2.5. Preliminary Design of Shell and Tube Heat Exchangers

2.5.1 Basic Principles of Design

1. *Design Criteria for Process Heat Exchangers*. The criteria that a process heat exchanger must satisfy are easily enough stated if we confine ourselves to rather broad statements:

First, of course, the heat exchanger must meet the process requirements. That is, it must effect the desired change in the thermal condition of the process stream within the allowable pressure drops, and it must continue to do this until the next scheduled shutdown of the plant for maintenance.

Second, the exchanger must withstand the service conditions of the plant environment. This includes the mechanical stresses of installation, startup, shutdown, normal operation, emergencies, and maintenance, and the thermal stresses induced by the temperature differences. It must also resist corrosion by the process and service streams (as well as by the environment); this is usually mainly a matter of choice of materials of construction, but mechanical design does have some effect. Desirably, the exchanger should also resist fouling, but there is not much the designer may do with confidence in this regard except keep the velocities as high as pressure drop and vibration limits permit.

Third, the exchanger must be maintainable, which usually implies choosing a configuration that permits cleaning - tube-side and/or shell-side, as may be indicated – and replacement of tubes and any other components that may be especially vulnerable to corrosion, erosion, or vibration. This requirement may also place limitations on positioning the exchanger and in providing clear space around it.

Fourth, the exchanger should cost as little as is consistent with the above requirements; in the present context, this refers to first cost or installed cost, since operating cost and the cost of lost production due to exchanger unavailability have already been considered by implication in the earlier and more important criteria.

Finally, there may be limitations on exchanger diameter, length, weight and/or tube specifications due to site requirements, lifting and servicing capabilities, or inventory considerations.

It is sometimes stated as a desirable feature that the exchanger design be specified with an eye to possible alternative uses in other applications. However, this has disturbing implications. Most heat exchangers are intended for projects having an expected life of five to twenty years - equal to or greater than the probable life of the exchanger. To suggest that a heat exchanger might become available sooner implies either that the exchanger or the process will prove unsatisfactory in its role. It is far better to labor under the positive compulsion that the only hope for success is by designing each item uniquely for the best performance in the task at hand.

2. *Structure of the Heat Exchanger Design Problem.* The basic logical structure of the process heat exchanger design procedure is shown in Fig. 2.21. The basic structure is the same whether we use hand design methods or computer design.

First, the problem must be identified - as completely and unambiguously as possible. This includes data like flow rates, pressures, temperatures and compositions, and it also includes qualitative information such as the likelihood of fouling and the difficulty of cleaning, special materials requirements, and any unusual conditions to be encountered during operation.

It is at this point in the design process that the single most important design decision is made: the basic configuration of the heat exchanger, whether it is to be double pipe, shell-and-tube, plate, etc.

The next step is to select a tentative set of major parameters for the exchanger: tube type, size, layout, shell diameter and length, baffle spacing, etc. A procedure for doing this will be given later in this chapter.

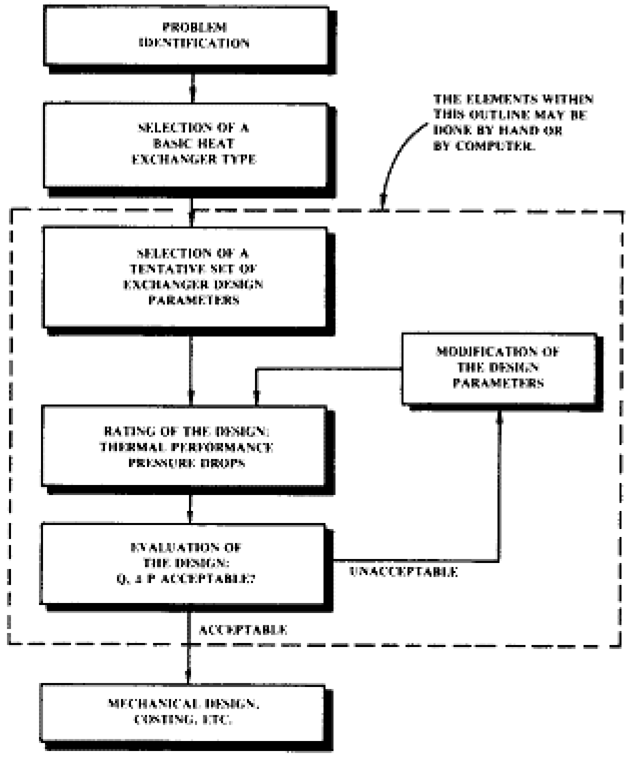

Fig. 2.21 Basic Logical Structure of Process Heat Exchanger Design Procedure.

Next, the thermal performance for the tentative configuration is rated using the procedure given later in this Chapter. That is, the overall heat transfer coefficient is calculated for the given flow rates in the design chosen; that value is combined with the required heat duty and the calculated value of the mean temperature difference to determine the heat exchanger area required. Finally, this value is compared to the area available in the chosen design. If the calculated area is reasonably close to the available area, the heat exchanger is acceptable from a thermal point of view and one may go on to the pressure drop calculation. However, if the areas do not correspond, it is necessary to adjust the tentative configuration parameters to increase or decrease the heat transfer area as required, and then the new configuration is re-rated. In the pressure drop calculation, the pressure drop of each stream must be less than - but not greatly less than – the allowable values. If the calculated pressure drop is much less than that allowed, it will probably prove possible to reduce the size of the exchanger.

Once the thermal performance requirement and the pressure drop limitations are satisfied, one can go on to do a mechanical design and cost estimation. These stages are not included in this manual.

2.5.2. Preliminary Design Decisions

1. *Allocation of Streams*. Having selected a shell and tube exchanger, the designer's next decision is to decide which stream goes into the tubes and which into the shell. There are several different circumstances, which control this decision, among which the following are the most important:

a. Possibility of using extended surface tubes. As we have repeatedly stressed, the use of Trufin tubes is generally advantageous when one fluid has a coefficient significantly lower than the other. In these cases, the fluid with the low coefficient will be allocated to the shell side.

b. One fluid is highly corrosive. The solution to handling a corrosive fluid is to use an alloy which is resistant to the fluid. Since all corrosion resisting alloys are relatively expensive, it follows that the corrosive substance should be put in the tubes. Then only the tubes, the tube sheets (often simply faced with the alloy), and the tube side channels and piping need to be made of the alloy. The shell, baffles, tie rods, etc., can be made of low carbon steel, at a considerable cost savings.

c. One fluid is at high pressure. When one fluid is at a much higher pressure than the other, it should be placed in the tubes. Only the tube-side components have to be constructed to resist the high pressure, and particularly it is not necessary to use a heavy and expensive high pressure shell.

d. One fluid is severely fouling. Since it is easier to clean the tube-side than the shell-side by mechanical methods (brushes, water jets, etc.), the more severely fouling fluid should be put on the tube-side.

f. If the allowable pressure drop for one stream is very limited, that stream should normally go on the shell-side. Even though the "efficiency of conversion" of pressure drop into heat transfer coefficient is not as high on the shell-side (because of form drag), there is usually so much flexibility in the selection of design parameters on the shell side that is easier to accommodate the low pressure drop limitation.

Clearly, there will frequently be conflicts between these "rules"; for example, one fluid may be at high pressure and the other one may be severely corrosive. The decision in this case may have to be delayed until complete designs have been prepared and priced for both cases. In general, the lower cost option would be selected, but safety considerations or differences in the reliability of operation might overrule a purely economic decision.

2. *Selection of Shell Type*. The most important factor in selecting a shell type is the thermal stress problem. As discussed in Chapter 1, thermal stresses arise because the tubes are at a different average temperature than the shell, and the differences in thermal expansion can result in a number of catastrophic events: the tubes may be pulled out of the tube sheets, or the tubes may be pulled apart or buckled, or the shell may buckle, or the tube sheets may be deformed enough to open up leaks through the gaskets.

The determination of whether or not a serious thermal stress problem exists is a complex calculation and only a few rules of thumb can be given here:

a. Fixed tube sheet exchangers having no specific arrangement to relieve or avoid thermal stresses can only be used when the difference in inlet temperatures of the two streams is less than about 100 ºF.

b. Fixed tube sheet exchangers with rolled expansion joints in the shell can be used for inlet temperature differences of up to about 200 ºF for moderate pressure shells (say on the order of 150 psia). Expansion joints have been used too much higher inlet temperature differences for low pressure (i.e., thin-walled) shells.

c. Where they can be used, U-tube bundles represent an essentially complete solution to the tube stress problem because each tube is free to expand or contract independently of the shell and, within wide limits, the other tubes. However, tube sheet thermal stress problems are not completely solved and must be further analyzed.

The relative advantages and disadvantages of the various floating head shell designs were discussed in Chapter I and will not be further treated. However, most of the floating head designs have multiple tubes passes and require a configuration correction factor on the LMTD.

3. *Selection of Shell Arrangement.* It is often necessary to use more than one exchanger to accomplish a given service. The two basic arrangements are exchangers in parallel (Fig. 2.22) and exchangers in series (Fig. 2.23):

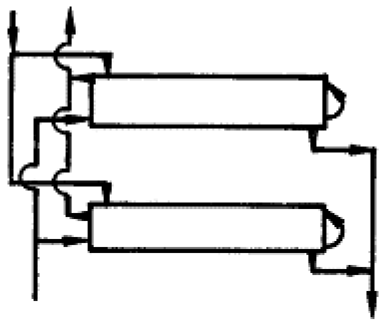

Fig. 2.22 Parallel Arrangement of Two Identical Exchangers

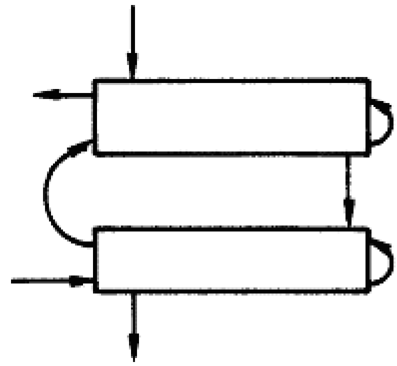

Fig. 2.23 Series Arrangement of Two Identical Exchangers

The parallel arrangement is mainly used when pressure drop limitations (coupled with length, diameter and baffle spacing limits) force a reduction in shell-side velocity and thus throughput per unit. For identical units, each exchanger may be separately analyzed using its proportional share of the flow rates.

The purely series arrangement is mainly useful when

a. The single shell with multiple tube passes gives too low a value for F, as previously discussed in the Mean Temperature Difference Concept,

b. There are limitations on shell length and/or diameter, requiring the total area to be disposed in more than one shell.

The shells are usually identical for economy in manufacturing and ease and flexibility of installation, operation and maintenance.

An infinite variety of series-parallel arrangements is possible. The most common is several exchanger trains in parallel (to split the flows down to rates that can be conveniently handled in the maximum acceptable exchanger size), with each train composed of several exchangers in series (to improve the Mean Temperature Difference). A single train can be analyzed using its proportion of the total flow and the total service determined by multiplying by the number of trains in parallel.

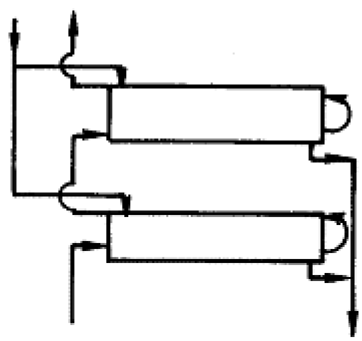

Fig. 2.24 A Plausible Series-Parallel Arrangement of Two Exchangers

An example of a more unusual series-parallel arrangement is shown in Fig. 2.24. Such an arrangement might be considered if the shell-side fluid had a severe pressure drop limitation (forcing parallel flow) and needed to be changed in temperature only over a narrow range, while the tube-side fluid had only a low flow rate, thereby encouraging many tube-side passes to keep the velocity up. It should be noted that the two shells will have different outlet temperatures and hence there will be some loss of efficiency when the two effluent streams are mixed. In general, a trial and error solution of the heat balance and rate equations for each exchanger is required in mixed series-parallel arrangements, easily enough accomplished by a computer program, but rather tedious by hand for any significant number of shells.

2.5.3. Procedure for Approximate Size Estimation

1. *Calculation of Q.* For the usual design case, sufficient data are given or can be chosen to calculate Q and MTD.

For sensible heat transfer on the shell-side, Q is calculated by

$$Q = W_s C_{ps} \Delta T \tag{2.32}$$

or for the tube-side by

$$Q = w_i c_{pi} (\Delta t) \tag{2.33}$$

2. *Calculation of MTD.* The logarithmic mean temperature difference (LMTD) can be readily calculated from the terminal temperature differences by Eq. (2.11). Alternatively, for preliminary design, the LMTD can be estimated within a few percent; the LMTD is always smaller than the arithmetic mean, the difference being roughly proportional to the ratio of the larger terminal temperature difference to the smaller.

The value of F can also be calculated from the terminal temperatures, using Figs. 2.5 to 2.12, depending upon the configuration. Or, for preliminary design of multiple tube pass designs, F may be estimated as 0.9, which is the average between the maximum possible value, 1.0, and the minimum recommended value, 0.8. This value may be shaded higher if the ratio of the terminal temperature differences is near unity and lower if the outlet stream temperatures are similar. In the latter case - and more especially if there is a temperature cross - the thermodynamic feasibility of the design should be checked before proceeding further. An absolute limit that may be quickly checked is

$$t_2 \le 2T_2 - t_1 \text{ for tube-side heating} \tag{2.34}$$

$$t_2 \ge 2T_2 - t_1 \text{ for tube-side cooling} \tag{2.35}$$

where t_2 is the outlet temperature on the tube-side, t_1 the inlet temperature on the tube-side, and T_2 the outlet temperature on the shell-side.

When it is required to use multiple shells in series (as when two fluids are specified to exchange heat over a wide temperature range in a feed-effluent exchanger), there is a rapid graphical technique for estimating a sufficient number of shells in series. The procedure is shown in Fig. 2.25 and goes as follows:

a. The terminal temperatures of the two streams are plotted on the ordinates of ordinary arithmetic graph paper, the hot fluid inlet temperature and the cold fluid outlet temperature on the left hand ordinate and the hot fluid outlet and cold fluid inlet temperature on the right hand ordinate. The distance between the ordinates is arbitrary, (corresponding to the total amount of heat exchanged between the two streams) and may be chosen to the convenience of the user.

b. If the specific heat of each stream is constant, straight lines ("operating lines") are drawn from the inlet to the outlet temperature point for each stream.

If the specific heat of one or both streams varies, it is necessary to calculate the temperature of that stream as a function of the amount of heat added or removed by the other, resulting in one or both operating lines being curved. In this case, the procedure given here for finding a sufficient number of shells in series will still be valid, but the Mean Temperature Difference concept will not be. The exact design procedures to be followed in this case are beyond the scope of this Manual.

c. Starting with the cold fluid outlet temperature (275°F in Fig. 2.25), a horizontal line is laid off until it intercepts the hot fluid line. From that point a vertical line is dropped to the cold fluid line (reaching it in our example at a cold fluid temperature of 228°F). This procedure defines a heat exchanger operation in which the hot fluid temperature is never less than any temperature reached by the cold; that is, there is no temperature cross and we know there is no thermodynamic difficulty that can arise if this operation is carried out in one shell.

d. The process is repeated until a vertical line intercepts the cold fluid operating line at or below the cold fluid inlet temperature. (Alternatively, the process is continued until a horizontal line crosses the right hand ordinate.)

e. The number of horizontal lines (including the one that intersects the right hand ordinate) is equal to the number of shells in series that is clearly sufficient to perform the duty. In the case of our example problem, this number is three.

We may do some calculations that give some quantitative feel for the present case:

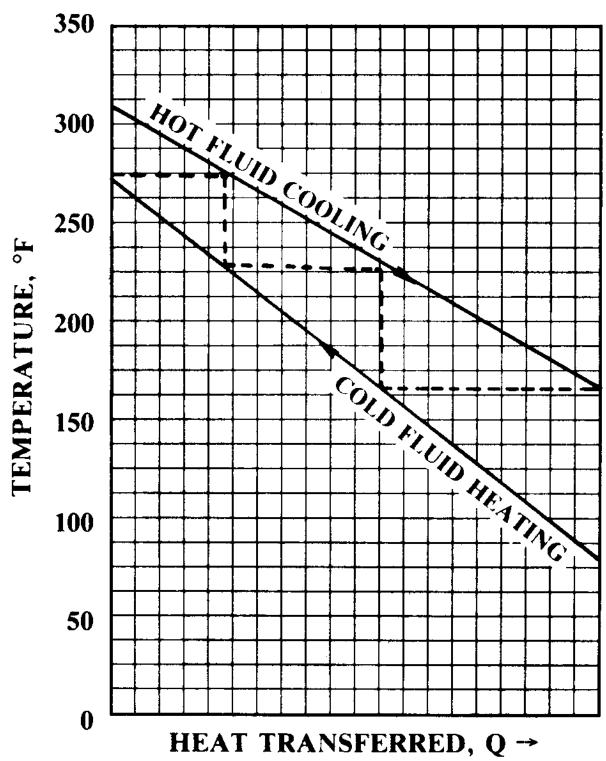

Fig. 2.25 Estimation of Number of Shells Required in Series. Example: Hot stream at 310°F to be cooled to 165°F using a cold stream to be heated from 80°F to 275°F.

The overall LMTD is:

$$LMTD = \frac{(310-275)-(165-80)}{\ell n\left(\frac{310-275}{165-80}\right)} = 56.4°F \qquad (2.11)$$

and

$$P = \frac{275-80}{310-80} = 0.848 \qquad (2.14)$$

$$P = \frac{310-165}{275-80} = 0.744 \qquad (2.13)$$

From Fig. 2.7 for 3 shells in series

F=0.81

This is a very acceptable value. For four shells in series, F = 0.91, and for two shells in series, F < 0.5 (in fact, this configuration is probably thermodynamically unworkable). Whether there would be any advantage in going to four shells rather than three would depend upon the particular circumstances of the case.

3. *Estimation of* U_o. The step with the greatest uncertainty in preliminary calculations is estimating the overall heat transfer coefficient. Tables of U's typical of various services are widespread; the drawback to their use is that, in trying to include the entire range ever encountered in practice, these tables give a spread of values so great as to be almost meaningless for more or less optimal design. A table of U_o for various services is included here as Table 2. 1. The values given are for U_o for Trufin tubes, with the controlling resistance on the finned side.

A better procedure is to build up the value of U_o from the individual h values, and wall, fin and fouling resistances using Eq. (2.2). It will be generally found that one or at most two terms will dominate the value of U, and attention can be focused upon these controlling values. It will generally be found that the range of reasonable values is far smaller than the range of possible values. It will also prove useful to make some estimate of the basic uncertainties in each value, i.e., the uncertainty in the value that would still exist even if the best available correlation and physical properties were used. This will often indicate rather clearly the futility of worrying too much about the precise value of a coefficient for preliminary design purposes.

4. Estimation of A_o and Key Exchanger Parameters. Once Q, MTD, and U_o are known, the total outside heat transfer area (including fin area) A_o is readily found from

$$A_o = \frac{Q}{U_o(MTD)} \qquad (2.36)$$

The next question is, "What set of heat exchanger dimensions will accommodate the heat transfer area?"

Fig. 2.26 is an aid in answering that question. In this figure the ordinate is A_o in ft^2, and the abscissa is the effective tube length (tube sheet to tube sheet for straight tube bundles, or length of a single straight section from tube sheet to tangent line for a U-tube bundle) in ft. The solid black lines in parameter are the commonly specified shell inside diameters in inches. Using standard tube count tables for 3/4 in.

O.D., type S/T Trufin, 19 fins/in. tubes in a 15/16 in. triangular pitch for a fixed tube sheet heat exchanger with one tube pass, the total outside tube heat transfer area that can be fitted into a shell has been calculated.

Therefore, once the required area A_o is known, Fig. 2.26 shows immediately the combinations of tube length and shell diameter that will provide that area in a single shell for an exchanger of the given tube size and layout.

The dashed lines shown in parameter and marked 3:1, 6: 1, etc, are ratios of tube length to shell inside diameter. These lines are included to give a rapid "feet" for the proper proportions of the exchanger. An exchanger with a length to diameter ratio too short - say, less than 3:1 - is likely to suffer from poor distribution of the streams and an excessive cost because of the large shell diameter.

An exchanger with an excessively large length to diameter ratio is likely to be difficult to handle mechanically (the bundle must be very carefully supported when pulled, because of its springiness) and requires a wide clearway for pulling the bundle and retubing. An arbitrary ratio of 15:1 has been assigned to the probable upper limit. Most heat exchangers fall into the 6:1 to 8:1 range, though there has been a pronounced trend towards higher values as pressure drop estimation procedures have improved.

5. *Generalization of Fig. 2.26.* The usefulness of Fig. 2.26 can be greatly extended by defining an effective area A' by the equation

$$A'_o = A_o F_1 F_2 F_3 F_4 \qquad (2.37)$$

where A'_o is the area on the ordinate of Fig. 2.26

A_o is the outside area of a finned tube heat exchanger as calculated from Eq. (2.36)

F_1 is the correction factor for the unit cell tube array (= 1.00 for 3/4 in. tubes on a 15/16 in. triangular pitch).

F_2 is the correction factor for the number of tube passes (= 1.00 for one tube pass).

F_3 is the correction factor for the shell construction/tube bundle layout type (= 1.00 for fixed tube sheet).

F_4 is the correction factor for the specific fin geometry and density (= 1.00 for S/T Trufin, 19 fins/in.)

Tables for the various correction factors are given at the end of this chapter.

6. *Application of the Preliminary Design Procedure*. The use of the estimation method given in the preceding chapter is best illustrated by an example:

Estimate the approximate size of an air compressor intercooler required to cool 13,000 SCFM (58,500 lb/hr) of air at 65 psig from $350°F$ to $125°F$, using water at $80°F$. The tubes are to be type S/T Trufin 3/4 in. O.D., 26 fins/in., of phosphorus deoxidized copper, and a U-tube bundle is desired. A tube layout on a 1 in. triangular pitch is specified.

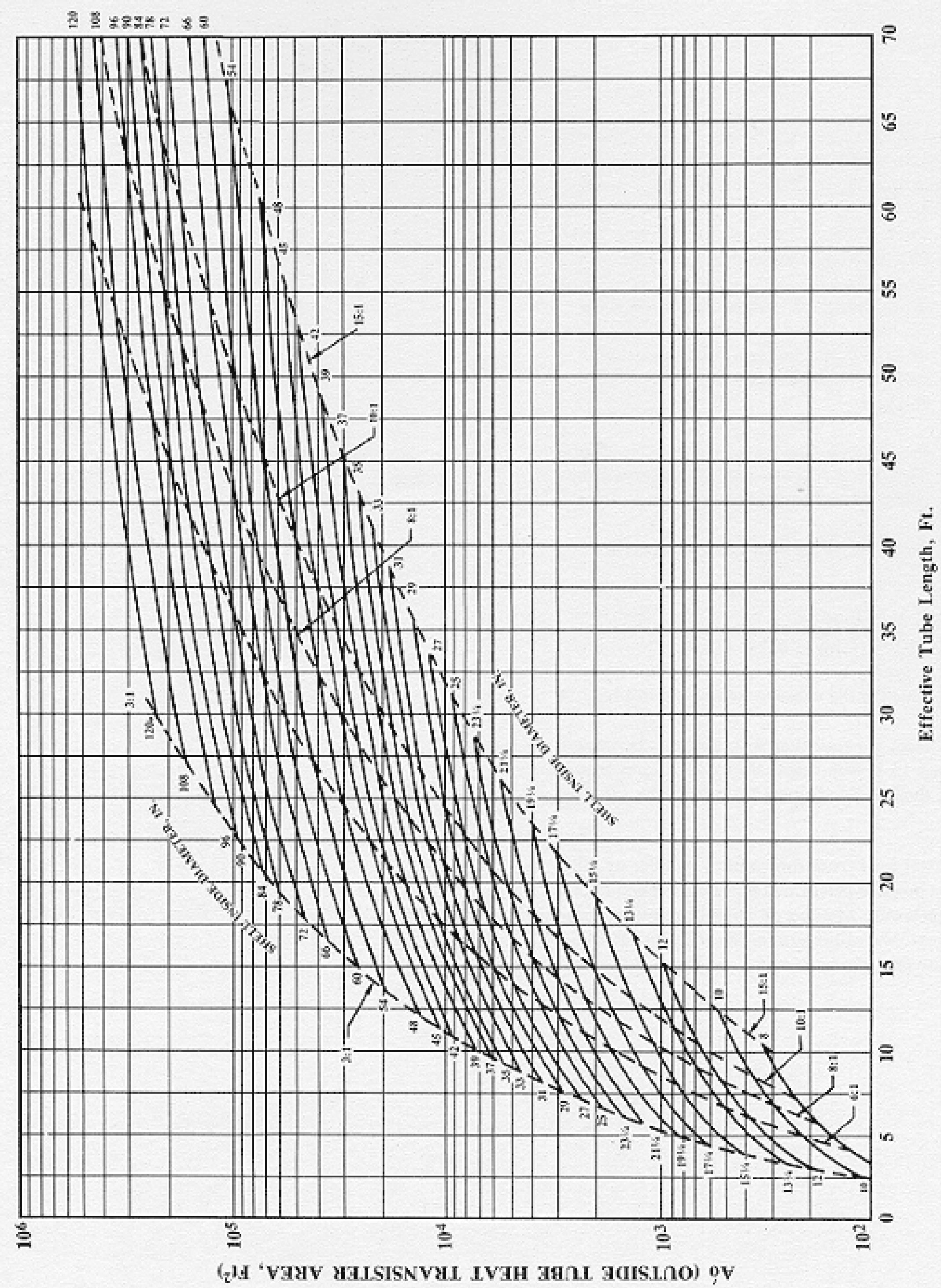

Fig. 2.26 Area as a Function of Shell Inside Diameter and Effective Tube Length for 3/4 in. O.D. S/T Trufin 19 Fins/In. on 15/16 in. Equilateral Triangular Tube Layout Fixed Tube Sheet, One Tubeside Pass, Fully Tubed Shell.

First, calculate the heat load:

$$Q = (58{,}500 \;\; lb/hr)\left(0.241 \frac{Btu}{lb°F}\right)(350-125)°F = 3{,}172{,}000 \;\; Btu/hr \qquad (2.32)$$

Next, calculate the mean temperature difference; assuming an outlet water temperature of $110°F$:

$$LMTD = \frac{(350-110)-(125-80)}{\ell n\left(\dfrac{350-110}{125-80}\right)} = 116.5°F \qquad (2.11)$$

$$P = \frac{110-80}{350-80} = 0.111 \qquad (2.14)$$

$$R = \frac{350-125}{110-80} = 7.5 \qquad (2.13)$$

F = 0.91 (from Fig.2.5)

MTD = 0.91 (116.5 °F) = 106 °F (2.10)

From Table 2.1, we estimate U_o to be about 25 Btu/hr ft^2 °F.

Then

$$(A_o)_{\mathrm{Re}q'd} = \frac{3{,}172{,}000}{(25)(106)} = 1197\,ft^2 \qquad (2.36)$$

Before entering Fig. 2.26, it is necessary to correct this area to the same basis as Fig. 2.26, by finding the proper factors to put into Eq. (2.37).

$F_1 = 1.14$ From Table 2.2, noting that a 3/4 in. O.D. by 1 in triangular pitch is specified

$F_2 = 1.06$ From Table 2.3, assuming that the shell diameter will be in the range of 13 1/4 in. to 17 1/4 in., and two tube passes will suffice. These assumptions have to be checked.

$F_3 = 1.08$ From Table 2.4, correcting for the U-tube design and that the shell diameter will be in the 13 1/4 to 21 1/4 inch range

$F_4 = 0.79$ From Table 2.5, for the tube specified.

Then

$$A'_o = (1197\ ft^2)(1.14)(1.06)(1.08)(0.79) = 1234\ ft^2 \quad (2.37)$$

which is the value to be entered in Fig. 2.26

From Fig. 2.26, we find that this area can be accommodated in a 15 1/4 in. i.d. shell, by 12 ft long or a 17 1/4 in. i.d. shell, by 10 ft. long, or a 19 1/4 in. i.d. shell 8 ft. long.

In order to proceed from this point a number of other calculations must be performed but these depend in part upon the Delaware Method of Shell-side Rating to be discussed next. We will return to this problem for a final design at the end of this chapter.