes and the wall, a disproportionate fraction of the flow will proceed through this bypass area and significantly reduce the effective heat transfer rate and the pressure drop. The bypass effect is especially serious in the laminar regime. Table 1 gives some of the results found.[8] A simple empirical rela-

*Strictly speaking, the tube length to diameter ratio should also be constant; this effect is very small and may be neglected if $\frac{L_t}{P - D_t}$ is greater than 5, which is almost always the case in practice. Results for Models 6 and 7 verify this.

1. Effect of Bypassing on Heat Transfer and Pressure

minar Regime.

Percentage of isothermal pressure drop for ideal bundle, Model 4, $100\left(\frac{\Delta p_{BP}}{\Delta p_B}\right)$

Geometry	
Ideal bundle	100
21% bypass area	42
30% bypass area	22
21% bypass area, one seal strip	68

Percentage of shell-side coefficient for ideal bundle, Model 4, $100\left(\frac{h_{BP}}{h_B}\right)$

Geometry	Heating	Cooling
Ideal bundle	100	100
21% bypass area	79	76
30% bypass area	65	60
21% bypass area, one seal strip	94	89

Turbulent Regime.

Percentage of isothermal pressure drop for ideal bundle $100\left(\frac{\Delta p_{BP}}{\Delta p_B}\right)$

Geometry	Model 1	Model 4
Ideal bundle	100	100
21% bypass area	55	43
30% bypass area	38	31
21% bypass area, one seal strip	71	67
30% bypass area, one seal strip	....	63
21% bypass area, two seal strips	81	

Percentage of shell-side coefficient for ideal bundle, $100\left(\frac{h_{BP}}{h_B}\right)$

Geometry	Model 1 Heating	Model 1 Cooling	Model 4 Heating	Model 4 Cooling
Ideal bundle	100	100	100	100
21% bypass area	78	78	80	77
30% bypass area	67	67	73	69
21% bypass area, one seal strip	90	87	94	89
30% bypass area, one seal strip	....	....	88	85
21% bypass area, two seal strips	87	93		

TABLE 2. Variation of $\frac{h_m}{h_\infty}$ with N_c in Turbulent Flow for Staggered Tube Arrays.

N_c	1	2	3	4	5	6	7	8
$\frac{h}{h_\infty}$	0.63	0.76	0.93	0.98	0.99	1.0	1.0	1.0
$\frac{h_m}{h_\infty} = x_t$	0.63	0.70	0.77	0.83	0.86	0.88	0.90	0.91
N_c	9	10	12	15	18	25	35	72
$\frac{h}{h_\infty}$	1.0	1.0	1.0	1.0	1.0	1.0	1.0	1.0
$\frac{h_m}{h_\infty} = x_t$	0.92	0.93	0.94	0.95	0.96	0.97	0.98	0.99

TABLE 3. Dimensions of Cylindrical Test Exchangers.

	Model 9	Model 10
Inside shell diameter, in.	5.25	8.378
Number of tubes	80	470
Length between tube sheets, in.	16.125	16.126
Baffle thickness, in.	0.0625	0.0625
Tube arrangement	Staggered Square	Triangular
Tube pitch ratio	1.25	1.33
Tubes	⅜", 18 BWG	¼", 18 BWG
Total heat transfer area, A, Sq. ft., approx.	9.90	39.40

tionship that fairly closely represents the results of Table 1 and condenses to the proper limiting cases is

$$\left.\begin{aligned}\xi_{\Delta p} &= \frac{\Delta p_{BP}}{\Delta p_B}\\ \xi_h &= \frac{h_{BP}}{h_B}\end{aligned}\right\} = e^{-\alpha F_{BP}\left[1 - \sqrt[3]{\frac{2N_s}{N_c}}\right]}$$

$$\text{for } N_s \leq \frac{N_c}{2}$$

where F_{BP} is the fraction of the total minimum crossflow area that is in the bypass stream and α is a constant taken from the following table:

	laminar	*turbulent*
$\frac{\Delta p_{BP}}{\Delta p_B}$	4.5	3.8
$\frac{h_{BP}}{h_B}$	1.25 ?	1.25 ✓

Baffled cylindrical heat exchangers with no baffle leakage

Following the ideal tube bank studies the next step in the Delaware project was to study the problem of baffled cylindrical exchangers in which the leakage streams were eliminated by experimental technique and design. Shell-to-baffle and baffle-to-tube leakage was eliminated in the test exchanger by coating the baffles with a thin layer of neoprene and then overcutting the neoprene on the shell side and undercutting on the tube holes. The resulting neoprene flaps effectively gasketed the clearances. The bypass stream was minimized by the use of tube layouts that matched quite closely the shell curvature without distorting the unit cell geometry of the tube bank. Baffle spacers were placed in the more open bypass areas. Later data suggests that the fractional bypass area, F_{BP}, of about 0.05, does introduce significant errors which must be considered in the interpretation of the data. Important geometric features of the cylindrical test exchangers are given in Table 3.

Mechanism proposed by the Delaware workers regarded the baffled cylindrical exchanger as a series of ideal tube banks, whose behavior is presumably well understood, connected by turnaround areas (windows) whose influence can be studied as a function of baffle spacing and cutdown. To verify this assumption and to obtain the pressure drop distribution, dummy pressure tap tubes were inserted in the test exchangers so that the pressure drops across a single crossflow section and across a single window section could be measured. Later in the work internal temperature profiles were obtained by a similar technique.

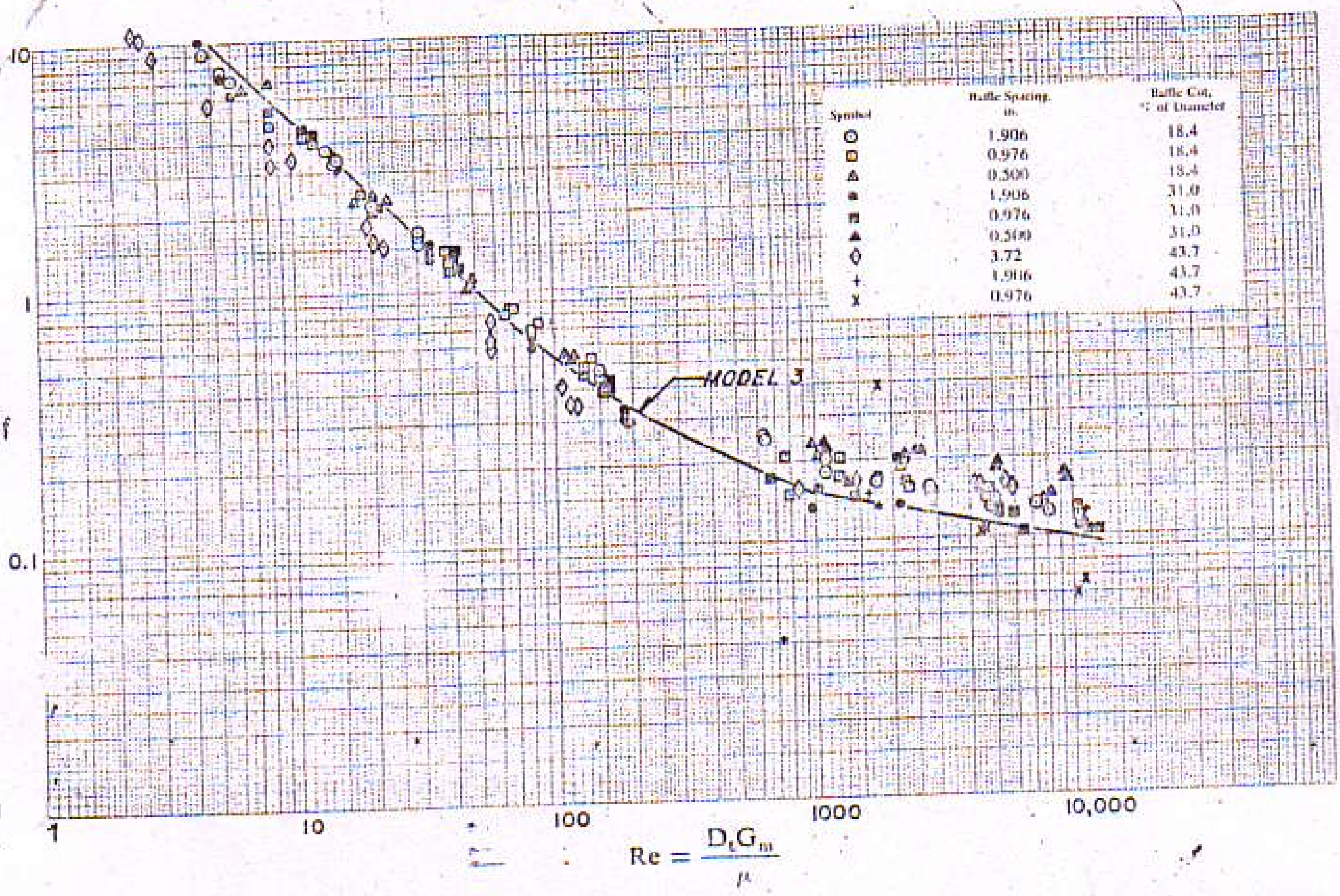
The results of Brown[9] using Model 9 for nine combinations of baffle spacing and cutdown over a Reynolds number range from 600 to 14,000 and of Tompkins[9] in the Reynolds number range from 2 to 200 for the pressure drop across a single crossflow section are shown in Fig. 3. The solid line in Fig. 3 is the curve for the ideal rectangular tube bank having the same tube layout as the cylindrical exchanger. The baffled exchanger results have been examined before plotting for bypass stream effects. Using Equation 2 with $F_{BP} = 0.05$, one finds for the laminar regime

$$\xi_{\Delta p_l} = \frac{\Delta p_{BP}}{\Delta p_{NBP}} = 0.798 \approx 0.80$$

and for the turbulent regime

$$\xi_{\Delta p_t} = \frac{\Delta p_{BP}}{\Delta p_{NBP}} = 0.839 \approx 0.84$$

FIG. 3. Correlation of $f = \frac{2\Delta p_B g_c \rho}{4\xi\Delta_p G_m^2 N_c}\left(\frac{\mu}{\mu_B}\right)^{0.14}$ vs Re for crossflow section, Model 9.



Therefore, for comparison with the ideal tube bank results, the single baffle section pressure drops have been divided by the above ξ's and plotted in Fig. 3. The turbulent flow results vary $^{+70}_{-10}$ % and the laminar results $^{+30}_{-50}$ % about the line for the ideal tube bank.

A later study by Leighton using Model 10, an exchanger with more than twice as many rows of tubes between the baffle cuts as Model 9, gave results within ±10% of the corresponding ideal tube bank results, suggesting that the concept of using ideal tube banks results works better when there is a large number of rows of tubes in the crossflow section. To correct for non-isothermal effects on the friction factor the use of $\left(\frac{\mu}{\mu_B}\right)^{0.14}$ is recommended. There does [s]eem to be some dependence of the exponent on Reynolds [n]umber for very low Reynolds numbers, but the effect appears not to be significant for most cases of practical interest.

Pressure drop through a single window section was measured on these same exchangers for a variety of baffle spacings and cutdowns, again both in the laminar and turbulent regimes. If the window pressure drop is plotted versus the window velocity alone, the pressure drops in the windows vary by a factor of 10 to 1 or more. The highest pressure drops are found for combinations of close baffle spacing and large baffle cutdown, indicating that a high crossflow velocity leads to a high pressure drop in the baffle window. When the window pressure drops are plotted versus the geometric mean window velocity $V_z = \sqrt{V_m V_w}$, the results fall within about ±50% of a straight line, suggesting the use of the geometric mean window velocity as the prime correlating variable.

Colburn[19] suggested that the pressure drop in the window could be separated into a contribution due to the reversal of flow from one crossflow section to the next and a contribution due to the flow of the fluid across a characteristic number of tube rows in the window. The pressure loss due to reversal of flow is taken to be $2\left(\frac{\rho V_z^2}{2g_c}\right)$ for both the laminar and turbulent regimes. Colburn considered the laminar and turbulent regimes separately in his evaluation of the frictional losses due to flow across the tubes in the window. As a measure of frictional losses in laminar flow he defined a viscosity-velocity unit, $\frac{V\mu}{sg_c}$, which is analogous to the velocity head so often used in turbulent flow calculations.

This particular choice of variables is suggested by the fact that the wall shear stress in laminar flow is equal to $\frac{\mu}{g_c}\left(\frac{du}{dy}\right)$ where $\left(\frac{du}{dy}\right)$ is the velocity gradient normal to the solid surface. In irregular geometries such as tube banks, $\left(\frac{du}{dy}\right)$ is of the same order of magnitude as the maximum velocity divided by the minimum spacing from tube to tube, $\frac{V_m}{s}$. For the tube bank geometry used in Model 9 the pressure loss per restriction in the laminar regime is equal to $23\ \frac{\mu}{g_c}\ \frac{V_m}{s}$. Applying this to the case of a baffle window, the velocity used in the viscosity-velocity ait is the geometric mean velocity V_z, and the number of restrictions N_w is taken to be the average number of restrictions crossed by the flow through the window. N_w is approximately the number of tube rows between the baffle cut and the shell, but due allowance must be taken here for the presence of relatively small number of tubes in the outer rows. Any tube row that has fewer than half the tubes of the central row is not counted.

In principle, the bypass effect must be taken into account by multiplying by $\xi_{\Delta p,1}$ which is 0.798 for the case tested. Colburn also applies the viscosity-velocity unit concept to the case of flow parallel to the tubes, again using for the velocity term the geometrical mean velocity V_z, but now using as the characteristic dimension the volumetric equivalent diameter D_v of the window region. As a measure of the length of parallel flow path, he uses the ratio of the baffle spacing to the volumetric equivalent diameter, L_B/D_v. Available data for longitudinal laminar flow suggest 26 velocity-viscosity units per unit $\frac{L_B}{D_v}$. No allowance for bypassing is made, since it is much less important in longitudinal flow (where most of the flow occurs in the central open regions of the unit cell of the tube array), and since the value of D_v reflects any change in the clearance between bundle and shell. Summing these terms gives

$$\Delta p_{w,l} = 23\,\xi\Delta p,_1 \left(\frac{\mu V_z}{g_c s}\right) N_w + 26\left(\frac{\mu V_z}{g_c D_v}\right)\left(\frac{L_B}{D_v}\right) + 2\left(\frac{\rho V_z^2}{2g_c}\right) \quad . \quad . \quad . \quad (3)$$

For calculation of the pressure drop due to flow across the tubes in the window in the turbulent regime, Colburn suggested that 0.6 of a velocity head be lost for each restriction crossed by the entire flow in the window. The use of 0.6 corresponds to a friction factor of 0.15, which is larger than that found for tube banks at Reynolds numbers above about 10,000. In view of the other assumptions and approximations in the argument, further refinement is pointless. Then the total window pressure drop in turbulent flow can be estimated by:

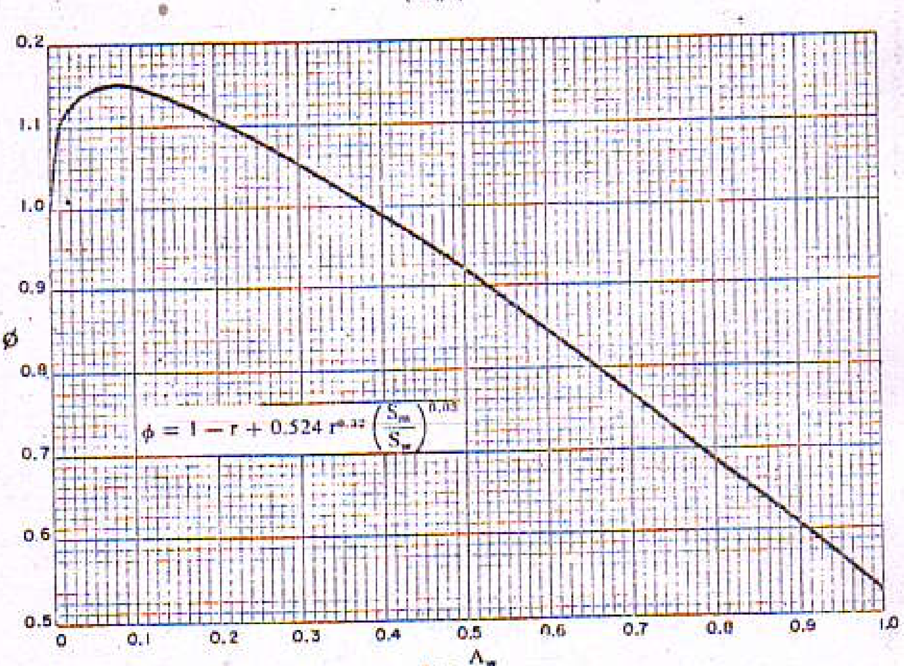
$$\Delta p_{w,t} = (2 + 0.6N_w)\,\frac{\rho V_z^2}{2g_c}. \quad . \quad . \quad . \quad (4)$$

Colburn's method predicts the window pressure drop for the test configurations within ±20% in the laminar regime and from −30 to +20% in the turbulent regime.

Calculation of pressure drop across the entire exchanger by summing individual pressure drops presupposes that all crossflow and window sections behave in essentially the same fashion. Leighton[9] measured all of the individual pressure drops in his exchanger and found that there was no more than ±15% variation in the no-leakage exchanger about the mean value for the window pressure drops and somewhat less variation for the crossflow pressure drops. Brown and Tompkins, who measured pressure drops for only one crossflow section and one window section, found ratios of the measured overall pressure drops to the summation of the measured individual section pressure drops to vary from 0.97 to 1.33. The greatest deviations were for the cases of wide baffle spacing and large baffle cutdown which lead to a low resistance heat exchanger, suggesting that at least some of the remaining effects were due to nozzle losses which could not be accurately calculated but which were estimated to be of the right order of magnitude to account for the difference. Using the method of Colburn to calculate individual pressure drops, and then summing these calculated values, it was found that the calculated overall pressure drop for the various configurations varied from 0.93 to 1.27 times the measured overall pressure drop, again the greatest deviations corresponding to the case of low exchanger resistance and significant nozzle effects. It is thus concluded that the method of summing individual pressure drops is satisfactory for the calculation of the overall exchanger pressure drop in the non-leakage exchanger.

Heat transfer results for nine combinations of baffle spacing and cutdown in both the laminar and turbulent regimes in a non-leakage exchanger have been reported.[9]

FIG. 4. Plot of ϕ vs r at $\left(\frac{S_m}{S_w}\right)^{0.03} = 1$.



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...liminary attempt at correlation of these data is pre... ...ed in the same paper. Brown[11] has suggested a better ...ans of correlation not previously published in the gen... ...al technical literature and which will be considered at some length here. For an exchanger with more than three or four baffles we may write with small error

$$h_{NL}A = h_B A_B + h_w A_w$$

Defining $r = \frac{A_w}{A}$, this equation may be rewritten

$$h_{NL} = h_B (1 - r) + h_w r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

For ideal tube banks in the turbulent regime, it is found[4,12] that

$$h_B = b\, V_m^{0.6} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

where b is a constant depending on tube bank geometry and fluid properties.

In order to write a similar equation for h_w it is necessary to define an appropriate velocity. Based on plausibility arguments we may assume that h_w is similarly related to the geometric mean velocity V_z. Then we may write

$$h_w = \beta V_z^{0.6} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (7)$$

and from Equation 5

$$h_{NL} = b\, V_m^{0.6} (1 - r) + r\beta V_z^{0.6} \quad . \quad . \quad . \quad . \quad . \quad (8)$$

We may now make a series of substitutions:

$$V_z^{0.6} = V_w^{0.3} V_m^{0.3}$$

$$V_w^{0.3} = V_m^{0.3} \left(\frac{S_m}{S_w}\right)^{0.3}$$

Hence

$$V_z^{0.6} = V_m^{0.6} \left(\frac{S_m}{S_w}\right)^{0.3}$$

and from Equation 8

$$h_{NL} = b\, V_m^{0.6} \left[(1 - r) + \frac{r\beta}{b} \left(\frac{S_m}{S_w}\right)^{0.3}\right] \quad . \quad (9a)$$

or

$$h_{NL} = \phi\, b\, V_m^{0.6} = \phi\, h_B \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9b)$$

where

$$\phi = (1 - r) + \frac{r\beta}{b} \left(\frac{S_m}{S_w}\right)^{0.3} \quad . \quad . \quad . \quad . \quad (9c)$$

In the turbulent regime the points clearly fall into groups depending upon the baffle cutdown. The smaller the baffle cut, the greater the overall heat transfer coefficient (for a given crossflow velocity). Let I be the value of h_{NL} when $V_m = 1$ ft per sec for each of the nine baffle configurations studied by Brown. Then from equations 9b and 9c

$$I = b\,\phi = b(1 - r) + r\beta \left(\frac{S_m}{S_w}\right)^{0.3}$$

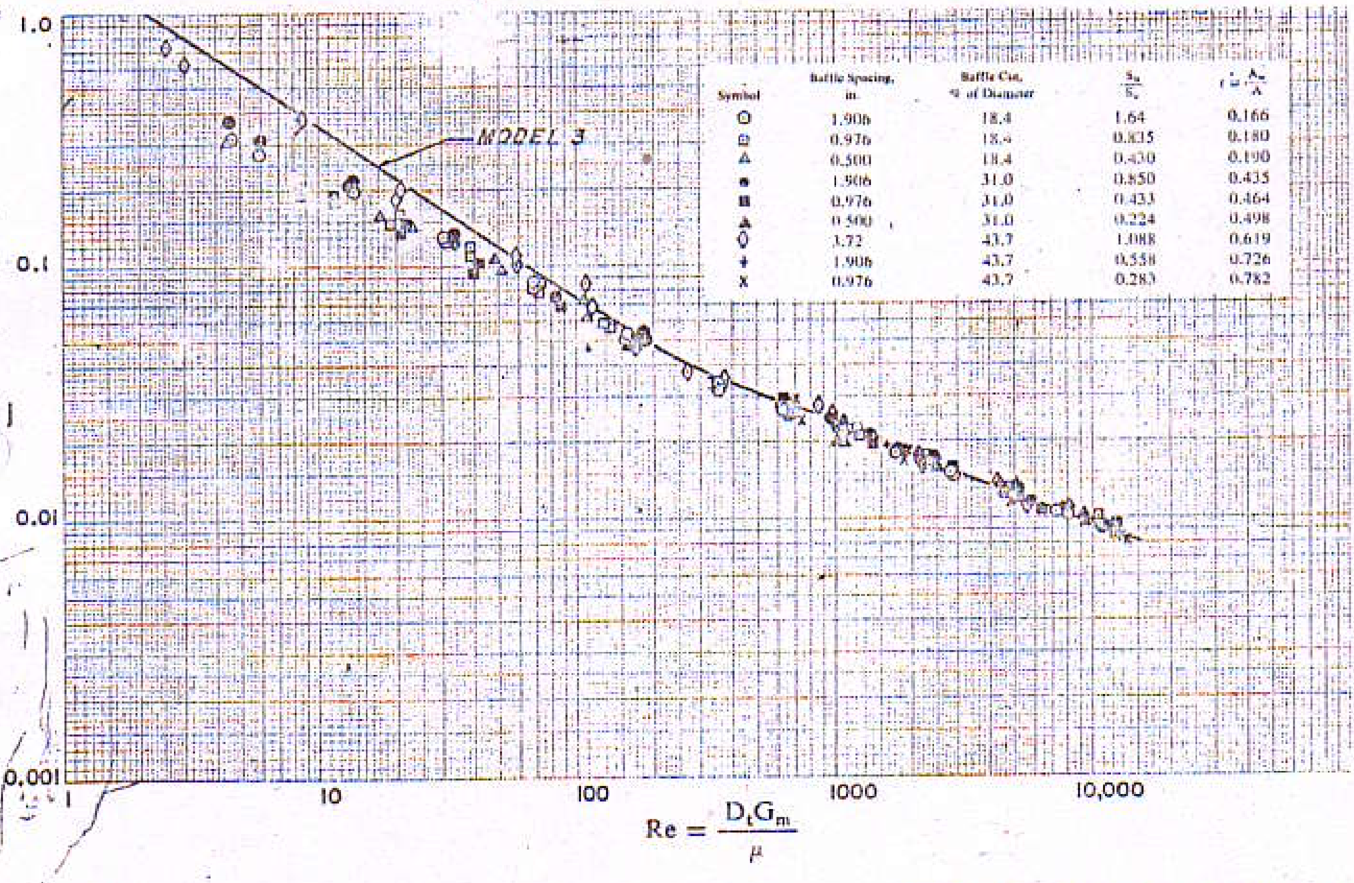
or

$$\beta = \frac{I - b(1 - r)}{r \left(\frac{S_m}{S_w}\right)^{0.3}} \quad . \quad . \quad . \quad . \quad . \quad (10)$$

For each configuration r and S_m/S_w are known, and I can be read from Brown's experimental data lines, and β may then be calculated and correlated versus r and S_m/S_w. Brown finds for his exchanger,

$$\beta = 108 \left(\frac{S_m}{S_w}\right)^{-0.27} r^{-0.68} \quad . \quad . \quad . \quad (11)$$

FIG. 5. Correlation of $j = \left(\frac{h_{NL} x_t}{\phi \xi_u c G_m}\right)\left(\frac{c\mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14}$ vs Re Model 9.



and the value of b is found to be 205[12]. Substituting these results into equation 9a we have

$$h_{NL} = 205 V_m^{0.6} \left[1 - r + 0.524\, r^{0.32} \left(\frac{S_m}{S_w} \right)^{0.03} \right]$$

or

$$\frac{h_{NL}}{\phi} = h_B \qquad \text{(12a)}$$

where

$$\phi = 1 - r + 0.524\, r^{0.32} \left(\frac{S_m}{S_w} \right)^{0.03} \qquad \text{(12b)}$$

Since ϕ is found to be so weak a function of S_m/S_w, the term $(S_m/S_w)^{0.03}$ is taken to be 1 for all cases of reasonable interest and ϕ is given as a function of r in Fig. 4. Before correlating the baffled exchanger turbulent flow results against those for the ideal tube tank, the problem of bypassing and number of rows must be considered. Estimating ξ_h from equation 2 for $F_{BP} = 0.05$ indicates that bypassing resulted in perhaps a 6% decrease in h_o compared to the no-bypass case. On the other hand, if it is assumed that the flow in the baffled exchanger over the range of Brown's tests is in fully developed turbulence, the measured overall coefficient should be increased 5 or 6% compared to that for the 14 tube rows of Model 3. In short, we expect that the bypassing effect and the number of tube rows effect should essentially cancel out for Brown's data, but they should both be considered and evaluated, for each individual case. Therefore, the correlating group to be used in comparing baffled exchangers with bypass but no baffle leakage to ideal tube bank results is $\frac{h_{NL}\chi_t}{\phi\xi_h}$.

In Fig. 5 Brown's results correlated on the basis of $j = \left(\frac{h_{NL}\chi_t}{\phi \xi_h c\, G_m} \right) \left(\frac{c\mu}{k} \right)^{2/3} \left(\frac{\mu_w}{\mu} \right)^{0.14}$ fall within −10 to +20% of the curve for Model 3, the ideal tube bank. The limited turbulent data of Knowles[9] confirm the validity of Brown's method of analysis in the turbulent flow regime.

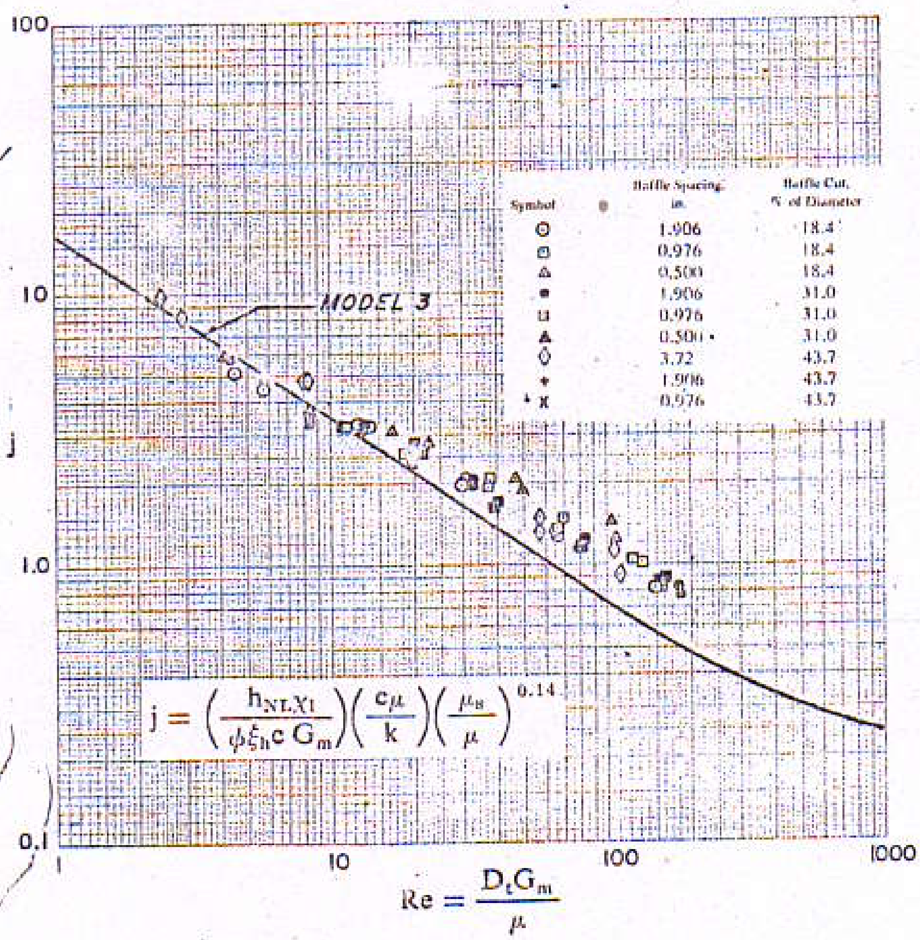
In the laminar flow regime the experimental results on Model 9 due to Tompkins are more scattered and do not show the well-defined variation with baffle configuration that the turbulent regime results do. For the sake of consistency Tompkins' results have been correlated on the same basis as the turbulent flow results in Fig. 5. They are found to be in good agreement for Re > 100 but fall 0 to 50% below the results of the corresponding ideal tube bank (Model 3) at lower Reynolds numbers. In view of the previously mentioned dependence of j on the number of tube rows in the laminar regime, it is appropriate to examine this further. Define the total effective number of cross-flow tubes in series in the exchanger to be:

$$N'_c = (N_b + 1)\, N_c + (N_b + 2) N_w \qquad \text{(13)}$$

From the ideal tube bank results quoted above, $j \sim N_c^{-0.18}$, assume for the present that this relation holds for the baffled exchanger. Hence, to compare results for Model 3 ($N_c = 13$) and the baffled exchanger, the usual j factor for the latter must be multiplied by

$$\chi_l = \left(\frac{N'_c}{13} \right)^{0.18} \qquad \text{(14)}$$

for the number of rows effect and divided by ξ_h (defined by equation 2) for the bypass effect. Division by ϕ as de-



Symbol	Baffle Spacing in.	Baffle Cut, % of Diameter	$\frac{S_m}{S_w}$	$r = \frac{A_B}{A}$
○	1.906	18.4	1.64	0.166
⊡	0.976	18.4	0.835	0.180
△	0.500	18.4	0.430	0.190
●	1.906	31.0	0.850	0.435
⊔	0.976	31.0	0.433	0.464
▲	0.500	31.0	0.224	0.498
◊	3.72	43.7	1.088	0.619
✦	1.906	43.7	0.558	0.726
X	0.976	43.7	0.283	0.782

FIG. 6. Correlation of j vs Re valid at low Reynolds numbers, Model 9.

Symbol	Clearances: Tube-baffle, in.	Clearances: Shell-baffle, in.	S_L, ft²	S_m, ft²
+	0.006	0	0.00181	0.0210
x	0.013	0	0.00392	0.0210
●	0	0.021	0.000881	0.0210
■	0	0.063	0.00264	0.0210
▲	0	0.133	0.00559	0.0210

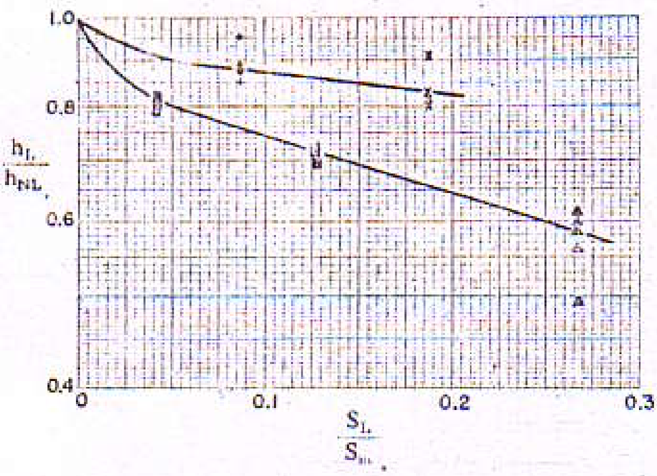


FIG. 7a. Effect of tube-to-baffle and shell-to-baffle clearances on overall shell-side heat transfer coefficient in Model 9.

Symbol	Clearances: Tube-baffle, in.	Clearances: Shell-baffle, in.	S_L, ft²	S_m, ft²
+	0.006	0	0.00181	0.0210
x	0.013	0	0.00392	0.0210
●	0	0.021	0.000881	0.0210
■	0	0.063	0.00264	0.0210
▲	0	0.133	0.00559	0.0210

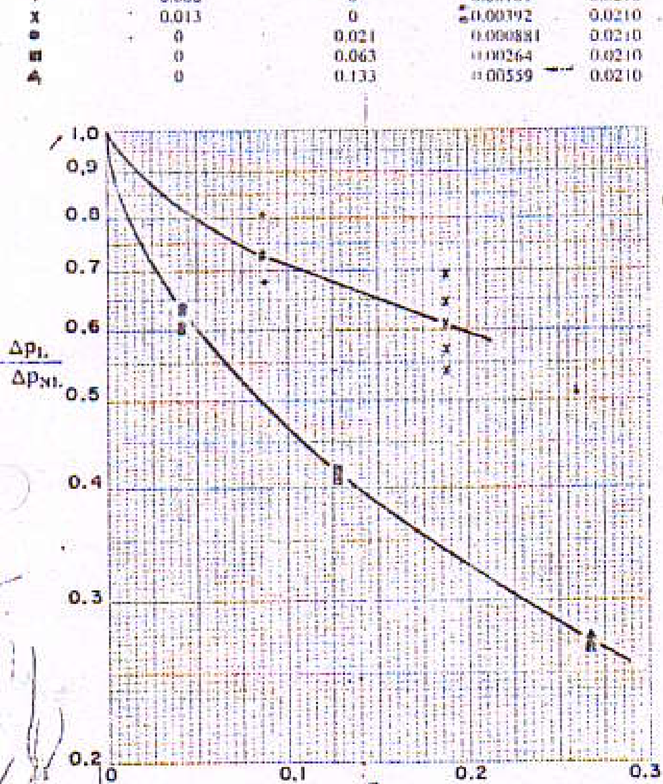


FIG. 7b. Effect of tube-to-baffle and shell-to-baffle clearances on overall shell-side pressure drop in Model 9.

fined by equation 12b is retained even though its application without modification is questionable.

In Fig. 6, Tompkins' data are replotted on the basis of $j = \left(\frac{h_{NL}X_l}{\phi\, \xi_h\, c\, G_m}\right)\left(\frac{c\mu}{k}\right)^{2/3}\left(\frac{\mu_s}{\mu}\right)^{0.14}$. Results for the baffled exchanger now fall within ±15% of the ideal tube bank curve at Reynolds numbers below 15, suggesting that the above argument has some merit. At higher Reynolds numbers the points scatter more and fall above the ideal curve. This behavior indicates that at higher Reynolds numbers, the adverse temperature structure is at least partially destroyed by mixing in the turnarounds. It would not be difficult to obtain an empirical equation for a curve which would fall through the points and give the proper limiting cases at very low and high Reynolds numbers. To do so would suggest undue reverence to a frankly tentative and possibly fortuitous argument. Rather, in the intermediate range, it is better to calculate the limiting values and evaluate the consequences before interpolating.

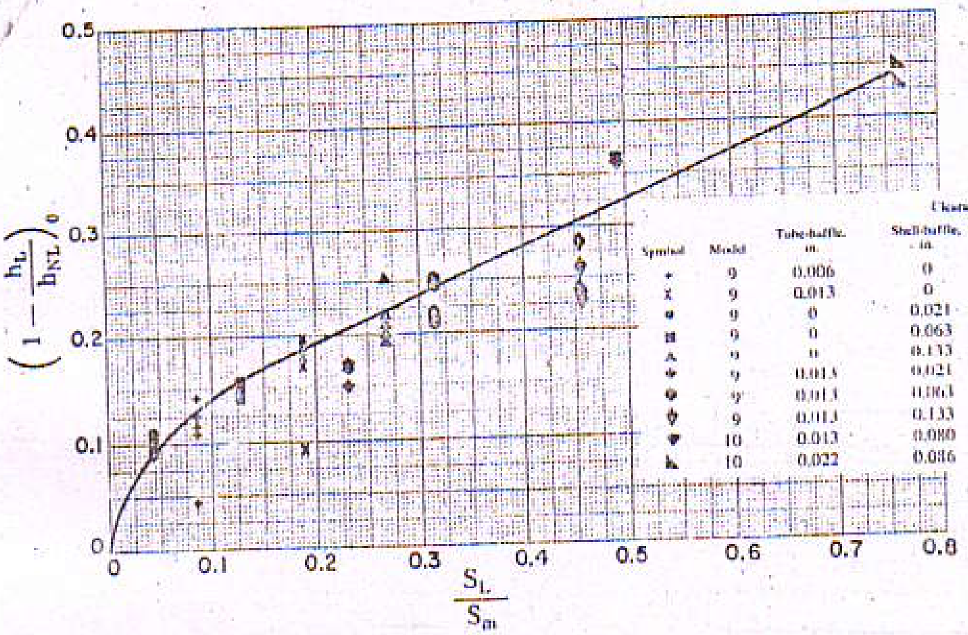
On the basis of the available data the design of non-leakage exchangers in the turbulent flow regime can be entered into with some confidence by the method proposed here. The situation in the laminar flow regime is not as satisfactory and will require extensive additional experimental work to firm up the correlation.

The effect of baffle leakage

The next step in the Delaware project was to examine the effect of clearances between the baffle and the shell and between the tubes and the baffle. Pressure drops exist across these clearances, and consequently there is flow from one baffle section to the next. In effect, part of the shell-side flow is short-circuited, decreasing local velocities and consequently decreasing pressure drops and heat transfer coefficients compared to the corresponding baffle non-leakage exchanger. Also, the leakage streams have a temperature corresponding to conditions upstream of the baffle across which they leak and consequently distort temperature profiles in the stream into which they are leaking. It is necessary to distinguish between the effects of the tube-to-baffle leakage streams and that of the shell-to-baffle leakage streams.

The tube-to-baffle leakage streams flow along the tube in the immediate vicinity of the baffle at a fairly high velocity and hence involve a heat transfer process of significant magnitude on the outside of the tube. The orifice-baffle heat exchanger makes use of this mechanism. In effect, this type of exchanger is nothing but a sequence of baffles with leakage areas around the tubes. The shell-to-baffle leakage stream on the other hand does not come in contact with heat transfer surface in its path and consequently represents a significant loss to the efficiency of the heat transfer process.

Much of the Delaware data relating to the effects of baffle leakage streams is reported in Reference[13], in which it is concluded that baffle leakage has a far greater effect upon pressure drop than it does upon heat transfer. It is also concluded that leakage area between the baffle and the shell has a greater effect upon the heat transfer coefficient and upon pressure drop than does a corresponding amount of leakage area around the tubes. In one particular case it was even noted that increasing the leakage area by increasing the tube-to-baffle clearance while leaving the shell-to-baffle clearance constant resulted in an increased overall heat transfer coefficient and the expected decrease in pressure drop. This is understandable when one considers that the tube-to-baffle leakage stream, which was enhanced in the second experiment compared to the first, has a significant heat transfer efficiency which more than overcame (in this particular case) the effect of an increase in total



Symbol	Model	Clearances Tube-baffle, in.	Clearances Shell-baffle, in.	S_{tb}, ft²	S_{sb}, ft²	S_m, ft²
+	9	0.006	0	0.00181	0	0.0210
×	9	0.013	0	0.00392	0	0.0210
●	9	0	0.021	0	0.000881	0.0210
■	9	0	0.063	0	0.00264	0.0210
▲	9	0	0.133	0	0.00559	0.0210
◆	9	0.013	0.021	0.00392	0.000881	0.0210
◉	9	0.013	0.063	0.00392	0.00264	0.0210
◊	9	0.013	0.133	0.00392	0.00559	0.0210
▼	10	0.013	0.080	0.0154	0.00527	0.0424
◣	10	0.022	0.086	0.0283	0.00566	0.0421

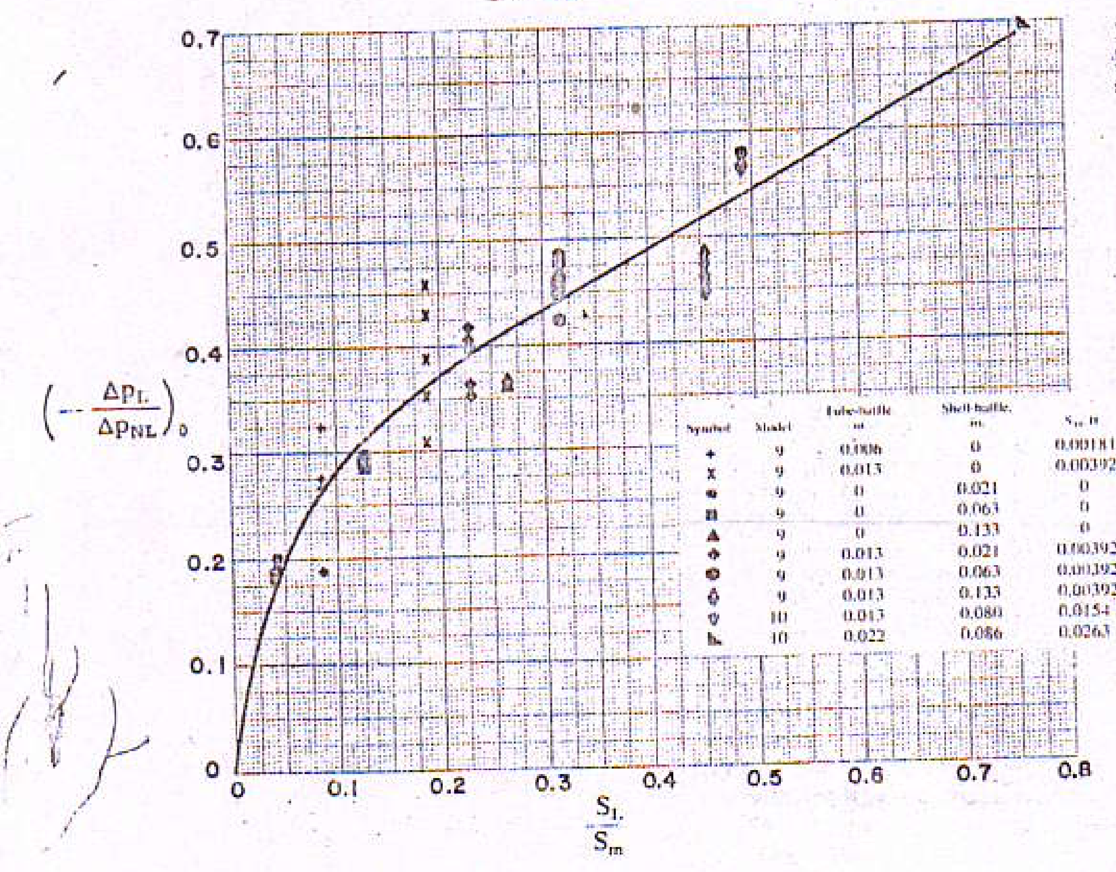
FIG. 8a. Effect of baffle leakage on overall shell-side heat transfer coefficient.

leakage. In that paper it was found that for reasonable clearances the pressure drop may be reduced to 25% and the heat transfer coefficient may be reduced to 50% of the non-leakage values.

To be useful for designing purposes these results must be interpretable quantitatively. Sullivan[11] obtained an analytical solution of the effect of leakage upon pressure drop using an idealized single baffle exchanger (Model 8) to verify his mathematical model. His analysis was also extended to the case of a multi-baffle exchanger. The results of his analysis apply to the pressure drop; and inasmuch as they are rather complex they are not especially useful for the usual purposes of the design engineer. However, a consequence of Sullivan's work was to show that the form of a correlating variable for leakage effects is

$$\frac{S_L C_L}{S_m} (fN_c)^{1/2} \left(\frac{\delta p}{\Delta p_{B}}\right)^{1/2}$$

where S_L is the leakage area per baffle, C_L is a weighted orifice coefficient for the leakage paths, δp is the mean pressure difference across a baffle, and S_m, f, N_c, and Δp_B have their usual definitions. This group is readily interpretable in physical terms: the product $S_L C_L (\delta p)^{1/2}$ is a measure of the rate of flow through the leakage areas, and the



Symbol	Model	Tube-baffle, in.	Shell-baffle, in.	S_{tb}, ft²	S_{sb}, ft²	S_m, ft²
+	9	0.006	0	0.00181	0	0.0210
×	9	0.013	0	0.00392	0	0.0210
●	9	0	0.021	0	0.000881	0.0210
■	9	0	0.063	0	0.00264	0.0210
▲	9	0	0.133	0	0.00559	0.0210
◆	9	0.013	0.021	0.00392	0.000881	0.0210
◉	9	0.013	0.063	0.00392	0.00264	0.0210
◊	9	0.013	0.133	0.00392	0.00559	0.0210
▼	10	0.013	0.080	0.0154	0.00527	0.0424
◣	10	0.022	0.086	0.0263	0.00566	0.0421

FIG. 8b. Effect of baffle leakage on overall shell-side pressure drop.

group $\frac{S_m(\Delta p_B)^{1/2}}{(fN_c)^{1/2}}$ is a measure of flow across the tube bank between the baffles.

The Sullivan group was employed in a correlation proposed in Reference[13]. The correlation is tedious to use. Worse, though it does properly indicate the effect of Reynolds number on leakage, it does not reflect the difference between shell-to-baffle and tube-to-baffle leakage on heat transfer. The definition of C_L could be changed to at least partially correct this, but it appears more rewarding at the present to pursue a simpler approach.

Some studies[9] were made on the effects of each leakage path alone on h and Δp. Results are shown in Fig. 7a and 7b. The abcissa is the ratio of total leakage area per baffle to minimum crossflow area per baffle section. As indicated above the tube-to-baffle leakage affects h and Δp less than does the shell-baffle leakage for the same relative leakage area, and h is less affected by leakage than is Δp. Very roughly, at given $\frac{S_L}{S_m}$, $\left(1-\frac{h_L}{h_{NL}}\right)$ and $\left(1-\frac{\Delta p_L}{\Delta p_{NL}}\right)$ for tube-to-baffle leakage are about half the values of shell-to-baffle leakage. Hence, a single curve of $1-\left(\frac{h_L}{h_{NL}}\right)_o$ vs S_L/S_m is obtained if the values of $\left(1-\frac{h_L}{h_{NL}}\right)$ for shell-to-baffle leakage only are divided by 2 and the values of $\left(1-\frac{h_L}{h_{NL}}\right)$ for tube-to-baffle leakage are divided by 1. For the case of exchangers having both leakage paths, assume that the effects of the leakages are additive if each leakage is weighted by its corresponding area, i.e.,

$$\left(1-\frac{h_L}{h_{NL}}\right)_{Exchanger} = \left(1-\frac{h_r}{h_{NL}}\right)_o \left[\frac{S_{TB}+2\,S_{SB}}{S_L}\right] \quad (15)$$

where $S_L = S_{TB} + S_{SB}$. The same result is found for the pressure drop:

$$\left(1-\frac{\Delta p_L}{\Delta p_{NL}}\right)_{Exchanger} = \left(1-\frac{\Delta p_r}{\Delta p_{NL}}\right)_o \left[\frac{S_{TB}+2\,S_{SB}}{S_L}\right] \quad (16)$$

All of the exchanger leakage results of the Delaware project are plotted on this basis in Fig. 8a and 8b. The agreement is no better than might be expected, but the method has the virtue of simplicity. It should be noted that it is not possible to show all of the points used in Fig. 7a, 7b, 8a, and 8b. Extreme points for each S_L/S_m show the total range of experimental points.

Design procedure

For purposes of shell-side design calculations, the foregoing discussion may be systematized into the following steps. It is assumed as a prerequisite that complete information of the shell-side geometry and the shell-side fluid properties are available. It is also assumed that f and j curves are available for the ideal tube bank geometry corresponding to the shell-side tube layout:

1. Calculate $Re = \frac{D_t G_m}{\mu}$
2. Find j from j vs Re curve for ideal tube bank.
3. Calculate the minimum F_{BP} for the crossflow sections.
4. Calculate ξ_h from Equation 2.
5. Find ϕ from Fig. 4.
6. If Re <100 calculate N'_c from Equation 13.
7. Find χ_t (Re >2000) or χ_l(Re <100) from Table 3 or Equation 14, respectively. For $100 <$ Re < 2000, use $\chi = 1$.
8. Calculate h_{NL} from j. If Re <100, calculate using $\chi = 1$ and using χ_l, then use appropriate limiting case or interpolation as judgment suggests.
9. Calculate S_{TB}, S_{SB}, and $\frac{S_{TB}+2\,S_{SB}}{S_L}$.
10. Find $\left(1-\frac{h_L}{h_{NL}}\right)$ from Fig. 8a, and Equation 15.
11. Calculate h_L from $\left(1-\frac{h_L}{h_{NL}}\right)^{Exchanger}$.
12. Calculate $\xi_{\Delta p,t}$ (Re>100) or $\xi_{\Delta p,l}$ (Re<100) from Equation 2.
13. Find f from f vs Re curve for ideal tube bank and calculate Δp_{BP}.
14. Calculate V_z.
15. If Re <100, calculate D_v (from definition) and $\Delta p_{w,l}$ from Equation 3.
16. If Re >100, calculate $\Delta p_{w,t}$ from Equation 4.
17. Calculate $\Delta p_{NL} = \left(N_B + 1 + \frac{2N_w}{N_c}\right)\Delta p_{BP} + N_B\,\Delta p_w$.
18. Find $\left(1-\frac{\Delta p_L}{\Delta p_{NL}}\right)_{Exchanger}$ from Fig. 8b, and Equation 16.
19. Calculate Δp_L from $\left(1-\frac{\Delta p_L}{\Delta p_{NL}}\right)_{Exchanger}$.

General design rules

The foregoing design procedure is suited to the problem of rating exchangers, that is, determining the suitability of a given exchanger to carry out a given job. The general procedure for exchanger rating is to start with the smallest or least expensive exchanger which it is thought may succeed in carrying out the required heat transfer and to determine whether in fact it will do so. The rating procedure is continued for other sizes of exchangers until an exchanger is found which is calculated to do the job at the least cost.

A problem of equal interest and importance is to design in part, or in toto, an exchanger specifically to do a given job. This latter problem would most likely arise in the case of a large required service in which the possible savings in apparatus cost would more than compensate for the added time involved in design and special machining and construction of the exchanger. For this purpose the procedure suggested above is not especially satisfactory; nor are other present methods of exchanger design particularly suited to this problem. Ideally, what would be desired here would be an analytical statement of exchanger performance as a function of all of the independent design variables of the exchanger and fluid properties. The optimum design—that which solves the required heat transfer problem at the lowest annual cost of exchanger and associated apparatus—would then be found by solution of the analytical performance equations to give the minimum cost. Such a procedure is at this time impossible because of lack of knowledge of the analytical expressions required to completely define exchanger performance and the difficulty of solving these equations for the optimum configuration even if the equations were available. It is, however, possible to make certain statements regarding certain design variables which will at least somewhat limit the manifold of possibilities of exchanger designs to meet a given service.

1. It may be stated that on the basis of heat transfer coefficient obtained for a given amount of pumping power expended per square foot of heat transfer service that staggered tube arrays with small pitch ratio are in general slightly more efficient than staggered arrays with large pitch ratio or inline arrays. The inline arrays are particularly inefficient in the transition Reynolds number regime.

2. On the basis of heat transfer coefficient obtained for a given pumping power expended per unit area, it is found

at all baffle configurations and all leakage and bypass conditions are inferior to that of the ideal tube bank. From this it follows that, insofar as possible, and within certain limitations to be discussed below, leakage and bypass flow should be minimized and the exchanger configuration approach as closely as possible to the ideal tube bank. Rigorous application of this principle would lead to excessive pumping power losses in the baffle turnarounds. In the light of available evidence, the optimum case seems to be that in which the velocity through the window is about equal to the maximum velocity through the tube bank.

3. By far the most serious loss of efficiency in the operation of a typical exchanger is bypass around the tube bundle between the bundle and shell. In any case this bypass stream must be minimized by designing the clearance between the tube bundle and the shell to be as small as possible. For exchanger designs such as the pull-through floating head type, which require a significant clearance, seal-strips should be inserted between the shell and the outermost row of tubes at two or three or even more points along the flow path in each crossflow section. To be most effective, these strips must block the entire bypass channel.

4. The next most serious loss of efficiency is due to leakage between the baffle and the shell. The shell and baffle system should be carefully designed and manufactured so as to keep this clearance to the absolute minimum required for insertion and removal of the tube bundle. Some manufacturers go so far as to bore the inside of the shell or expand it over a mandrel to bring the shell as close to a true circular cylinder as possible and thereby minimize the shell-to-baffle clearance.

5. The least serious of the leakage and bypass streams is the tube-to-baffle leakage stream. In general this stream should be minimized, but not if by so doing the shell-to-baffle leakage stream is substantially enhanced. In general, the leakage area between the tubes and baffle should be greater than that between the baffle and the shell.

6. Since the entire flow passes across essentially the entire tube bundle in the two end crossflow sections in contrast to the internal crossflow sections, the end sections should have a substantially larger baffle spacing than the internal ones.

Illustrative example

This example is not a complete design problem, but is intended to display the method of design proposed in this paper and to clarify definitions. It also seemed desirable for purposes of comparison to work with an exchanger whose performance was known. The design procedure will be applied to a later test exchanger configuration of the Delaware project designated as 10-TL3, whose dimensions are given in Table 4. There are no seal strips. The shell-side fluid is Gulf 896, 23,600 lb per hr at 140.0 F in and 159.0 F out. The tube side was water in at 175.5 F and 173.7 F. The tests on this configuration were not employed to develop the correlations presented here and hence represent a semi-independent check on the method.

1. Viscosity of Gulf 896 at 149.5 F is 4.50 lb per ft hr. The tube rows immediately above and below the centerline have 24 tubes each. Therefore

$$S_m = \frac{[8.502 - 24(0.249)]\ 2.643}{144} = 0.0460\ \text{ft}^2$$

and

$$Re = \frac{(0.25)\ (23{,}600)}{(12)\ (0.0460)\ (4.50)} = 2380$$

2. From Fig. 2, by interpolation between curves 1 and 4,

$$j = 0.0170 = \left(\frac{h_{NL}\,\chi_t}{\phi\, \xi_h\, c\, G_m}\right)\left(\frac{c\mu}{k}\right)^{2/3}\left(\frac{\mu_w}{\mu}\right)^{0.14}$$

TABLE 4. Exchanger 10-TL3: Exchanger Dimensions.

Tube array	Equilateral triangle
Tube-baffle clearances, on diameter, in.	0.013
Shell-baffle clearances, on diameter, in.	0.080
OD baffles, in.	8.422
ID shell, in.	8.502
Baffle edge, in.	6.50
Baffle cut-down, %	17.56
Baffle hole diameter, in.	0.2623
Baffle spacing, in.	2.643
Baffle thickness, in.	0.0625
Number of baffles	5
No. tubes in each window	46
No. tubes plus spacers in window	49
No. tubes in crossflow	378
OD tubes, in.	0.2493
Effective tube length, in.	16.125
Distance between tube centers, in.	0.34375
Pitch	1.33
Distance between tubes, in.	0.0944

3. Evaluate F_{bp} at the 24-tube rows: The diametral clearance between the outer tubes and the shell is
$8.502 - 23\ (1.33)\ (0.249) - 0.249 = 0.64$ in.

Then $F_{bp} = \dfrac{0.64\ (2.643)}{144\ (0.0460)} = 0.255$

4. From Equation 2
$\xi_h = e^{-1.25\ (0.255)} = 0.728$

5. To an excellent approximation, r is the ratio of the number of tubes in both upper and lower windows (92) divided by the total number of tubes (470), i.e., $r = 0.196$. From Fig. 4, $\phi = 1.11$.

6. Skip, since $Re > 100$.

7. There are 18 rows in each crossflow section; the j factor plot (Fig. 2) is based on 10 rows. From Table 2, $\chi_t = \dfrac{0.96}{0.93} = 1.03$.

8. For Gulf 896 at 149.5 F, c = 0.526 Btu per lb F and
$\left(\dfrac{c\mu}{k}\right)^{2/3} = 9.55.$

Also estimate $\left(\dfrac{\mu_w}{\mu}\right)^{0.14} = 0.97$; in principle, this quantity must be found by trial-and-error. Then

$$h_{NL} = \frac{0.0170(1.11)(0.728)(0.526)(23{,}600)}{1.03\qquad (0.0460)(9.55)(0.97)}$$
$$= 389\ \text{Btu/(hr)(ft}^2\text{)(F)}$$

9. Calculate S_{tb} observing that each baffle has $(378 + 46) = 424$ tubes.

Then $S_{tb} = \dfrac{424}{144}\left(\dfrac{\pi}{4}\right)(0.2623^2 - 0.2493^2) =$ 0.0154 ft².

Calculate S_{sb} assuming that the baffle is concentric to the shell. The angle subtended by the baffle cut is

$2\ \text{Sin}^{-1}\dfrac{6.360}{8.422} = 98°.$

Then

$$S_{sb} = \left(\frac{360\text{-}98}{360}\right)\frac{\pi}{4}\left(\frac{8.502^2 - 8.422^2}{144}\right) = 0.00539\ \text{ft}^2$$

$$S_L = S_{tb} + S_{sb} = 0.0208$$

$$\frac{S_{tb} + 2S_{sb}}{S_L} = 1.26;$$

10.

$$\frac{S_L}{S_m} = \frac{0.0208}{0.0460} = 0.453$$

From Fig. 8a, $\left(1 - \frac{h_L}{h_{NL}}\right)_0 = 0.30$

and from Equation 15 $\left(1 - \frac{h_L}{h_{NL}}\right)_{Exchanger}$
$= 0.30(1.26) = 0.378$

11. Then $\frac{h_L}{h_{NL}} = 0.622$ and finally

$$h_L = 0.622(389) = 242 \; \frac{Btu}{(hr)(ft^2)(F)}$$

12. From Equation 2

$$\xi_{ap,t} = e^{-3.8(0.255)} = 0.380$$

13. From Fig. 2 by interpolation,

$$f = 0.17 = \left[\frac{2\ \Delta p_{BP}\ g_c \rho}{4\ \xi_{ap,t} G_m^2}\right]\left(\frac{\mu}{\mu_s}\right)^{0.14}$$

or

$$\Delta p_{BP} = \frac{0.17\ (4)(0.380)(23{,}600)^2\ (18)(0.97)}{2\ (32.2)\ (3600)^2\ (0.0460)^2\ (49.0)}$$

or

$$\Delta p_{BP} = 29.1 \frac{lbf}{ft^2}$$

$$\left(\rho \text{ for Gulf 896 at 149.5F} = 49.0 \frac{lbm}{ft^3}\right) \leftarrow$$

14. The window flow area S_w' may be calculated as

$$S_w' = \frac{\pi}{144}\left(\frac{8.502}{2}\right)^2 \left(\frac{98}{360}\right) - \left(\frac{8.502}{2}\right)^2 \frac{(\sin 98°)}{(2)(144)}$$

$$-49\left(\frac{\pi}{4}\right)\left(\frac{0.249}{144}\right)^2 = 0.0289\ ft^2$$

$$V_w = \frac{23{,}600}{0.0289\ (49.0)\ (3600)} = 4.65 \text{ ft per sec}$$

$$V_m = \frac{23{,}600}{0.0460\ (49.0)\ (3600)} = 2.91 \text{ ft per sec}$$

$$V_z = \sqrt{4.65\ (2.91)} = 3.68 \text{ ft per sec}$$

15. Skip, since Re >100.

16. From a tube layout, it is found that there are five tube rows between the baffle cut and the shell. The outer two, however, have less than half the tubes of the centerline rows (4 and 11 respectively) so $N_w = 3$. From Equation 4

$$P_{w,t} = [2 + 0.6(3)]\ \frac{(49)\ (3.68)^2}{2(32.2)}$$

$$= 39.2 \frac{lbf}{ft^2}$$

17. Because the two end sections will have substantially higher pressure drops than the internal sections and there are only five baffles, it is desirable to modify the last three steps. Treat the two end crossflow sections as non-leakage sections. Then the pressure drop in each end crossflow section is $\left(1 + \frac{N_w}{N_c}\right)\ \Delta p_{BP}$ or, $1.167\ (29.1) = 34.0\ lbf/ft^2$. The non-leakage pressure drop for the remainder of the exchanger is

$$4(29.1) + 5(39.2) = 312.4\ lbf/ft^2$$

18. From Fig. 8b at $S_L/S_m = 0.453$,

$$\left(1 - \frac{\Delta p_L}{\Delta p_{NL}}\right)_0 = 0.51$$

and

$$\left(1 - \frac{\Delta p}{\Delta p_{NL}}\right)_{Exchanger} = 0.51(1.26) = 0.643$$

19. Then the pressure drop for the entire exchanger, Δp_L is $2(34.0) + 0.357(312.4) = 176\ lbf/ft^2$ or $\Delta p_L = 1.22\ lbf/in.^2$

Experimental results gave $h_L = 264$ Btu/(hr)(ft²)(F) compared to the calculated 242, an error of 8.3%, and $\Delta p_L = 1.30 \frac{lbf}{in.^2}$ compared to the calculated 1.22, an error of 6.2%.

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Nomenclature

The units shown are consistent; any other consistent set will do as well.

A = Total shell-side area for heat transfer, ft².

A_B = Shell-side heat transfer area of tubes in one crossflow section, ft².

A_w = Shell-side heat transfer area of tubes in one window section, ft².

b = Coefficient in Equation 6, (Btu) (sec$^{0.6}$)/(hr) (ft$^{2.6}$) (F).

c = Specific heat of shell-side fluid, $\frac{Btu}{lbm\ F}$

D_t = Tube diameter, ft.

D_v = Volumetric equivalent diameter of the window, ft.

$$D_v = \frac{4\ S_w L_B}{A_w}$$

F_{BP} = Fraction of total minimum crossflow area that is in the bypass channel between bundle and shell, dimensionless.

f = Crossflow friction factor for ideal tube bank defined by

$$f = \frac{2\ \Delta p_B\ g_c\ \rho}{4\ G_m^2\ N_c}\left(\frac{\mu}{\mu_s}\right)^{0.14} \text{ dimensionless.}$$

A slightly different definition involving bypass correction is used and indicated in the correlation.

G_m = Mass velocity based on S_m, lb per sec ft².

g_c = Gravitational conversion constant, $32.2 \frac{\text{lbm ft}}{\text{lbf hr}^2}$

h = Heat transfer coefficient, Btu per hr ft² F; h_B, coefficient for crossflow section or ideal tube bank, no bypass or leakage; h_{BP}, coefficient for crossflow section or overall coefficient for the exchanger, after bypass correction, no leakage. h_L, overall coefficient for the exchanger with bypass and leakage; h_m, mean coefficient for tube bank of finite number of rows; h_{NL}, overall coefficient for the exchanger with no bypass or leakage; h_w, coefficient for a window section, no bypass or leakage; h_∞, coefficient for an ideal tube bank with an infinite number of tube rows, turbulent flow.

I = Value of h_B at $V_m = 1$ ft per sec, $\frac{\text{Btu}}{(\text{hr})(\text{ft}^2)(\text{F})}$

j = Colburn heat transfer factor defined by

$$j = \left(\frac{h_B}{c G_m}\right)\left(\frac{c\mu}{k}\right)^{2/3}\left(\frac{\mu_s}{\mu}\right)^{0.14}$$

for ideal tube bank dimensionless.

Slightly different definitions involving exchanger geometry corrections are used and indicated in the correlations.

k = Thermal conductivity of shell-side fluid, $\frac{\text{Btu}}{(\text{hr})(\text{ft}^2)(\text{F/ft})}$

L_B = Baffle spacing, ft.

m = Empirical exponent, dimensionless.

n = Empirical exponent, dimensionless.

N_B = Number of baffles.

N_c = Number of major constrictions encountered during flow through one crossflow section. N_c is the number of tube rows between baffle tips in inline square or equilateral triangle arrays and one less than the number of tube rows between baffle tips in staggered square arrays.

N'_c = Total effective number of major constrictions encountered during flow through the entire shell side of a baffled exchanger.

N_s = Number of sealing strips encountered during flow across one crossflow section.

N_w = Effective number of restrictions for crossflow in window. N_w is approximately the number of tube rows between the baffle tip and the shell.

P = Tube pitch, the minimum distance between centers of adjacent tubes, ft.

Δp = Pressure drop, lbf/ft². Δp_B, pressure drop for crossflow section or ideal tube bank, no bypass or leakage; Δp_{BP}, pressure drop for crossflow section or entire exchanger, after bypass correction, no leakage; Δp_L, overall pressure drop for the exchanger with bypass and leakage; Δp_{NL}, overall pressure drop for the exchanger with no bypass or leakage; Δp_w, pressure drop for a window section, no bypass or leakage.

r = Ratio of shell-side heat transfer area in the windows to total heat transfer area $r = A_w/A$, dimensionless.

Re = Reynolds number for crossflow, $Re = \frac{D_t G_m}{\mu}$ dimensionless.

s = Minimum spacing between adjacent tubes, $s = P - D_t$, ft.

S_m = Minimum area for crossflow near centerline of exchanger, ft².

S_w = Area for flow through baffle cutdown, ft².

V_m = Crossflow velocity based on S_m, ft per sec.

V_w = Window velocity based on S_w, ft per sec.

V_z = Geometric mean velocity, $V_z = \sqrt{V_m V_w}$, ft per sec.

α = Empirical constant in Equation 2.

β = Coefficient in Equation 7, (Btu)(sec)/(hr)(ft$^{2.6}$)(F).

ξ = Correction for bypass flow effects, dimensionless; $\xi_h = \frac{h_{BP}}{h_{NBP}}$, correction for heat transfer coefficient; $\xi_{\Delta p,t} = \frac{\Delta p_{BP}}{\Delta p_{NBP}}$, correction for crossflow pressure drop in turbulent regime; $\xi_{\Delta p,l} = \frac{\Delta p_{BP}}{\Delta p_{NBP}}$, correction for crossflow pressure drop in laminar regime.

μ = Absolute viscosity, lbm per ft sec.

μ_s = Absolute viscosity at heat transfer surface, lbm per ft sec.

ρ = Density of shell-side fluid, lbm per ft³.

ϕ = Correction factor on ideal crossflow heat transfer coefficient for window effects, dimensionless.

χ = Correction factor on ideal crossflow heat transfer coefficient for number of tube rows, dimensionless. χ_t — correction factor for turbulent regime from Table 2; χ_v — correction factor for very low Reynolds numbers from Equation 14.

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