



# **Selection and costing of heat exchangers Shell-and-tube type**

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## SELECTION AND COSTING OF HEAT EXCHANGERS SHELL-AND-TUBE TYPE

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## SELECTION AND COSTING OF HEAT EXCHANGERS

### Shell-and-tube type

#### 1. NOTATION AND UNITS

|              |                                                           | <i>Units</i>   |                 |
|--------------|-----------------------------------------------------------|----------------|-----------------|
|              |                                                           | <i>SI</i>      | <i>British</i>  |
| $a$          | constant in Equation (7.1)                                | —              | —               |
| $A$          | heat transfer surface area of heat exchanger              | m <sup>2</sup> | ft <sup>2</sup> |
| $b$          | constant in Equation (7.1)                                | —              | —               |
| $c_p$        | specific heat capacity                                    | J/kg K         | Btu/lb°F        |
| $C$          | cost per unit $\dot{Q}/\Delta T_m$                        | £/(W/K)        | £/(Btu/°F)      |
| $CR$         | cost ratio                                                | —              | —               |
| $CR_{L/D}$   | cost ratio for change in $L/D$ ratio                      | —              | —               |
| $CR_p$       | cost ratio for change in pressure                         | —              | —               |
| $CR_s$       | cost ratio for change in shell material                   | —              | —               |
| $CR_{shell}$ | cost ratio for change in shell type                       | —              | —               |
| $CR_t$       | cost ratio for change in tube material                    | —              | —               |
| $CR_{tube}$  | cost ratio for change in tube diameter                    | —              | —               |
| $D$          | shell diameter                                            | m              | ft              |
| $D_t$        | tube diameter                                             | m              | ft              |
| $E$          | heat exchanger effectiveness, defined by Equation (4.7)   | —              | —               |
| $F_T$        | logarithmic mean temperature difference correction factor | —              | —               |
| $h$          | specific enthalpy                                         | J/kg           | Btu/lb          |
| $L$          | shell length                                              |                |                 |
| $L_{b, max}$ | maximum baffle spacing                                    | m              | ft              |
| $\dot{M}$    | mass rate of flow                                         | kg/s           | lb/h            |

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|            |                                                         |                    |                          |
|------------|---------------------------------------------------------|--------------------|--------------------------|
| $N_{TU}$   | number of transfer units, defined by Equation (4.5)     | –                  | –                        |
| $P$        | flow heat capacity ratio defined by Equation (9.5)      | –                  | –                        |
| $\dot{Q}$  | heat load of exchanger                                  | W                  | Btu/h                    |
| $C^*$      | flow heat capacity ratio defined by Equation (4.8)      | –                  | –                        |
| $R_K$      | flow heat capacity ratio defined by Equation (9.4)      | –                  | –                        |
| $S$        | number of shells                                        | –                  | –                        |
| $T$        | stream temperature                                      | K or °C            | °F                       |
| $T_i$      | interpass temperature within exchanger, see Section 4.2 | K or °C            | °F                       |
| $U$        | overall heat transfer coefficient                       | W/m <sup>2</sup> K | Btu/h ft <sup>2</sup> °F |
| $\Delta T$ | temperature difference                                  | K or °C            | °F                       |
| $\eta$     | parameter defined by Equation (4.10)                    | –                  | –                        |

## Subscripts

|                   |                                                                                      |
|-------------------|--------------------------------------------------------------------------------------|
| $c, h$            | refer to cold and hot stream respectively                                            |
| $cs$              | carbon steel of standard configuration                                               |
| $in, out$         | refer to values of parameter at inlet to and outlet from heat exchanger respectively |
| $larger, smaller$ | refer to larger and smaller value of parameter in the two streams respectively       |
| $lm$              | logarithmic mean                                                                     |
| $m$               | mean                                                                                 |
| $s$               | refers to single shell                                                               |

## 2. PURPOSE AND SCOPE

This Data Item is one of a group on the selection and costing of heat exchangers. The first Item in the group, ESDU 92013<sup>11</sup>, provides a means of selecting, from a wide range of exchanger types, those feasible for a particular duty. Its objective is to provide the *process designer* with a simple and rapid method of selecting between the feasible types, and subsequently of obtaining rough cost estimates. Such estimates are often required well before the actual configuration of the equipment is decided.

This Item extends the costing data for shell-and-tube exchanger types, accounting for the variation of design parameters for which single 'standard' values were previously assumed.

In the procedure outlined in ESDU 92013, the first step is to select feasible types of heat exchanger on the basis of thermal and hydraulic requirements, compatibility with fluids and operating conditions (for example, temperature and pressure), maintenance, availability, space and weight, and fouling characteristics. Having established a list of feasible types, the next stage in the selection process is to carry out approximate costing respective to each type and then to select a heat exchanger (and its associated circuitry) having the lowest cost. It was recognised that the cost calculations in such procedures are of limited accuracy and that quotations for more than one type of exchanger should be sought if the costs are similar (that is within 20-50%).

In the costing method used in ESDU 92013 costs are expressed in terms of  $C$ , the cost per unit  $\dot{Q}/\Delta T_m$  where  $\dot{Q}$  is the heat load (heat transferred from the hot to cold fluid in the heat exchanger per unit time), and  $\Delta T_m$  is the mean temperature difference, corrected for heat exchanger configuration. A further summary of the method is given in Section 4.1 of that Item. For a given exchanger type, a 'standard' set of design parameters is chosen. However, costs are often highly dependent on variations in these design parameters, even for a constant value of  $\dot{Q}/\Delta T_m$ , and particularly for tubular exchangers such as air-cooled and shell-and-tube types. Simple correction factors are suggested; for instance, it is suggested in ESDU 92013 that stainless steel exchangers would be 1.5 times the cost of carbon steel exchangers. However, the use of such simple correction factors is of doubtful validity. For instance, it is self-evident that the ratio of cost for a stainless steel exchanger to that for a carbon steel exchanger would vary with the relative importance of the material content with respect to labour content, which would increase as the size of the exchanger increased.

In this present Item the specific case of the shell-and-tube heat exchangers is investigated further, with an attempt to develop correction factors to the table for the 'standard' parameters in order to produce more realistic costings over a wider range of configurations. ESDU 94043 provides similar extended data for the other tubular heat exchanger case, namely air-cooled types.

## 3. SELECTION OF CONFIGURATION

Shell-and-tube heat exchangers are almost universally designed according to the Standards of the Tubular Exchanger Manufacturers Association (TEMA)<sup>2</sup>, and the configurations laid down by this body are illustrated in Sketch 3.1. Configurations are indicated in terms of the front end type (A-D), the shell type (E-K and X) and the rear end type (L-W) as illustrated. Thus, a BES shell has a bonnet (integral cover) front end, a one-pass shell and a floating head with backing device at the rear end. The most common types of shell-and-tube heat exchanger used are as follows:

### BEM

This is termed the “fixed tube sheet” type and is commonly used for its simplicity and low relative cost. The fixed tube sheet type has the advantage of no gasketed internal joints. Disadvantages are that whilst the inside of tubes are accessible for mechanical cleaning the outside surface is not, and that expansion bellows often need to be fitted into the shell to absorb differential expansion between tube and shell.

### BEU

The “U-tube” is also simple and economical. Again, there are no gasketed internal joints. The bundle is removable so the outside of tubes can be mechanically cleaned, but mechanical cleaning is difficult for the inside of the tubes due to the U-bend. The presence of U-tubes eliminates the requirement for bellows but also leads to a reduced tube count for the same shell size and makes it difficult to replace tubes. As “U-tubes” are at least two-pass, and a BEU design cannot have pure counterflow, there is reduced effectiveness (that is, reduced effective mean temperature difference). An alternative option is to use a longitudinal baffle (BFU) which allows countercurrent flow and is relatively economic, though care has to be taken to minimise leakage at the baffle edge. If the baffle is welded to the shell to achieve this, the bundle is not removable. Alternatively a shroud cylinder may be used between the tubes and the shell.

### AES

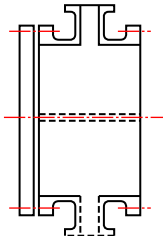
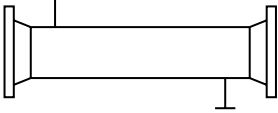
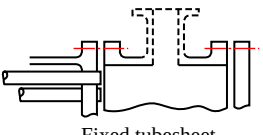
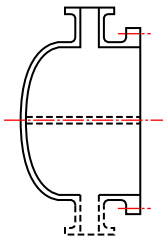
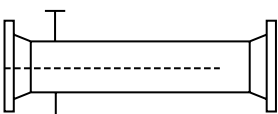
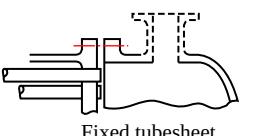
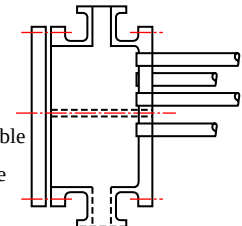
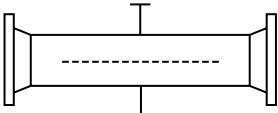
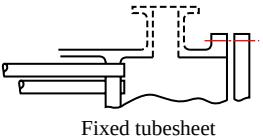
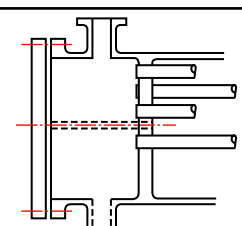
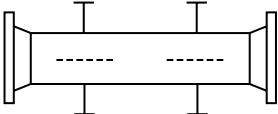
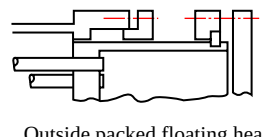
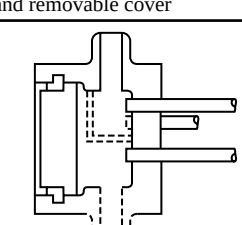
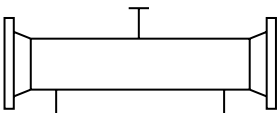
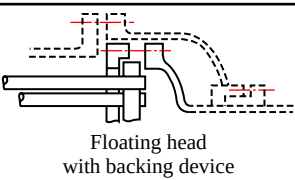
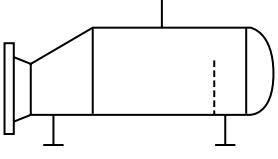
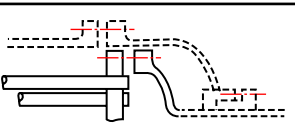
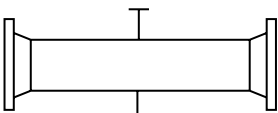
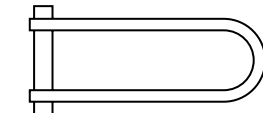
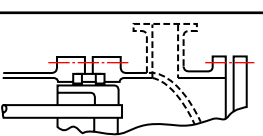
Despite its relatively high cost, the “Floating Head” type is the preferred exchanger for many petrochemical and process users. The tube bundle is fully removable and maintainable, though maintenance time is longer than for other types. Tubes can be mechanically cleaned internally and externally. The “Floating Head” design allows fewer tubes than a fixed tube sheet of the same diameter, but in this case the difference is minimised by the use of a split backing ring to attach the floating cover to the tube sheet. The design is robust and versatile but only allows pure counterflow with the additional complexity of an internal bellows. However, the use of the AES configuration does introduce the need for a gasketed internal joint.

### BES

The use of a bonnet rather than a channel and removable cover plate is cheaper in cost but, for large sizes, this is offset by the higher maintenance costs and the A-type closure is usually chosen in these cases.

Other types may be required for certain process or maintenance reasons. Removable covers (A or L type heads) may be desired for access to tubes without disturbing pipework. Integral tube sheets (C or N type) would be considered for higher pressure or high temperatures where they are cheaper than removable covers, and where gasketed joints need to be minimised. The T type of floating head is similar to the S type but has a more robust tube sheet to floating cover joint at the expense of greater bypass area around the bundles (and, thus larger shells) and, therefore, is more expensive. However, with the T type, the bundle may be removed without dismantling the floating head assembly, unlike the S type, thus reducing maintenance time. The W and P type floating heads utilise packed glands and are therefore limited on application by pressure and temperature. They are suitable for applications where leakage of the gasket

joint is to atmosphere. The P type is considerably more expensive than the W type. The NEN configuration is sometimes found to be an economic option.

|   | Front end stationary head types                                                                                                                                          |   | Shell types                                                                                                                  |   | Rear end head types                                                                                                             |
|---|--------------------------------------------------------------------------------------------------------------------------------------------------------------------------|---|------------------------------------------------------------------------------------------------------------------------------|---|---------------------------------------------------------------------------------------------------------------------------------|
| A | <br>Channel and removable cover                                                         | E | <br>One pass shell                          | L | <br>Fixed tubesheet like "A" stationary head |
| B | <br>Bonnet (integral cover)                                                             | F | <br>Two pass shell with longitudinal baffle | M | <br>Fixed tubesheet like "B" stationary head |
| C | <br>Removable tube bundle only<br>Channel integral with tubesheet and removable cover | G | <br>Split flow                              | N | <br>Fixed tubesheet like "N" stationary head |
| N | <br>Channel integral with tubesheet and removable cover                               | H | <br>Double split flow                     | P | <br>Outside packed floating head            |
| D | <br>Special high pressure closure                                                     | J | <br>Divided flow                          | S | <br>Floating head with backing device      |
|   |                                                                                                                                                                          | K | <br>Kettle type reboiler                  | T | <br>Pull through floating head             |
|   |                                                                                                                                                                          | X | <br>Cross flow                            | U | <br>U-tube bundle                          |
|   |                                                                                                                                                                          |   |                                                                                                                              | W | <br>Externally sealed floating tubesheet   |

Sketch 3.1 CONFIGURATION NOMENCLATURE ADOPTED BY THE TUBULAR EXCHANGERS MANUFACTURERS ASSOCIATION (TEMA)

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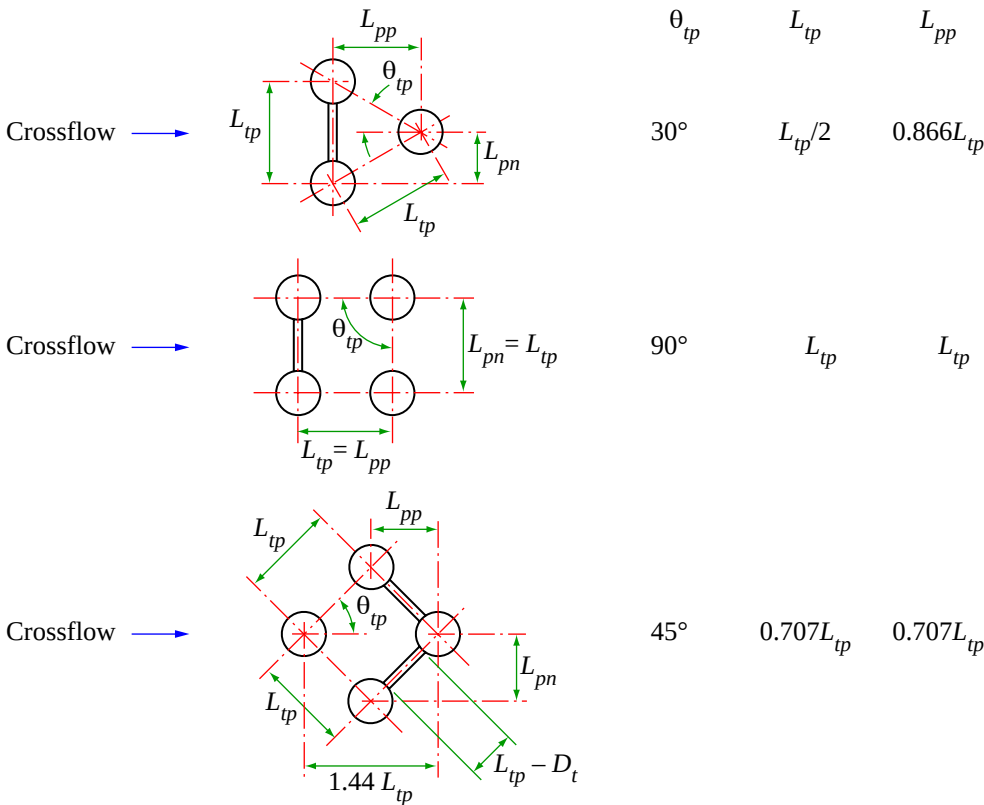
The various configurations may be listed in order of increasing cost (left to right) as follows:

|            |                          |                         |
|------------|--------------------------|-------------------------|
| Front end: | B, N, C, A               |                         |
| Rear end:  | U, M, N, L, W*, S, T, P* | (* Limited application) |

(For high pressure either a U-tube or a special type of D header would be used.)

Selection of the shell type is largely a function of the thermal-hydraulic design. However, an E-shell is usually chosen unless there are good reasons to do otherwise. For example, a K type would be chosen for many vaporisers. Types G and H are also often used for vaporising duties where low pressure drop is required, (for example, thermosyphons). Types F are used to provide pure counterflow in a two tube pass unit. The cost factor depends upon the requirements for the baffle seal which varies from simple flexible strips, to a central baffle welded to the shell, to a completely shrouded section. Pressure drop is approximately eight times that for an E shell. Potential leakage through the gap between the longitudinal baffle and the shell may restrict the range of application. The J type, or “divided flow” type, has one inlet and two outlet nozzles (thus dividing the flow in half) or vice versa. This results in a pressure drop of approximately one eighth that of the E type, and the main use of the J type is therefore for low pressure applications such as gas coolers and condensers. The X type shell has the shell side flow in pure crossflow without cross baffles. The result is an extremely low pressure drop and, again, it is used for gases and condensing vapour at low pressures.

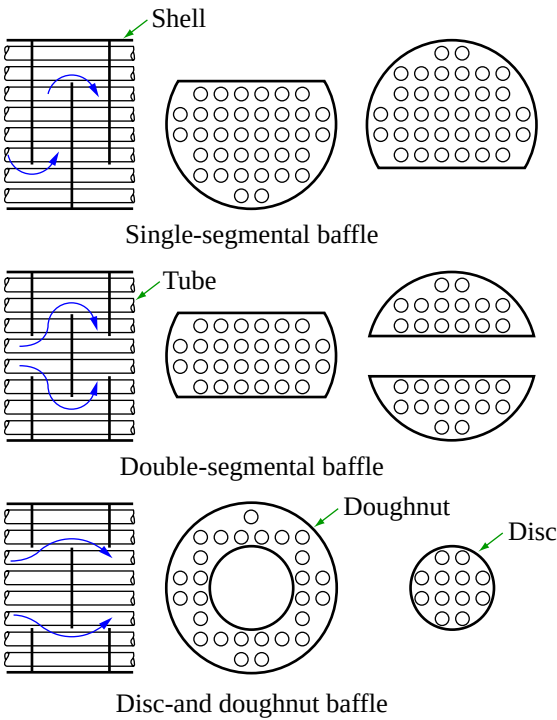
In addition to pressure and materials of construction (investigated in the present study), factors which are important to consider in selecting the configurations are as follows.



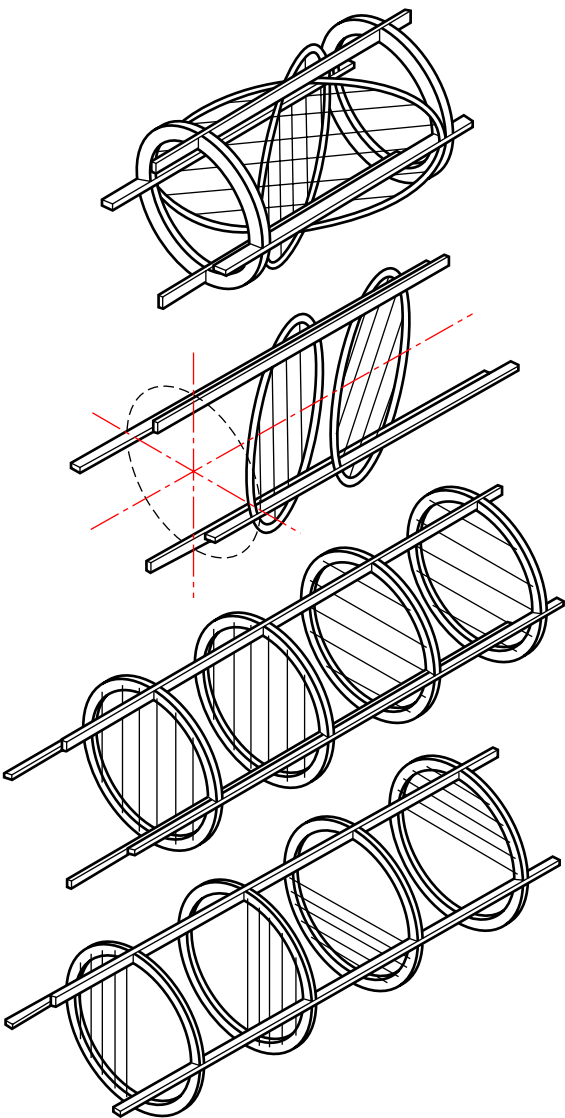
Sketch 3.2 Some examples of tube arrays used in shell-and-tube heat exchangers

- (1) **Tube diameter:** Thermal hydraulic considerations favour small tube diameter because of the greater heat transfer surface per unit volume. However, tube cleaning practices normally limit tube diameters to a minimum of approximately 16 mm outer diameter. Reducing the tube diameter in design leads to shorter tubes but, when cleaning becomes necessary, more operations per tube sheet are required.
- (2) **Tube length:** The best overall performance is obtained for heat exchangers having tube length to shell diameter ratios in the range typically from 5 to 12, but economics may push towards longer lengths, as will be discussed below.
- (3) **Tube layout and pitch:** Sketch 3.2 illustrates the tube arrays commonly used in shell-and-tube heat exchangers, namely equilateral triangular, square and staggered square. The triangular arrangement leads to the most compact design whereas the square array simplifies some maintenance operations (for example, cleaning on the shell side). A typical tube layout has a pitch of  $1.25 \times$  tube diameter. Generally, for clean services, the smallest pitch and a triangular layout is the preferred choice. Square or staggered square layouts are appropriate when mechanical cleaning on the shell side is required.
- (4) **Baffle design:** Common baffle configurations are illustrated in Sketch 3.3. In the present study, single segmental baffles were assumed. If a low pressure drop is required, then double segmental or possibly disc and doughnut baffles may be employed. Sometimes flow induced vibration is predicted to be a problem (see ESDU 87019<sup>9</sup>); to deal with this problem the tubes may be omitted from the "window" regions between the straight edge of the baffle and the shell circumference. Alternatively, rod baffle designs as shown in Sketch 3.4 may be used and for highly viscous fluids, alternative baffle configurations, for instance the helical baffle design shown in Sketch 3.5, may be used.
- (5) **Augmentation:** The use of tube inserts (twisted tapes, wire loops, etc.) for turbulence promotion is often economic in laminar and transitional/laminar conditions and should be considered. Correspondingly, augmentation of the outside surface (for example, by using low fin tubes (see ESDU 84016<sup>8</sup>)) should be considered when the shell side heat transfer coefficient is significantly lower than the tube side coefficient. These methods can be particularly effective for feed effluent exchangers. Low-fin tubes are often also used for the condensation of low surface tension fluids where advantage can be taken of the increased surface area.

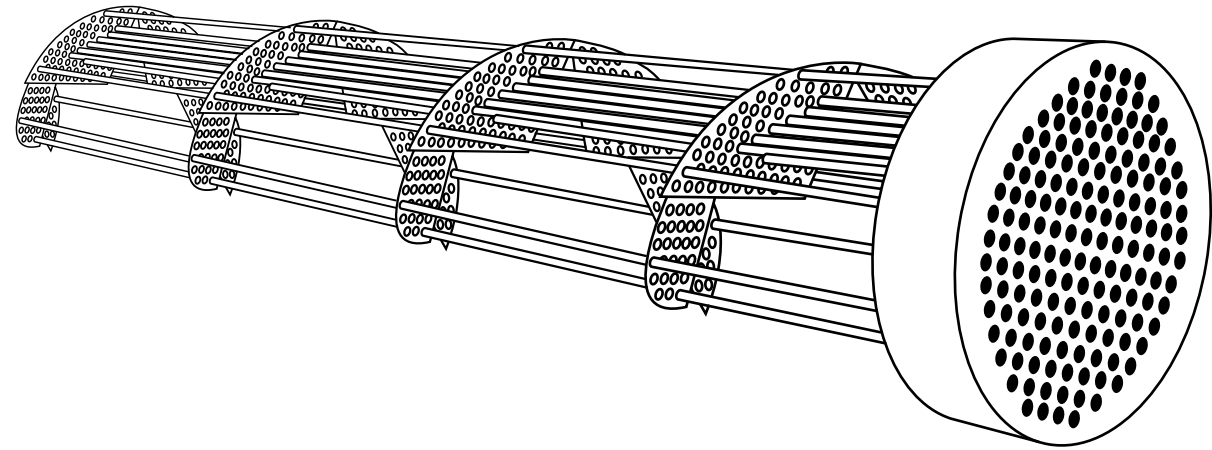
Factors 1 - 4 only were investigated in the present work



Sketch 3.3 Baffle designs



Sketch 3.4 Rod baffle support systems



Sketch 3.5 Spiral baffle arrangements

## 4. SUMMARY OF COSTING METHOD

The costing method given in ESDU 92013 is for single shells, and is summarised in Section 4.1. However, since this Item is primarily aimed at the process designer (rather than the designer of a specific exchanger specified by the process designer) it is also important to address the problem of multiple shells, the use of which is often dictated by process thermal requirements; this issue is addressed in Section 4.2.

### 4.1 Single Shells

The heat load  $\dot{Q}$  is calculated from the expression:

$$\dot{Q} = \dot{M}_h(h_{h,in} - h_{h,out}) = \dot{M}_c(h_{c,out} - h_{c,in}) . \quad (4.1)$$

For a single phase flow with constant specific heat capacity, it follows that:

$$\dot{Q} = \dot{M}_h c_{p,h}(T_{h,in} - T_{h,out}) = \dot{M}_c c_{p,c}(T_{c,out} - T_{c,in}) . \quad (4.2)$$

For a pure countercurrent flow heat exchanger with constant  $U$  and constant specific heat capacities, the mean temperature difference  $\Delta T_m$  is given as the logarithmic mean temperature difference  $\Delta T_{lm}$ :

$$\Delta T_{lm} = \frac{(T_{h,in} - T_{c,out}) - (T_{h,out} - T_{c,in})}{\log_e \frac{(T_{h,in} - T_{c,out})}{(T_{h,out} - T_{c,in})}} . \quad (4.3)$$

For exchangers which depart from pure countercurrent flow, typified by the multi-pass shell-and-tube heat exchanger, the mean temperature difference is given by:

$$\Delta T_m = F_T \Delta T_{lm} , \quad (4.4)$$

where  $F_T$  is a correction factor, whose value can be determined from standard charts (see for instance those of Taborek<sup>4</sup>) in terms of the end temperatures of the respective fluids. Typical practical values of  $F_T$  are around 0.9 for multi-pass shell-and-tube exchangers and around 0.7 for pure cross-flow exchangers. For  $F_T$  values less than about 0.8, multiple shell units are usually recommended (see Section 4.2).

The ratio  $\dot{Q}/\Delta T_m$  may be evaluated using the above equations, evaluating  $\Delta T_m$  using the  $F_T$  method. However, it is recommended here that the effectiveness –  $N_{TU}$  method be employed, as presented by ESDU<sup>11, 12</sup>. Bearing in mind the definition of  $N_{TU}$  as

$$N_{TU} = \frac{UA}{(\dot{M}c_p)_{smaller}} = \frac{(\dot{Q}/\Delta T_m)}{(\dot{M}c_p)_{smaller}} , \quad (4.5)$$

the ratio  $\dot{Q}/\Delta T_m$  maybe calculated from:

$$\frac{\dot{Q}}{\Delta T_m} = (\dot{M}c_p)_{smaller} N_{TU} , \quad (4.6)$$

where the Number of Transfer Units,  $N_{TU}$ , can be calculated as a function of effectiveness,  $E$ , and the ratio of flow heat capacities,  $C^*$ , using, for example, data published by ESDU<sup>12</sup> for a wide range of exchanger types.

These data are typified by Figure 1, which shows the curves for  $E$  as a function of  $C^*$  and  $N_{TU}$  for a two-pass shell-and-tube heat exchanger. Here,  $E$  and  $C^*$  are defined as:

$$E = \frac{(\dot{Q})}{(\dot{Q})_{max}} = \frac{(T_{in} - T_{out})_{larger}}{(T_{h,in} - T_{c,in})} \quad (4.7)$$

and

$$C^* = \frac{(\dot{M}c_p)_{smaller}}{(\dot{M}c_p)_{larger}}. \quad (4.8)$$

Referring to Figure 1, it will be seen that, for a given value of  $C^*$ ,  $E$  asymptotes to a maximum value ( $E_{max}$ ) as  $N_{TU}$  increases. Shown on Figure 1 is a line representing  $E = 0.95 E_{max}$ . This represents a practical limit line for  $E$ , corresponding approximately to the limit of  $F_T = 0.8$  usually adopted for the  $F_T$  method as mentioned above; if  $E$  is greater than  $0.95 E_{max}$ , then multiple shells in series should be used (see Section 4.2).

The ESDU Items<sup>11, 12</sup> on the Effectiveness –  $N_{TU}$  method also give, for many cases, formulae from which the  $N_{TU}$  value can be calculated if  $E$  and  $C^*$  are known. For shell-and-tube heat exchangers with even numbers of tube side passes, the value of  $N_{TU}$  is given by

$$N_{TU} = \frac{\log_e \left[ \frac{2 - E(1 + C^* - \eta)}{2 - E(1 + C^* + \eta)} \right]}{\eta} \quad (4.9)$$

where

$$\eta = (C^{*2} + 1)^{\frac{1}{2}}. \quad (4.10)$$

The value of  $E_{max}$  is given by

$$E_{max} = \frac{2}{1 + C^* + \eta}. \quad (4.11)$$

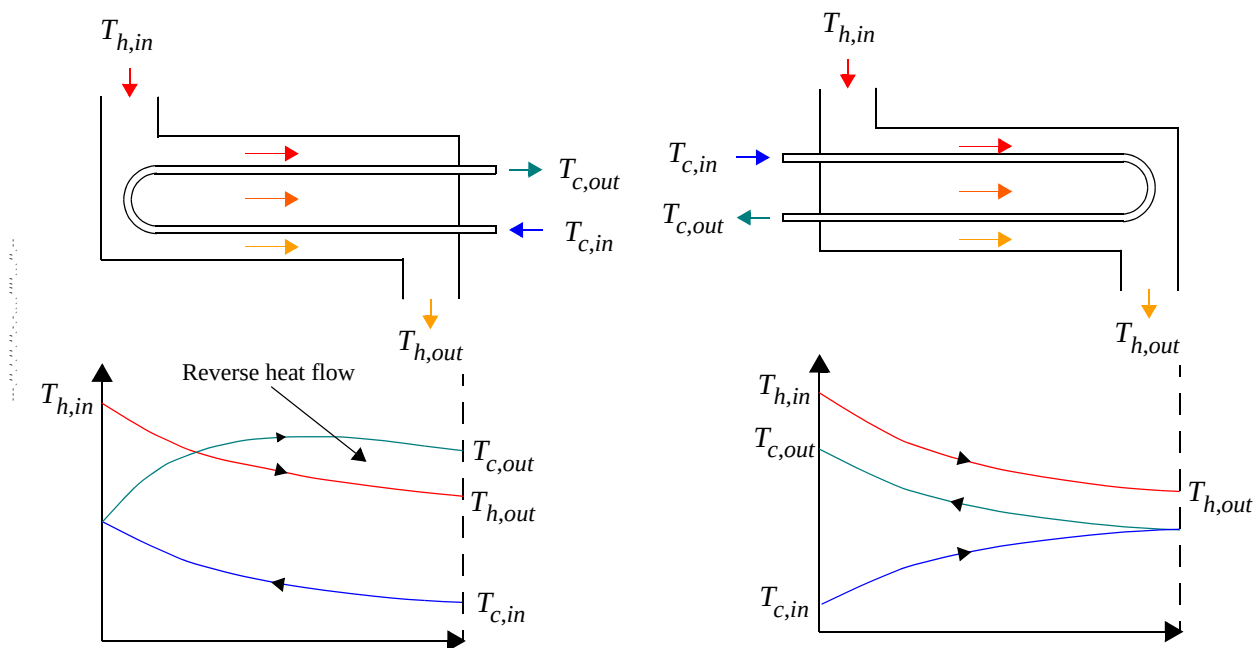
Having established the value of  $\dot{Q}/\Delta T_m$ , the next stage is to evaluate the cost of the exchanger per unit  $\dot{Q}/\Delta T_m$ , by logarithmic interpolation of the costing data in Table 12.1. The cost of the exchanger is given by the product of  $C$  and  $\dot{Q}/\Delta T_m$ . The tables for  $C$  also include values for  $U$ , the overall heat transfer coefficient. In this simple methodology,  $U$  is taken as constant for a given pair of fluids. Thus, the heat exchanger surface area,  $A$ , is given by:

$$A = \left( \frac{\dot{Q}}{\Delta T_m} \right) \cdot \frac{1}{U}. \quad (4.12)$$

In the case of the shell-and-tube heat exchanger, the overall heat transfer coefficient is defined with respect to a surface area corresponding to the outside of the tubes.

## 4.2 Multiple Shells

It is normal practice to use shell-and-tube heat exchangers with two or more tube side passes. This can give rise to the so-called “temperature meet” effect, which is illustrated in Sketch 4.1. Consider first the case where the fluid in the first tube side pass is flowing in the opposite direction (that is, “countercurrent”) to the hot side fluid (as shown in Sketch 4.1a). Countercurrent flow gives the highest efficiency and the interpass temperature,  $T_i$ , reaches a high value. However, in the second pass the tube side temperature may exceed the shell side temperature, as illustrated. This gives a reverse heat flow and a reduction of efficiency. Now consider the case where the fluid in the first tube side pass is flowing in the same direction as the shell side fluid (that is, “co-current”, see Sketch 4.1b). Here, with co-current flow, the value of  $T_i$  is much smaller since co-current flow is much less efficient. Even though the second pass is countercurrent, the overall efficiency is still low relative to a pure counter-flow exchanger. Note that the cold side fluid outlet temperature,  $T_{i,out}$  is identical for the two cases, as can be shown analytically.



Sketch 4.1 Temperature meet effects

The phenomenon in which  $T_{c,out} > T_{h,in}$  in multiple pass shells is often referred to as a “temperature meet” or “temperature cross”. The effects of such temperature meets (or crosses) should normally be minimised (hence the limits of  $F_T < 0.8$  and  $E/E_{max} > 0.95$  described in the previous section). To avoid a temperature meet in a single shell, the following condition must be satisfied<sup>2</sup>

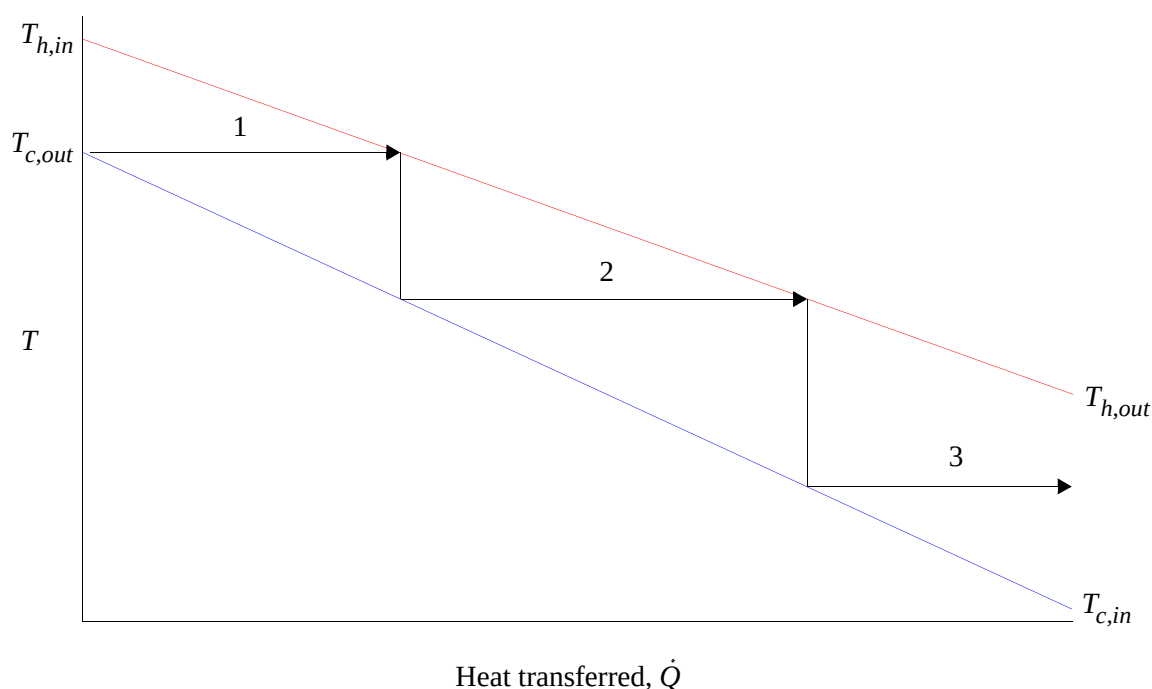
$$E < \frac{1}{1 + C^*} \quad (4.13)$$

This condition is also shown on the ESDU graphs (see Figures 1-6). However, there are many process conditions where it is not possible to meet this condition and, if exchangers with two tube side passes are to be used, the usual approach is to employ multiple identical shells in series. In effect, this brings the heat exchange process closer to one with pure countercurrent flow. The number of shells in series required to avoid a temperature cross is given by:

$$S > \frac{\log_e \left( \frac{1 - E}{1 - EC^*} \right)}{\log_e C^*} \quad (4.14)$$

In practice, the value of  $S$  is chosen as the next largest integral value above that given by Equation (4.14).

An alternative approximate graphical method of determining the number of shells required in series is shown in Sketch 4.2. Here,  $T_h$  and  $T_c$  are plotted as a function of heat transferred,  $\dot{Q}$ . Starting at  $T_{c,out}$ , a line is drawn horizontally across to the  $T_h$  versus  $\dot{Q}$  curve. From the point of intersection, a vertical line is drawn downwards until it meets the  $T_c$  versus  $\dot{Q}$  curve. The process is continued until the last horizontal line crosses the limit for  $\dot{Q} = \dot{Q}_{TOT}$ , the total amount of heat to be transferred. The number of shells required is then the number of horizontal lines drawn (this is equivalent to taking the next largest integer in using Equations (4.10)). As will be seen, this would be three shells in the example shown in Sketch 4.2.



**Sketch 4.2 Stepping procedure for determining number of shells in series required to avoid temperature cross**

Having established the number of shells in series, the procedure is then similar to that used for a single shell. However, the  $F_T$ <sup>4</sup> and Effectiveness –  $N_{TU}$ <sup>12</sup> charts used are those appropriate for the number of shells in series. The ESDU charts<sup>12</sup> for 2 to 6 shells in series are given in Figures 2 to 6 respectively. Again, it is more accurate to use formulae to calculate the  $N_{TU}$  value. If  $E$  is defined as the overall effectiveness of the assembly of heat exchangers, as given by Equation (4.7), then an effectiveness for each sheet in the assembly may be defined as  $E_s$ . Note that, since it is assumed that each shell has an equal surface area, equal  $U$  and equal  $(\dot{M}_{cp})_{smaller}$ , then the  $N_{TU}$  values for the shells (see Equation (4.5)) is equal. This implies that, for a given value of  $C^*$  (Equation (4.8)),  $E_s$  is also constant. If  $E_s$  can be calculated, the  $\dot{Q}/\Delta T_m$  value for each shell (that is  $(\dot{Q}/\Delta T_m)_s$ ) can be estimated by calculating the corresponding value of  $N_{TU}$  ( $= (N_{TU})_s$ ) and using Equation (4.6) to calculate  $(\dot{Q}/\Delta T_m)_s$ .

An equation relating the overall effectiveness  $E$  and the effectiveness per shell  $E_s$  has been derived by Domingos<sup>1</sup>

$$E = \frac{\left( \frac{1 - E_s C^*}{1 - E_s} \right)^S - 1}{\left( \frac{1 - E_s C^*}{1 - E_s} \right)^S - C^*} \quad (4.15)$$