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Selection and costing of heat exchangers

Plate-fin type

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SELECTION AND COSTING OF HEAT EXCHANGERS: PLATE-FIN TYPE

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SELECTION AND COSTING OF HEAT EXCHANGERS: Plate-fin type

1. NOTATION AND UNITS

		Units	
		SI	British
A	heat transfer projected surface area of heat exchanger	m^2	ft^2
a	surface areas between plates per unit area (see Appendix A1)	–	–
B	overall volumetric heat transfer coefficient	$\text{W}/\text{m}^3 \text{ K}$	$\text{Btu}/\text{ft}^3 \text{ h } ^\circ\text{F}$
b	plate gap (see Appendix A1)	m	ft
c_p	specific heat capacity	$\text{J}/\text{kg K}$	$\text{Btu}/\text{lb } ^\circ\text{F}$
c	correlating parameter of Chisholm, Equation (A4.3)	–	–
C	cost per unit $\dot{Q}/(\Delta T)$	$\text{£}/(\text{W}/\text{K})$	$\text{£}(\text{Btu}/\text{h})/^\circ\text{F}$
d_H	hydraulic diameter of flow channel	m	ft
f	Fanning friction factor ($= d_H \rho \Delta p / (2 L \dot{m}^2 g_c)$)	–	–
f_a	parameter given by Equation (A3.2)	–	–
f_l	parameter given by Equation (A3.3)	–	–
f_n	fin frequency (fins/length)	$1/\text{m}$	$1/\text{ft}$
F	parameter given by Equation (A4.6)	–	–
F_h, F_c	fractions of volume for hot and cold streams respectively, (see Appendix A2)	–	–
g_c	constant of proportionality in Newton's second law of motion	– (1)	$\text{lb ft}/\text{h}^2/\text{lbf}$ (417×10^8)
H	flow enthalpy	W	Btu/h
h	specific enthalpy	J/kg	Btu/lb
Δh	latent heat of vaporisation	J/kg	Btu/lb
j	Colburn j -factor (dimensionless heat transfer coefficient)	–	–
k_h, k_c	number of hot and cold streams, respectively (see Appendix A2)	–	–
L	exchanger length	m	ft
m	parameter given by Equation (A5.2)	–	–

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$\dot{m}$	mass flux	kg/m ² s	lbf/ft ² h
$\dot{M}$	mass rate of flow	kg/s	lb/h
n	number of streams within a zone (see Section 7 and Equation (A2.6))	—	—
n_z	number of zones on composite curve (see Section 7)	—	—
Pr	Prandtl number, $c_p \eta / \lambda$	—	—
p	pressure	N/m ² (Pa)	lbf/ft ²
Δp	pressure drop	N/m ² (Pa)	lbf/ft ²
$\dot{Q}$	rate of heat transfer (heat load) of heat exchanger	W	Btu/h
s	element volume depth (see Appendix A1)	m	ft
T	fluid temperature	°C	°F
t_f, t_p	fin and plate thickness respectively (see Appendix A1)	m	ft
ΔT	temperature difference	K (°C)	°F
U	overall heat transfer coefficient	W/m ² K	Btu/ft ² h °F
V	total active heat transfer volume of exchanger	m ³	ft ³
ΔV	element volume (see Appendix A1)	m ³	ft ³
X	Martinelli parameter (see Equation (A4.2))	—	—
x	quality (fraction of mass flow that is vapour)	—	—
Δx	change in quality of two-phase stream in passing through heat exchanger	—	—
Δy	element width (see Appendix A1)	m	ft
Δz	element length (see Appendix A1)	m	ft
α	stream heat transfer coefficient	W/m ² K	Btu/ft ² h °F
β	film volumetric heat transfer coefficient	W/m ³ K	Btu/ft ³ h °F
η	dynamic viscosity	N s/m ²	lbf/ft h
λ	thermal conductivity	W/m K	Btu/h ft °F
ρ	density	kg/m ³	lb/ft ³
ϕ	two-phase multiplier (Equation A4.1)	—	—
ψ	fin efficiency (see Appendix A5)	—	—
Ω	total surface effectiveness (see Appendix A5)	—	—

Subscripts

1, 2	streams 1 and 2, respectively
<i>c</i>	cold stream
<i>F</i>	frictional quantity
<i>f</i>	finned surface
<i>fm</i>	fin material
<i>F</i>	frictional component
<i>G</i>	gaseous (vapour) phase
<i>h</i>	hot stream
<i>i</i>	i^{th} stream in composite curve zone (see Section 7)
<i>in, out</i>	values of parameter at inlet to and outlet from heat exchanger respectively
<i>L</i>	liquid phase
<i>lm</i>	logarithmic mean values
<i>m</i>	mean value
<i>min</i>	minimum value of quantity
<i>tp</i>	stream with two-phase flow
<i>uf</i>	unfinned surface
<i>w</i>	wall
<i>wa</i>	wall average
<i>z</i>	z^{th} zone on composite curve (see Section 7)

Dressing

—	average
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2. PURPOSE AND SCOPE OF THIS DATA ITEM

ESDU 97006 is one of a group of Data Items on the selection and costing of heat exchangers. The group provides information that can be used at the *process design* stage, allowing the *process designer* to evaluate costs that can be used in the selection of heat exchanger type and in the establishment of a budget price for the whole process. In the first Item, ESDU 92013¹, a procedure is given whereby the engineer can select feasible types of heat exchanger for a given duty; the selection is on the bases 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 cost approximately the respective types. The final selection between feasible types is made by the engineer usually on the basis of overall budget cost. It is recognised that the cost calculations in such procedures are of limited accuracy (20-50%) and it is important to consider further all types of heat exchanger for which the costs are similar to those of the exchanger with the minimum cost.

Three further Data Items provide specific guidance and cost data for particular exchanger types, namely shell-and-tube heat exchangers², air-cooled heat exchangers³ and plate-and-frame exchangers⁴. The objectives of this group of exchanger-type specific Items are as follows.

- To describe in more detail the construction and functionality of each type, discussing the advantages and disadvantages and the role of various alternative configurations of the given type.
- To explore in more detail the calculation of budget costs for the given type, investigating in particular the effect of variations from a "standard" configuration.

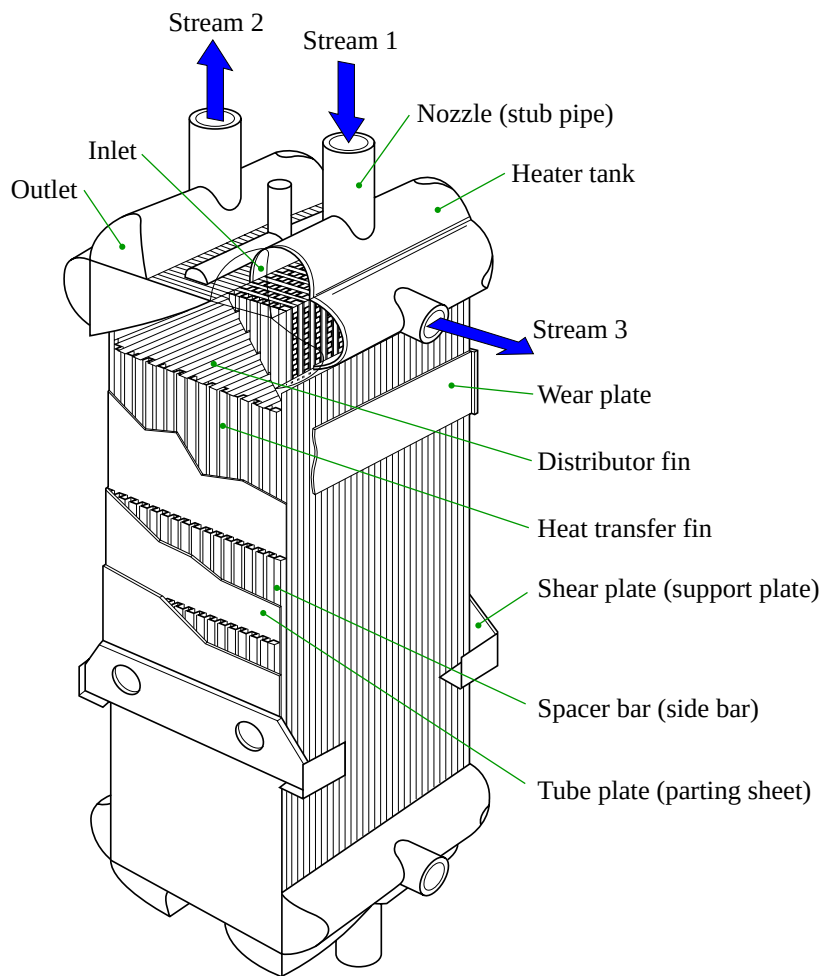
This Data Item extends this group with a presentation on plate-fin type heat exchangers. Plate-fin heat exchangers are part of an extensive family of compact heat exchangers but, in this Item, the focus is on brazed heat exchangers. Such exchangers are usually made from aluminium, though some reference will be made to construction from stainless steel. The plate-fin heat exchanger is compact and of similar basic form to that mass-produced for aircraft, automobile and other applications; however, the costing data of this Item relate to larger scale units built for industrial applications. It is not the intention of this Item to present a comprehensive treatise on plate-fin heat exchangers; the subject is well treated in the books by Kays and London⁵ and Webb⁶ and extensive information is also given in the article by Webb⁷ in the *Heat Exchanger Design Handbook*, in the *ALPEMA Standards*⁸ and in the *HTFS Guide (Taylor*¹²*)*.

However, despite the wide range of information available on heat transfer, fluid flow and pressure drop in plate-fin exchangers, there is a dearth of information about the costs of such exchangers. A limited amount of information is given in ESDU 92013 but it should be borne in mind that there are continuous developments in this area (in manufacturing technology, for example, particularly given the more widespread use of vacuum brazing) and these have a strong effect on costs. In the production of this Item, cost data have been generated with assistance from IMI Marston Ltd and have been first configured into the format used in the previous Data Items. That is, 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 in the exchanger from the hot to the cold fluid per unit time) and ΔT_m is the mean temperature difference, the latter including a correction where appropriate for the heat exchanger configuration. Also, a methodology is developed for the approximate design and costing of multi-stream plate-fin exchangers in which heat transfer and costs are estimated on a volumetric basis.

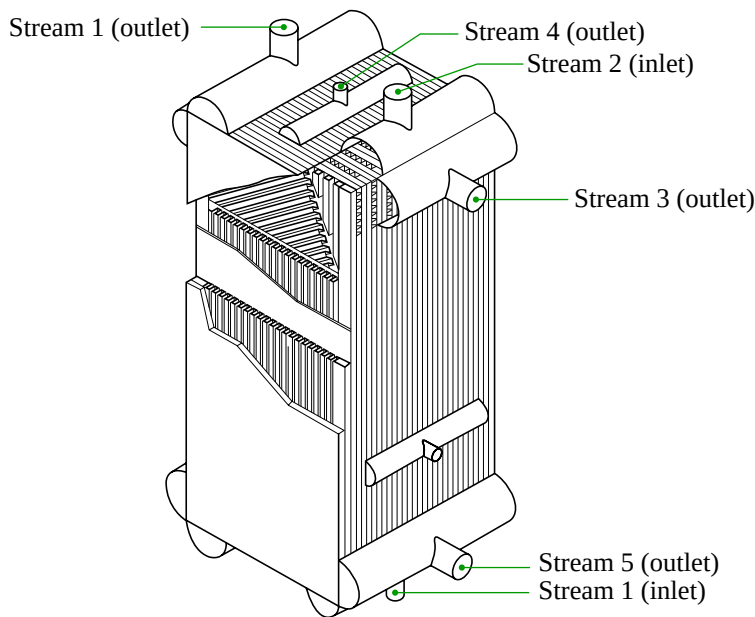
In what follows:

- Section 3 discusses the construction and configurations of plate-fin heat exchangers, including configurations and materials (Section 3.1), fin types (Section 3.2) and applications (Section 3.3).
- Section 4 gives a summary of the *C*-value costing method.
- Section 5 discusses the background to the costing assessment, the methodology for obtaining the data base on costs for plate-fin exchangers, specifying the "standard" fluids and also presenting the calculation procedure. The extension of the costing method to multi-stream applications is introduced in this section.
- Section 6 describes the variations of cost from the standard configurations described in Section 5.
- Section 7 discusses the multi-stream capabilities of plate-fin exchangers and describes an approximate design methodology (based on the techniques of process integration) that allows the assessment of cost savings due to multi-streaming. This methodology is illustrated in Sections 7.1 and 7.2 by worked examples.
- Section 8 describes the steps in the cost estimation methods.
- Section 9 gives Worked Examples illustrating how to use the data.
- Appendix A provides a full description of the bases for the methods behind the cost estimates.

3. CONSTRUCTION AND CONFIGURATIONS OF PLATE-FIN HEAT EXCHANGERS



Sketch 3.1a Typical brazed plate-fin construction (courtesy of IMI Marston)

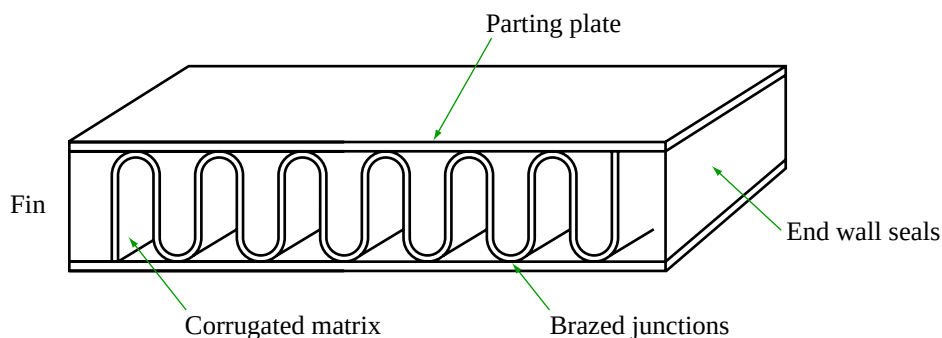


Sketch 3.1b Typical multi-stream unit (courtesy of ALPEMA)

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3.1 Configurations and Materials

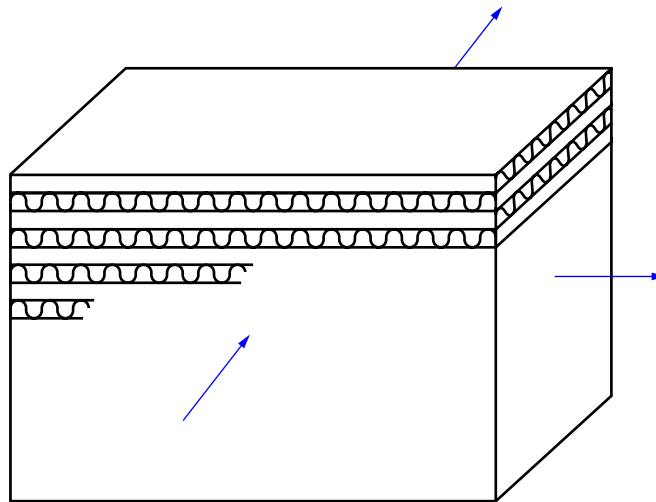
Typical plate-fin exchanger configurations are shown in Sketch 3.1. The component channels of a plate-fin heat exchanger are as illustrated in Sketch 3.2. The fluid flows between two *parting plates* in between which are mounted fins (plain-corrugated type in the illustration). The channel is sealed by side bars and is stacked with other channels, the whole assembly being brazed together using vacuum brazing (or in earlier days, salt bath brazing) technology. Each stream flows through many channels. The whole block (including distributors) is brazed together using the same vacuum brazing operation. Subsequently, the headers and nozzles are welded on.



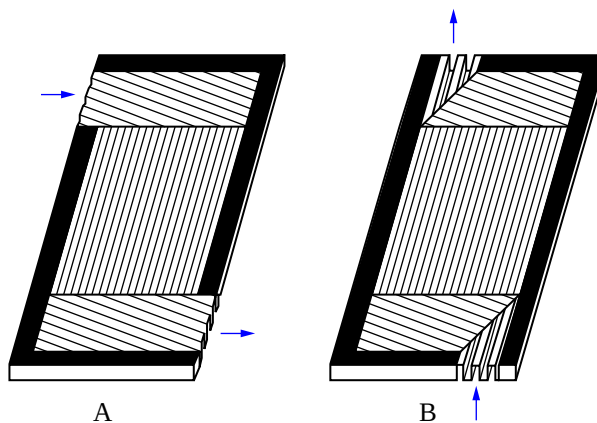
Sketch 3.2 Basic constructional unit of a plate-fin heat exchanger

A specific feature of plate-fin heat exchangers that is demonstrated below and makes them very attractive in conjunction with pinch-technology techniques is their capability of operation with multiple streams (see Section 7). For the purposes of illustration, Sketch 3.1 shows typical five stream units.

The simplest arrangement for flow distribution is to use crossflow as illustrated in Sketch 3.3. Here, the fluids flow in crossflow through the exchanger, passing between pairs of faces of the block, these pairs being at right angles to each other as shown. Crossflow heat transfer is disadvantageous in comparison with counter-current flow due to the reduction in effective temperature difference in the case of crossflow. Counter-current flow operation can be obtained by using diagonal corrugations for distribution and a counter flow operation can be arranged as shown in Sketch 3.4. Mixed counter and co-current flow can also be used in some circumstances. Further information on constructional details of plate-fin exchangers is given in the Standards of the Brazed Aluminium Plate-fin Heat Exchanger Manufacturers' Association (ALPEMA)⁸.



Sketch 3.3 Crossflow plate-fin heat exchanger



**Sketch 3.4 Distributor arrangements for counter flow plate-fin heat exchanger
(Courtesy of ALPEMA)**

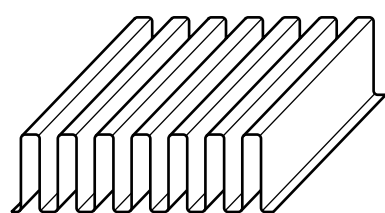
Plate-fin heat exchangers are most commonly made from aluminium. The maximum temperature depends on the alloy chosen. A maximum design temperature of 204°C is possible with alloys such as 3003, 3004, 5454, 5052 and 6061. However, economics often dictate the use of alloys 5083 or 5086 which have a limit of 65°C. The most common alloy for the matrix components (parting sheets, corrugations and side bars) is alloy 3003 with the headers and nozzles being made from alloy 5083. For temperatures in excess of 65°C, alloy 5454 or 6061 is normally specified. The minimum design temperature is -269°C. The maximum design pressure for brazed aluminium plate-fin heat exchangers varies from 0 to over 100 bar g. Brazed aluminium plate-fin heat exchangers are in general used with fluids that are clean, dry and non-corrosive to aluminium. Water is suitable as a cooling medium provided that it is contained in a closed circuit loop with corrosion inhibition. Mercury-containing fluids need to be handled with special care since an amalgam can be formed leading to deterioration of the material strength; if such fluids are to be handled, the manufacturer should be consulted.

Because of the small passage sizes and the changes of flow direction that occur, particulate fouling must be avoided by the installation of appropriate filters upstream of the exchangers.

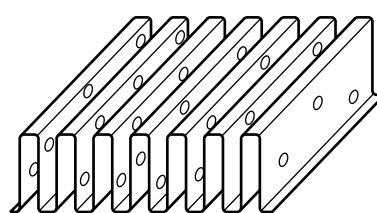
The problem of fouling in plate-fin heat exchangers is often perceived as a major potential difficulty. However, Clarke and Pritchard¹³ note that compact heat exchangers offer reduced susceptibility to fouling because of their uniform flow distribution, well-controlled wall temperature, high shear and low holdup volume. However, to achieve reliable performance, it is necessary at the initial design stage to adhere to certain guidelines when dealing with potentially fouling or corrosive fluids. Simple analysis of available fouling information provides the basis for rational fouling allowances typically resulting in a 10 per cent “operational allowance” (*i.e.* extra length) for liquid streams and condensing or wet gas streams (this corresponds to fouling resistances that are 10 to 20 times lower than those normally recommended for shell-and-tube heat exchangers). For dry gas streams, the assumption of a fouling resistance of zero is recommended. Industrial studies have demonstrated the desirability of having suitable upstream filtration for streams containing particulate material. When such filtration is not feasible, non-snagging yet undulating surfaces (that is, wavy fins) should be used. Carefully chosen and maintained velocities are required in order to keep shear stresses sufficiently high. Appropriate materials of construction, continuous pressure drop monitoring and (if required) cleaning in-place should be used if potentially fouling streams are processed.

3.2 Fin types

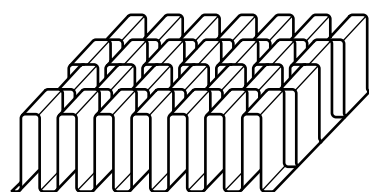
The various types of fin that are commonly used are illustrated in Sketch 3.5; they are as follows (in the preferred order from the point of view of heat transfer performance):



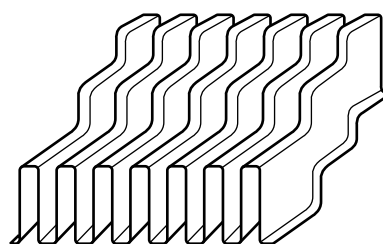
Plain



Perforated



Serrated



Herringbone

Sketch 3.5 Fin types (Courtesy of ALPEMA)

- (1) **Serrated Fins.** Here, the fins are offset by half a pitch, usually every 3 - 4 mm. This type of fin is frequently used especially for low pressure gas streams in air separation plants. Of all the fin types, it has the highest pressure drop, but also the highest heat transfer. Serrated fins are not recommended for fouling duties.
- (2) **Long Lanced Serrated Fins.** These are a form of serrated fins with a long serration pitch and offer some of the benefits of herringbone fins.

- (3) **Herringbone (Wavy) Fins.** This form of finning is also frequently used and can be smooth with waves of around 10 mm wave length and an amplitude of around 3 mm. It is often selected as an alternative to the serrated fin when there is a low allowable pressure drop. The pressure drop for this form of fin is high compared to perforated and plain fins but lower than that for standard serrated fins. Similarly, the heat transfer coefficients are high compared with perforated and plain fins but lower than those for serrated fins. On the other hand, herringbone fins show less increase in friction factor for a given increase in heat transfer than do serrated fins (this observation is true only at higher Reynolds numbers). Herringbone fins are usually chosen for relatively dirty fluids and are usually the fin of choice for hydrocarbon duties.
- (4) **Perforated Fins.** This type of fin has higher pressure drop and heat transfer than plain fins but less than serrated or herringbone fins. It is most frequently used for distributors and for boiling applications.
- (5) **Plain Fins.** These are applicable for general use and have the lowest pressure drop of any form of fin. Similarly, they also have the lowest relative heat transfer. Plain fins are not often used nowadays except for distributors.

A large variety of fin geometries is available. Within each type of finning, typical heights vary between 0.15 and 0.35 inches (3.8 to 8.9 mm) and fin frequency ranges from 6 to 22 fins per inch (236 to 866 fins per metre). By using carefully selected fin geometries, it is usually possible to match available pressure drops and achieve the required heat transfer rates in an optimal way.

3.3 Applications

Plate-fin heat exchangers have been used in a very wide variety of applications, and this range is continuously increasing. Their standard use has been for sub-zero temperatures (that is, for cryogenic systems) where their multi-stream capability and high surface areas per unit volume make them ideal. However, plate-fin exchangers have an unrealised potential for many other applications. Table 11.2 is produced from the ALPEMA Standards⁸ and lists the applications with their typical temperature and pressure ranges. The advantages of plate-fin exchangers include the following.

- (1) They are a highly compact form of exchanger and this is advantageous in uses where space is at a premium (on offshore platforms or inside an insulated cold box, for instance) or where the exchanger needs to be small to fit in with other equipment (for instance to be used as an internal reboiler within a distillation column).
- (2) Minimum temperature differences achievable with plate-fin exchangers are commonly much less than those associated with, say, shell-and-tube exchangers. Values of 1 K are typical of plate-fin units whereas a typical working figure for a shell-and-tube unit might be 10 K.
- (3) The possibility of multi-stream operation makes the plate-fin exchanger competitive with other units in many applications, and this range of applications could be widened.

The principal disadvantages of plate-fin units are as follows.

- (1) Limited temperature and pressure range. This range can, of course, be extended by using stainless steel rather than aluminium exchangers.
- (2) The exchangers are generally limited to clean fluids that do not corrode the material of construction (aluminium or stainless steel).

- (3) Care has to be exercised in lifting and handling the units, and excessive stresses (including cyclic stresses) must be avoided in operation. Manufacturers always give clear guidelines on the avoidance of these problems.

Thus, in considering new applications for these exchangers, their advantages must be assessed against any disadvantages.

4. SUMMARY OF COSTING METHODS

The C -value method is described in detail in ESDU 92013¹. Here, just a brief summary is given.

Assuming there are no heat losses from the exchanger, the rate of heat transfer between the two streams, $\dot{Q}$, is calculated from the expression

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

where $\dot{M}$ is the mass rate of flow of the respective stream and h is the specific enthalpy. The subscript h refers to the hot stream and the subscript c to the cold stream, and subscripts in and out refer to the inlet and outlet conditions, respectively. For a single-phase flow with a constant specific heat capacity, it follows that

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

where c_p is the specific heat capacity of the respective stream.

For a simple counter-current flow heat exchanger, the mean temperature difference, ΔT_m , is the logarithmic mean given by

$$\Delta T_m = \Delta T_{lm} = \frac{(T_{h, in} - T_{c, out}) - (T_{h, out} - T_{c, in})}{\ln \left[\frac{(T_{h, in} - T_{c, out})}{(T_{h, out} - T_{c, in})} \right]}, \quad (4.3)$$

where T is the inlet or outlet temperature of the respective stream, as identified by the subscripts. For the purposes of this Item, it is assumed that the flows are counter-current in nature (that is, with arrangements such as those shown in Sketch 3.4) and that the mean temperature difference can be calculated from Equation (4.3). The basic limitations and underlying assumptions in Equation (4.3) are discussed, for instance, in Reference 11. Also

$$\dot{Q} = UA\Delta T_m, \quad (4.4)$$

where U is the overall heat transfer coefficient and A the heat exchanger effective surface area.

In the C -value method, a cost per unit ($\dot{Q}/\Delta T_m$) is estimated. C -values are presented for the various fluid pairs for a range of fixed values of ($\dot{Q}/\Delta T_m$) and values in these tables can be interpolated between logarithmically. Thus, if a value C_1 is known for a value ($\dot{Q}/\Delta T_m$)₁ and the further value C_2 is known for a value ($\dot{Q}/\Delta T_m$)₂, then the C -value appropriate for the actual value ($\dot{Q}/\Delta T_m$) can be estimated from the equation:

$$C = \exp \left\{ \ln C_1 + \frac{\ln(C_1/C_2) \ln[(\dot{Q}/\Delta T_m)/(\dot{Q}/\Delta T_m)_1]}{\ln[(\dot{Q}/\Delta T_m)_1/(\dot{Q}/\Delta T_m)_2]} \right\}. \quad (4.5)$$