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## **Baffled shell-and-tube heat exchangers: flow distribution, pressure drop and heat transfer coefficient on the shellside**

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## BAFFLED SHELL-AND-TUBE HEAT EXCHANGERS: flow distribution, pressure drop and heat transfer coefficient on the shellside

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## BAFFLED SHELL-AND-TUBE HEAT EXCHANGERS: flow distribution, pressure drop and heat transfer coefficient on the shellside

### 1. NOTATION AND UNITS

Two sets of self-consistent units are given, the SI and the British.

|       |                                                                           | Units          |                 |
|-------|---------------------------------------------------------------------------|----------------|-----------------|
|       |                                                                           | SI             | British         |
| $A_b$ | bypass flow area (Equation (4.1))                                         | m <sup>2</sup> | ft <sup>2</sup> |
| $A_c$ | crossflow area (Equation (4.6))                                           | m <sup>2</sup> | ft <sup>2</sup> |
| $A_e$ | escape area under nozzle (Section 8.1)                                    | m <sup>2</sup> | ft <sup>2</sup> |
| $A_m$ | minimum area for flow between tubes in centre (Equation (4.7))            | m <sup>2</sup> | ft <sup>2</sup> |
| $A_n$ | nozzle flow area ( $\pi D_n^2/4$ )                                        | m <sup>2</sup> | ft <sup>2</sup> |
| $A_o$ | segmental area between baffle tip and outer tube limit (Equation (4.5))   | m <sup>2</sup> | ft <sup>2</sup> |
| $A_s$ | leakage area between baffle and shell (Equation (4.8))                    | m <sup>2</sup> | ft <sup>2</sup> |
| $A_t$ | leakage area between baffle and tubes passing through it (Equation (4.9)) | m <sup>2</sup> | ft <sup>2</sup> |
| $A_w$ | window flow area (Equation (4.10))                                        | m <sup>2</sup> | ft <sup>2</sup> |
| $c_p$ | specific heat capacity of fluid                                           | J/kg K         | Btu/lb °F       |
| $D_n$ | internal diameter of nozzle                                               | m              | ft              |
| $D_o$ | external diameter of tubes                                                | m              | ft              |
| $D_s$ | internal diameter of shell                                                | m              | ft              |
| $D_t$ | diameter of outer tube limit                                              | m              | ft              |
| $D_v$ | volumetric mean diameter (Equation (6.3))                                 | m              | ft              |
| $F_i$ | flow fraction through $i^{\text{th}}$ path                                | —              | —               |
| $f$   | friction factor                                                           | —              | —               |

For footnotes, refer to end of Notation Section

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|             |                                                                   |                                                  |                                                        |
|-------------|-------------------------------------------------------------------|--------------------------------------------------|--------------------------------------------------------|
| $g_c$       | constant in 2 <sup>nd</sup> law of motion*                        | –<br>(1.0)                                       | lbf ft/h <sup>2</sup> lbf<br>(4.17 × 10 <sup>8</sup> ) |
| $g_n$       | gravitational acceleration                                        | m/s <sup>2</sup><br>(9.81)                       | ft/h <sup>2</sup><br>(4.17 × 10 <sup>8</sup> )         |
| $h$         | distance between edges of baffles (Sketch 4.3)                    | m                                                | ft                                                     |
| $h_e$       | distance between the outer tube limit and the window (Sketch 8.1) | m                                                | ft                                                     |
| $K_i$       | pressure loss coefficient for $i^{\text{th}}$ path                | –                                                | –                                                      |
| $L_b$       | space between baffles                                             | m                                                | ft                                                     |
| $L_e$       | space between end baffle and tubesheet                            | m                                                | ft                                                     |
| $\dot{M}_b$ | mass flow rate bypassing bundle in crossflow                      | kg/s                                             | lb/h                                                   |
| $\dot{M}_c$ | mass flow rate in crossflow through the bundle                    | kg/s                                             | lb/h                                                   |
| $\dot{M}_s$ | mass flow rate in baffle/shell leakage                            | kg/s                                             | lb/h                                                   |
| $\dot{M}_T$ | total mass flow rate along the shell                              | kg/s                                             | lb/h                                                   |
| $\dot{M}_t$ | mass flow rate in baffle/tube leakage                             | kg/s                                             | lb/h                                                   |
| $\dot{M}_w$ | mass flow rate through window                                     | kg/s                                             | lb/h                                                   |
| $\dot{m}_i$ | mass flux in $i^{\text{th}}$ path                                 | kg/s m <sup>2</sup>                              | lb h ft <sup>2</sup>                                   |
| $N_b$       | number of baffles                                                 | –                                                | –                                                      |
| $N_c$       | number of tubes on centre-line                                    | –                                                | –                                                      |
| $N_s$       | number of sealing strips                                          | –                                                | –                                                      |
| $N_t$       | total number of tubes                                             | –                                                | –                                                      |
| $N_w$       | number of tubes passing through window                            | –                                                | –                                                      |
| $n_i$       | flow resistance of $i^{\text{th}}$ path given by Equation (5.2)   | N s <sup>2</sup> /m <sup>2</sup> kg <sup>2</sup> | lbf h <sup>2</sup> /ft <sup>2</sup> lb <sup>2</sup>    |
| $P_t$       | pitch of tubes                                                    | m                                                | ft                                                     |
| $P_x$       | pitch in direction of flow                                        | m                                                | ft                                                     |
| $p$         | static pressure                                                   | Pa (i.e. N/m <sup>2</sup> )                      | lbf/ft <sup>2</sup>                                    |

For footnotes, refer to end of Notation Section

|              |                                                                                                        |                    |                         |
|--------------|--------------------------------------------------------------------------------------------------------|--------------------|-------------------------|
| $Re$         | Reynolds number (Equation (6.2))                                                                       | –                  | –                       |
| $TLA$        | tube-layout-angle (see Sketch 4.5)                                                                     | degrees            | degrees                 |
| $t_b$        | baffle thickness (see Sketch 6.2)                                                                      | m                  | ft                      |
| $t_s$        | radial clearance between baffle and shell (see Sketch 6.1)                                             | m                  | ft                      |
| $t_t$        | radial clearance between baffle and tubes (see Sketch 6.2)                                             | m                  | ft                      |
| $w$          | clearance between outer tubes and shell (Sketch 4.1)                                                   | m                  | ft                      |
| $w'$         | clearance between tubes in centre of shell with pass partition parallel to flow direction (Sketch 4.2) | m                  | ft                      |
| $\alpha_c$   | shellside fluid heat transfer coefficient                                                              | W/m <sup>2</sup> K | Btu/h ft <sup>2</sup> F |
| $\Delta H$   | increase in height above datum from inlet to outlet nozzle                                             | m                  | ft                      |
| $\Delta p_i$ | pressure loss <sup>†</sup> in $i^{\text{th}}$ path                                                     | Pa                 | lbf/ft <sup>2</sup>     |
| $\Delta p_p$ | pressure loss <sup>†</sup> per crossflow pass                                                          | Pa                 | lbf/ft <sup>2</sup>     |
| $\Delta p_T$ | total pressure drop                                                                                    | Pa                 | lbf/ft <sup>2</sup>     |
| $\Delta p_w$ | pressure loss <sup>†</sup> in window                                                                   | Pa                 | lbf/ft <sup>2</sup>     |
| $\eta$       | viscosity of fluid <sup>‡</sup>                                                                        | kg/s m             | lb/h ft                 |
| $\theta$     | angle in Sketch 4.2                                                                                    | radians            | radians                 |
| $\theta'$    | angle in Sketch 4.3                                                                                    | radians            | radians                 |
| $\lambda$    | thermal conductivity of fluid                                                                          | W/m K              | Btu/h ft °F             |
| $\rho$       | density of fluid                                                                                       | kg/m <sup>3</sup>  | lb/ft <sup>3</sup>      |

## Extra subscripts

|       |          |
|-------|----------|
| $e$   | end zone |
| $in$  | inlet    |
| $out$ | outlet   |

\* Force = mass  $\times$  acceleration/ $g_c$ . In the SI system, the unit of force (N) is such that  $g_c$  is unity. In the British system using lb force and lb mass,  $g_c$  is numerically equal to the acceleration of a free-falling body.

† The term “loss” denotes the irreversible component of the pressure drop, due to skin friction and form drag.

‡ If the viscosity is given in centipoises, this should be divided by 1000 to convert to SI units or multiplied by 2.42 to convert to British units.

## 2. PURPOSE AND SCOPE OF THIS DATA ITEM

The purpose of this Data Item is to provide a simple method for estimating the flow rate through each of the various paths on the shellside of a baffled shell-and-tube heat exchanger with plain tubes on a square or a triangular array, and hence to predict the pressure drop and heat transfer coefficient of the shellside fluid. This gives an adequate solution for most cases, but it may be followed by more sophisticated calculations using a proprietary computer program. It is necessary to know the total mass flow rate of the shellside fluid, full geometrical details of the heat exchanger, with dimensions, and the physical properties listed in Section 2.2. The accuracy of this method is discussed in Section 2.3. Section 3 introduces the construction of shell-and-tube heat exchangers and summarises the development of the method given here, which is detailed in Sections 4 to 9. A summary of the calculation procedure is given in Section 10 and illustrated by an example in Section 11.

### 2.1 Types of Problem

This Data Item may be required for the solution of one of three problems.

- (i) Rating the performance of a given heat exchanger, when the rates and inlet temperatures are given and it is required to calculate the outlet temperatures and heat transfer rate.
- (ii) Evaluating the results of tests on a heat exchanger, where the measured heat transfer rate and pressure drop must be compared with predictions.
- (iii) Designing a new heat exchanger, where the thermal duty and maximum pressure drop are specified and a detailed design must be produced.

The last involves a considerable amount of iteration, starting with an approximate design, rating this and modifying it until a suitable design is evolved.

### 2.2 Physical Data Required

The method of calculation given in this Data Item is applicable to a single-phase Newtonian fluid. The physical properties required are the density and dynamic viscosity, for the fluid flow calculations, and also the specific heat capacity and thermal conductivity, if the shellside heat transfer coefficient is to be estimated. It is assumed that the variations of these physical properties are small and that they remain constant at the values corresponding to the mean temperature of the fluid along the length of the exchanger. If the properties are given at the inlet and outlet temperatures, an arithmetic mean may be taken, except in the case of the density of a gas; here there may be a considerable change in density, which should be estimated as described in Section 8.3, which also explains how to estimate the increase in pressure drop due to compressibility effects if the total pressure drop is found to be more than 1 or 2% of the absolute pressure.

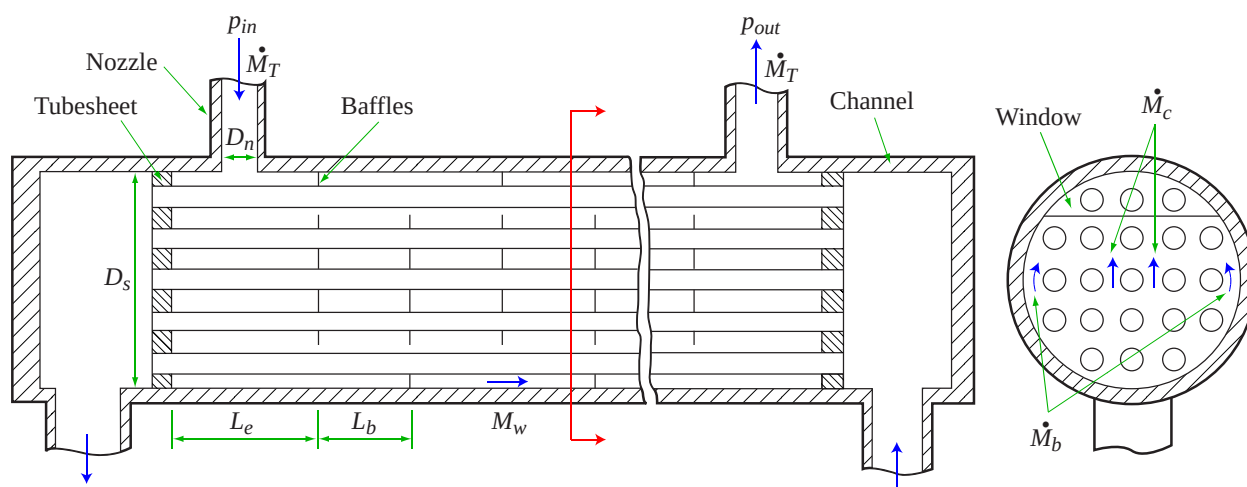
### 2.3 Accuracy of the Method

Figure 1 shows the ratio of the predicted pressure drop to the experimentally determined pressure drop, obtained from research at the University of Delaware<sup>9, 10</sup> and from proprietary research. The ratio is plotted against the superficial crossflow Reynolds number, defined by Equation (6.2). It is seen that when the Reynolds number is more than 1000, most of the points lie between 0.7 and 1.3. Thus it should be possible to predict pressure drop to within  $\pm 30\%$  at high Reynolds numbers, for the type of geometry used in the experiments; this is described in Section 3.2, which discusses how the method might be used for other geometries. In view of the difficulty of estimating the flow fractions and the fact that the pressure drop is roughly proportional to the square of the velocity, this degree of accuracy is considered to be very

satisfactory. The method underpredicts pressure drop when the Reynolds number is in the range 100 to 800, due to laminar flow in some or all of the paths. This can be allowed for by further iteration, but it then becomes rather too complicated for a hand calculation and it is preferable to use a computer program in such cases. In terms of industrial heat exchangers, the method described in this Item is likely to be suitable for all gases and all but highly viscous liquids. Steps (4) and (14) of Section 10 describe the calculations necessary to determine whether the method described in this Item is suitable for estimating pressure drop. The prediction of the heat transfer coefficient is likely to be more accurate.

### 3. BACKGROUND TO SHELL-AND-TUBE HEAT EXCHANGERS AND THE APPROXIMATE METHOD FOR ESTIMATING FLOW-PATH RATES, PRESSURE DROP AND HEAT TRANSFER COEFFICIENT

Sketch 3.1 shows a typical shell-and-tube heat exchanger, for which the mechanical design and construction would generally be in accordance with a suitable pressure vessel code, for example the American Society of Mechanical Engineers (ASME), supplemented by the Standards of the Tubular Exchanger Manufacturers' Association<sup>3</sup> (TEMA). It has a single pass on the tubeside and a single pass on the shellside, which is of the type known as a TEMA 'E' shell. The method given in this Data Item has been validated against experiments carried out on a shell of this type; extension of the method to other types is discussed in Section 3.2.



Sketch 3.1

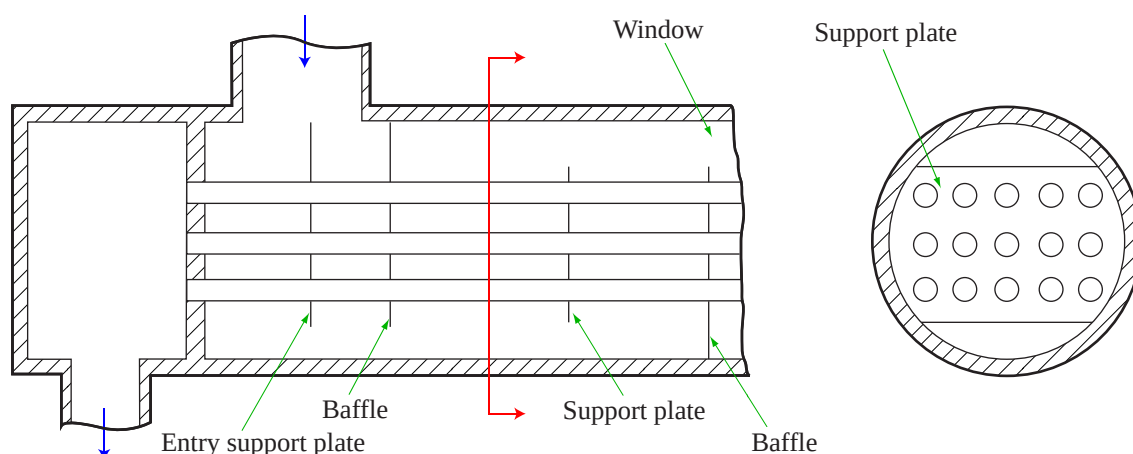
Sketch 3.1 Shell-and-tube heat exchanger, showing shellside pressure ( $p$ ) and flow rates ( $\dot{M}$ ), except for  $\dot{M}_s$  (leakage between baffles and shell) and  $\dot{M}_t$  (leakage between baffles and tubes). The window is the segmental space between the shell and the straight edge at the top or bottom of each baffle. Further information is contained in the sketches in Section 4 and in Section 7. If the clearances between a baffle and the shell and between a baffle and the tubes are not given, the values given by TEMA should be assumed. These clearances are very important; they are difficult to measure, and they have a considerable influence on the amount of fluid that leaks through them.

## 3.1 Baffles

The baffles perform the dual function of supporting the tubes and of deflecting the flow of fluid at right-angles to the tubes, except in the region of flow from one baffle space to the next. In all experiments used to validate the method described in this Data Item the baffles were of the single-segmental type, shown in Sketch 3.1; with this type the main flow of fluid from one crossflow pass to the next is through the gap between the edge of the baffle and the shell, known as the “window”.

It is assumed here that all baffles are equally spaced, but allowance must be made for the fact that this spacing usually differs from the end spacing between the end baffles and the tubesheet. It is very important to check that these spacings are sufficiently close to give adequate support to the tubes and prevent serious vibration of the tubes.

One method of keeping within the required maximum unsupported length of tubes is to design the heat exchanger so that there are no tubes in the window, as shown in Sketch 3.2. With this arrangement, additional support plates may be added between the baffles. This is an important design for exchangers handling a large volume rate of gas on the shellside. No experimental data are available for this arrangement, but a method of dealing with it is suggested. It is expected that the addition of the support plates will not affect the flow distribution or pressure drop.



**Sketch 3.2**

Sketch 3.2 End of shell-and-tube heat exchanger with no tubes in the window. The sketch also shows optional support plates to steady the tubes at entry (on the centre-line of the nozzle) and in the centre of the spaces between baffles.

## 3.2 Extension to Other Shell and Baffle Types

Although Sketch 3.1 shows a single pass on the tubeside, the method in this Data Item is applicable to any number of tubeside passes. In addition to being applicable to TEMA ‘E’ shells, the method may be applied to TEMA ‘J’ shells (where the fluid enters the centre of the shell and leaves at the two ends, or vice versa). It is not applicable to shells where there is a central longitudinal baffle (TEMA ‘F’, ‘G’, ‘H’) because no allowance can be made for fluid and thermal leakage through a central baffle. (The various types are shown on page 3 of TEMA<sup>3</sup>.)

The method, as defined, may not be applied to other types of baffle, *e.g.* double segmental or “disc and doughnut”, because it has been devised specifically for single-segmental baffles and the flow pattern is considerably different with other types. However, it would not be difficult to modify the method to suit other types of baffle.

## 3.3 Exchanger Performance

For rating, evaluating or designing a shell-and-tube heat exchanger, it is necessary to predict the pressure drop of both fluids and the overall heat transfer coefficient. Reference 1 gives the recommended procedure for estimating the heat transfer coefficient of the tubeside fluid, and Reference 2 deals with the heat transfer coefficient of a fluid flowing at right-angles across a bank of tubes of constant cross-section. On the shellside of a baffled shell-and-tube heat exchanger the flow area is not constant and the fluid does not flow entirely across the bundle, owing to the presence of bypass and leakage paths. Thus in the typical arrangement shown in Sketch 3.1, the total flowrate entering through the inlet nozzle ( $\dot{M}_T$ ) is split into the following streams as it flows through the shell, before the streams are reunited and pass through the outlet nozzle:—

$\dot{M}_b$  — flow through the bypass space between the bundle and the shell,

$\dot{M}_c$  — flow across the tubes,

$\dot{M}_s$  — leakage between the shell and the baffles,

$\dot{M}_t$  — leakage between the tubes and the baffles,

$\dot{M}_w$  — flow through the window.

The lower-case suffixes used above are based on the first letter of the name of each stream. They are preferred to the less rational system of Tinker<sup>6</sup>, which have been copied in many other publications. In Tinker's nomenclature, the crossflow stream is denoted by B, the bypass stream by C and the leakage streams by A and E.

It is possible that fouling on the outside of the tubes also occurs in the space between the tubes and the baffles, thus considerably reducing, or completely eliminating the leakage  $\dot{M}_t$ . When it is not possible to predict whether the high velocity of flow through this leakage path will prevent fouling, it is advisable to estimate the lowest possible heat transfer coefficient by assuming no fouling between tubes and baffles, and the highest possible pressure drop by assuming the normal amount of fouling all over the tubes, which can be determined from the specified fouling resistance and an approximate estimate of the thermal conductivity of the deposit.

As  $\dot{M}_c$  ranges typically from 40% to 70% of  $\dot{M}_T$  in a clean heat exchanger, the use of the procedure in Reference 2 would lead to a serious over-estimation of the external heat transfer coefficient if no allowance were made for the amount of fluid that passes through the leakage and bypass paths. Using the procedure in Reference 4 in the same way to estimate the shellside pressure drop would also lead to overprediction; here the error would be greater because the pressure drop is proportional to the flow rate raised to a power of about 1.8 whereas the external heat transfer coefficient is proportional to the rate raised to a power of only about 0.6 and there is a further reduction when the thermal resistance of the wall and internal fluid are allowed for.

The purpose of this Data Item is to enable the user to estimate the flow rate through the streams listed above, given details of the heat exchanger, the total flow rate ( $\dot{M}_T$ ) and the density and viscosity of the fluid. A procedure is then given for estimating the total pressure drop. An approximate method is given for estimating the external heat transfer coefficient, but this will be dealt with more thoroughly in a future Data Item.

## 3.4 Background to the Proposed Method of Calculation

The first analysis of shellside flow was that of Tinker<sup>6,7,8</sup>. He suggested the various streams, but his method is complex, difficult to understand and involves extensive iteration. Also at the time of its first publication in this country (1951) it was generally neglected, owing to the absence of computers suitable for such

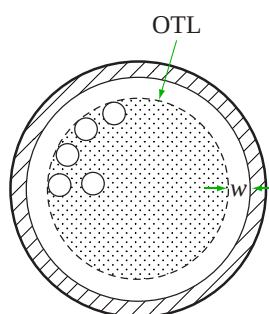
complicated calculations and the lack of experimental confirmation. During the next decade the subject was considerably advanced by the extensive research at the University of Delaware, which has been summarised by Bergelin *et al*<sup>9</sup> and by Bell<sup>10</sup>. Bell eliminated the need to estimate the flow fractions by deriving correction factors to be applied to the pressure drop and heat transfer coefficient calculated on the assumption that the total flow is in crossflow. This method has been developed and expressed as a calculational procedure by Taborek<sup>5</sup>. However the largest heat exchanger in the Delaware research had only 470 ¼-inch tubes. Thus there is some apprehension in extrapolating this semi-empirical approach to heat exchangers with many more tubes of larger diameter.

To overcome the above problems, the procedure recommended in this Data Item is based on a simplification of Tinker's method. It is felt that as it has a sound theoretical basis, it can be extrapolated with less apprehension. Details of the development of this method are given by Wills<sup>12</sup>. The procedure, which is detailed in Section 10, is quite simple. Figure 1 shows Wills' comparison between predicted and experimental values of the pressure drop per baffle space.

## 4. GEOMETRICAL CALCULATIONS

As a preliminary to calculations in fluid flow, it is necessary to carry out the following geometrical calculations to determine the cross-sectional areas of the five flow paths that convey the streams  $\dot{M}_b$ ,  $\dot{M}_c$ ,  $\dot{M}_s$ ,  $\dot{M}_t$  and  $\dot{M}_w$  (defined in Section 3.3).

### 4.1 Bypass Area ( $A_b$ )



Sketch 4.1 Tube layout with one tubeside pass

The bypass stream flows parallel to the crossflow stream across the tubes. Sketch 4.1 shows a cross-section of a shell with a single pass on the tubeside, the space occupied by the tubes being shown speckled. The circle which circumscribes all the tubes is known as the "outer tube limit" (OTL); it is shown by the dotted line in Sketch 4.1. It is necessary to determine the diameter of the outer tube limit, which is denoted by  $D_t$ . The area for flow associated with the bypass stream ( $\dot{M}_b$ ) in this case is

$$A_b = 2wL_b \quad (4.1)$$

where  $A_b$  is the bypass flow area,  $L_b$  is the space between baffles (see Sketch 3.1) and  $w$  is the width of the space between the outermost tubes and the shell; it equals  $\frac{1}{2}(D_s - D_t)$ , where  $D_s$  is the internal diameter of the shell (see Equation (4.3)).

When several passes are required for the tubeside fluid, it is necessary to install plates in the end spaces known as the "channels" and indicated in Sketch 3.1. The function of these plates, known as "pass partition plates", is to direct the tubeside fluid through the tubes allocated to each pass. A gasket must be used to