



Selection and costing of heat exchangers

Plate-and-frame type

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SELECTION AND COSTING OF HEAT EXCHANGERS: PLATE-AND-FRAME TYPE

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SELECTION AND COSTING OF HEAT EXCHANGERS: Plate-and-frame type

1. NOTATION AND UNITS

		Units	
		<i>SI</i>	<i>British</i>
A	heat transfer projected surface area of heat exchanger	m ²	ft ²
c_p	specific heat capacity	J/kg K	Btu/lb °F
C	cost per unit $\dot{Q} / (\Delta T)$	£/(W/K)	£(Btu/h)/°F
CR	cost ratio factor (see Section 8)	–	–
h	specific enthalpy	J/kg	Btu/lb
Dh	latent heat of vaporisation	J/kg	Btu/lb
$\dot{M}$	mass rate of flow	kg/s	lb/h
N_{TU}	number of transfer units	–	–
p	pressure	Pa	psia
$\dot{Q}$	rate of heat transfer (heat load) of heat exchanger	W	Btu/h
R_U	ratio of overall heat transfer coefficient for plate heat exchanger to that for 'equivalent' shell-and-tube exchanger	–	–
T	fluid temperature	°C	°F
DT	temperature difference	K (°C)	°F
U	overall heat transfer coefficient	W/m ² K	Btu/ft ² h °F
x	quality (fraction of mass flow which is vapour)	–	–
Dx	change in quality of a two-phase stream in passing through heat exchanger	–	–
α	heat transfer coefficient	W/m ² K	Btu/ft ² h °F
β	chevron angle (see Figure 7)	deg	deg
η	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
σ	surface tension	N/m	lbf/ft

Subscripts

<i>c</i>	refers to cold stream or to plate construction factor on <i>CR</i>
<i>crit</i>	critical (reference to pressure)
<i>g</i>	gasket material factor on <i>CR</i>
<i>G</i>	gaseous (vapour) phase
<i>h</i>	refers to hot stream
<i>in, out</i>	refer to values of parameter at inlet to and outlet from heat exchanger respectively
<i>L</i>	liquid phase
<i>LG</i>	refers to phase change
<i>lm</i>	logarithmic mean values
<i>m</i>	mean value or plate material factor on <i>CR</i>
<i>p</i>	pressure factor on <i>CR</i>
<i>PHE</i>	refers to plate heat exchanger
<i>s</i>	standard value
<i>sat</i>	refers to saturation
<i>smaller</i>	refers to smaller value of parameter in streams
<i>sp</i>	refers to stream with single phase flow
<i>spec</i>	refers to specified value
<i>S&T</i>	refers to shell-and-tube heat exchanger
<i>tp</i>	refers to stream with two-phase flow

2. PURPOSE AND SCOPE OF THIS DATA ITEM

ESDU 95007 is one of a group of Data Items on the selection and costing of heat exchangers. The objective of the group is to provide information which can be used at the *process design* stage, allowing the *process designer* to evaluate costs which 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 92013³, 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 carry out approximate costing of 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 that it is worthwhile considering further all types of heat exchanger for which the costs are rather similar to the exchanger with the minimum cost.

In the costing method used in the group of Items, 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 heat exchanger from the hot to the cold fluid per unit time) and ΔT_m is the mean temperature difference (the latter being corrected for heat exchanger configuration). The method is described in ESDU 92013³ and summarised here in Section 5. For a given exchanger type, a "standard" set of design parameters is chosen. However, costs are often very dependent on variations in these design parameters, even for a constant value of $\dot{Q}/(\Delta T_m)$. As an interim measure, simple correction factors are suggested for some heat exchanger types in ESDU 92013. However, it is obvious that the effect of variations is actually more complex and this led to the development of two further Data Items, on shell-and-tube heat exchangers⁴ and on air-cooled heat exchangers⁵.

Plate exchangers are increasingly used in a wide range of processes and offer many advantages in terms of convenience, compactness and economy. There are many forms of plate exchanger including the plate-and-frame type, spiral plate exchangers, plate-in-shell exchangers, fully-welded plate exchangers and fully brazed plate exchangers. To restrict the scope, this Item deals only with plate-and-frame exchangers (PHE's), though this is taken to include the semi-welded type where pairs of plates are welded together.

The objective of this present Data Item is to provide the process designer with the means of establishing approximate costs. Normally, the process designer will specify the duty of an exchanger; thermal design then follows, leading to the establishment of a configuration that can then be mechanically designed and, ultimately, costed. However, there is an increasing requirement for the process designer to establish the approximate cost of process equipment before this detailed design sequence is entered. This allows choices to be made between at an early stage of development of the process flowsheet.

There is little information in the literature on the costing of plate heat exchangers. Shell-and-tube exchangers are most commonly made from carbon steel, whereas the usual construction material for PHE's is stainless steel. Comparisons on the basis of cost per unit heat exchanger surface area are somewhat misleading since heat transfer coefficients are normally much higher in PHE's. Edwards and Stinchcombe¹ carried out comparisons of capital costs for carbon steel shell-and-tube exchangers and stainless steel PHE's for 54 different duties and concluded that, for the majority of cases, the shell-and-tube heat exchangers were significantly more expensive. This conclusion was borne out in the original development of the C -value method (Hewitt, Guy and Marsland²) and in its further development in ESDU 92013; for given values

of $\dot{Q} / (\Delta T_m)$, shell-and-tube exchanger costs could be typically several times those for PHE's. However, due to the limitations on availability of data, the range of C -values given in that Item is somewhat small and the aims of this current Data Item are therefore as follows:

- (1) To present a new data base for "standard" PHE's, operating at pressures up to 10 bar, with AISI-316 stainless steel plates and standard gasket materials - for instance Nitrile or EPDMTM. It is also assumed that the number of transfer units, N_{TU} , on the hot side has a "typical" value of 2.
- (2) To show the effect of variations from the "standard" cases on the C -values, and specifically to show the effect of pressure (16 and 25 bar), plate material (AISI-304 stainless steel and titanium), the use of special gaskets (for example VitonTM), exchanger construction and the use of alternative hot side values.

In making calculations of heat transfer coefficient, the fouling resistances built into the prediction codes used by the suppliers have been employed. It should be noted that much lower fouling resistances are normally assumed for plate-and-frame heat exchangers than for, say, shell-and-tube heat exchangers because the high turbulence generated in the flow between the plates inhibits the formation of fouling deposits.

The assumption is made that the heat exchanger has only *two streams* (one hot and one cold). However, it is worth noting that plate heat exchangers are capable of operating in a *multi-stream* mode. Such multi-stream operation is a useful option in striving for minimisation of process energy utilisation using the techniques of *process integration*.

In what follows:

- Section 3 discusses the configurations of plate-and-frame heat exchangers, including construction principles, plate types, gasket materials and pass arrangements,
- design methods are briefly summarised in Section 4 and Section 5 gives a summary of the C -value costing method,
- Section 6 discusses the methodology for obtaining the data base on costs for PHE's, specifying the "standard" fluids and also presenting the calculation procedure,
- Section 7 presents the results both for "standard" units and for the variations to them,
- Section 8 presents the methodology by which the data produced in this exercise can be used for cost calculation,
- Section 9 gives Worked Examples illustrating how to use the data and
- the basis of the costing calculation is laid out in Appendices; Appendix A describes the idealised fluid pairs and Appendix B summarises the calculation procedure by which entry data for manufacturer's methods were generated.

3. CONFIGURATIONS OF PLATE-AND-FRAME HEAT EXCHANGERS

3.1 Construction Principles and Operational Limits

The basic component of construction in a PHE is the gasketed plate which is illustrated in Figure 1. This consists of a pressed stainless-steel plate with ports in the corners. The most common flow arrangement is illustrated schematically in Figure 2. The first fluid enters at the bottom right and flows along a distributor channel formed by connecting successive "ports" in the plate (see Figure 1). The fluid passes from the distributor channel between each alternate gap between the plates, flows up the gap between the plates and then passes back along the exchanger through the channel formed by connecting the "ports" in the upper right hand corner of the plate. The second fluid enters at the top left hand corner, is similarly distributed and leaves through the bottom left hand corner*. Thus, a counter-current flow (and, of course, heat transfer) is set up with the respective fluids occupying alternate plate gaps. The plates are sealed with gaskets as shown in Figure 1. The constructional arrangement is illustrated in Figure 3.

Though many of the applications of plate-and-frame exchangers are for single phase fluids, the units are being increasingly used for condensation and evaporation duties. Figure 4 shows a plate exchanger used to concentrate sugar juice by evaporation, the heating being provided by condensing steam. The evaporator shown in Figure 4 has evaporation in rising flow but another design (Figure 5) uses alternate rising and falling flow. Falling film evaporation can also be incorporated into plate evaporators where special plates are designed to allow uniform distribution of the liquid at the top of the plate and avoidance of liquid film channelling in the flow down the plate.

The main advantages of the plate-and-frame exchanger are as follows.

- (1) The exchanger is an extremely flexible design in that it can be custom built from standard components which can be mass produced.
- (2) The exchanger can be changed in capacity with little cost by simply adding or subtracting plates as planned requirements dictate. Care has to be taken not to violate design constraints (*eg.* pressure drop) in making such changes.
- (3) Despite the relatively narrow passages involved, the plate exchanger often has a much lower propensity to fouling than does the tubular exchanger. This is because of the controlled flow distribution and also because of the high degree of turbulence within the exchanger.
- (4) Since the plate-and-frame heat exchanger can be dismantled, this facilitates the cleaning of the plates. Furthermore, the plate exchanger can be dismantled within its own "footprint" whereas bundle removal from, say, a shell-and-tube exchanger requires an additional space of length equal to the exchanger length.
- (5) The plate-and-frame heat exchanger is considerably more compact than tubular exchangers for the same duty. Typically, the PHE has around 100-150 m²/m³ surface area per unit volume, approximately double that for shell-and-tube exchangers but still an order of magnitude less than plate-fin and printed circuit exchangers.
- (6) Plate exchangers have smaller fluid inventories than the equivalent shell-and-tube exchanger.

PHE's are limited in operation by the various factors (temperature, pressure, chemical attack) associated with the gasket material. Typical maximum temperatures are 175°C and maximum pressures 25 bar (see Section 3.4 for more details). The gasket materials have to be compatible with the fluids and the plates and

* Less commonly, the fluid can enter and leave at diagonally opposite ports.

flow ports must be fully seated. Sealing problems can be overcome by welding or brazing all the plates together but this then vitiates the advantages of flexibility, cleanability, *etc.* Some amelioration of gasket problems can be achieved by welding pairs of plates together (though gaskets are still required for the flow ports and, of course, for the ports between the plate pairs for the other fluids).

3.2 Plate Types

Although standardisation and mass production of plates is a vital feature in the economy of plate-and-frame heat exchangers, the principal manufacturers have a wide range of plate types and forms within their portfolio as is exemplified by the range illustrated in Figure 6. The various types are described in more detail in what follows.

3.2.1 Chevron (herringbone) plates

In this form of plate (probably the most popular design) the corrugations are inclined and the plates are arranged in such a way that the corrugations on the adjacent plate (forming the flow channel) cross to give a large number of contact points, and hence considerable mechanical strength. The plates can have just two sets of opposed corrugations as shown in Figure 7 or they can have multiple chevrons as illustrated in Figure 8. By changing the chevron angle (β , see Figure 7) it is possible to change the heat transfer coefficient. For a given flowrate, plates with small angles have high heat transfer coefficient and pressure drop and plates with large angles have lower heat transfer coefficient and pressure drop. It is also possible to have control between the extremes by mixing plates of dissimilar chevron angle. Note that the amplitude of the corrugations (pressing depth) varies from one type of plate to another; the smaller the amplitude, the bigger the heat transfer coefficient and the pressure drop.

3.2.2 Ribbed plates

A typical ribbed plate is shown in Figure 9. A series of undulations are pressed into the plate, with pips pressed at their crests, and the pips make contact with those pressed on the adjacent plate. Typical plate spacings are in the range 3-7 mm. This type of plate has relatively low heat transfer coefficients but, commensurate with this, the pressure drop is also relatively low.

3.2.3 Double wall plate

If heat is to be exchanged between two fluids between which cross contamination must be prevented at all costs, one option is to use the double wall plate system as shown in Figure 10. If leaks occur due to corrosion or wear on one side of the exchanger, the liquids in the plates would not mix but, rather, would be discharged between the plate walls to atmosphere. The gap does, of course, introduce a significant thermal resistance to heat transfer which may be important in some circumstances. Typically, this resistance would add 15-20% to the plate area required.

3.2.4 Semi-welded plates

In this design, the plate pack consists of welded channels which alternate with traditional gasketed channels. A typical arrangement is illustrated in Figure 11. The aggressive fluid flows in the welded channels and the only gaskets in contact with it are the two circular porthole gaskets between the welded plate pair. Typically, these porthole gaskets are manufactured in highly resistant elastomer or non-elastomer material. The channels for the non-aggressive medium are sealed by traditional elastomer gaskets.

3.2.5 Tubular plates

In this form of plate exchanger, the plate pattern is such as to build tubular channels as is shown in Figure 12. This form of plate is suitable for "asymmetrical" duties where there are typically large differences in flow rates between the two sides. The construction is also suitable for low-pressure condensing and vaporising duties and for fibrous and particle-laden fluids.

3.2.6 Wide gap plates

An alternative to the tubular plate design is to use conventional plates but with wide gaps between them. This design is illustrated in Figure 13. The wider channels would carry liquids and products containing fibres or coarse particles or high-viscosity liquids which normally clog or cannot be satisfactorily treated in shell-and-tube heat exchangers. The free-flow product channels have a typical gap of about 16 mm. The plate configuration gives high turbulence and high heat transfer efficiency.

3.2.7 Graphite plates

Graphite plate exchangers have been developed for use with media which are normally too corrosive even for advanced metals and alloys. A typical plate is made of a composite material composed of graphite and fluoroplastics. The material is compressed into corrugated plates fitted with thin flat corrosion-resistant gaskets. A typical construction is shown in Figure 14.

3.3 Plate Materials

In principle, plates can be made from any pressable material. The most common materials used are:

Stainless steel AISI 316, AISI 304, Avesta 254 SMO™, Avesta 254 SLX™

Titanium

Titanium-palladium alloy

Incoloy™

Hastelloy™

Diabon F-100 (graphite).

Most plates are made from AISI 316 stainless steel and, clearly, the use of the more advanced materials implies additional cost. In the present study, the effects of using AISI 304 stainless steel or titanium rather than AISI 316 stainless steel have been investigated.

3.4 Gasket Materials

A great deal of work has been done to develop gasket design and gasket materials, a crucial factor in plate-and-frame heat exchangers. Modern gaskets are often of the clip-on design as illustrated in Figure 15. In older units, adhesive was used to maintain the gasket in the correct position in the plate; adhesive is used nowadays mainly for very large units.

A wide variety of materials have been used for gaskets but medium Nitrile rubbers and EPDM™ are the most common. Viton™ gaskets are employed where the fluids involved would attack Nitrile or EPDM™ gaskets; Viton™ gaskets are more expensive.

Nitrile rubbers will withstand maximum temperatures of 135°C, EPDM™ gaskets 160°C and Viton™ gaskets are acceptable up to 175°C. In some applications, graphite-based gaskets can be used up to 250°C. The temperature range can also be extended using fully brazed or welded PHE's but these are outside the scope of the present Data Item.

3.5 Pass Arrangements

Plate-and-frame heat exchangers are normally operated in counter-current flow. Multiple pass arrangements are sometimes used but are becoming ever less common as the manufacturers develop plates with larger and larger N_{TU} values (see Section 4). In principle, each fluid can enter at one end of the exchanger and leave at the other (the "Z-arrangement"). In practice, however, wherever possible, all connections are made at one end of the exchanger (the "U-arrangement") as illustrated in Figures 2 and 3. This avoids the necessity for dismantling pipe connections before taking the exchanger apart for cleaning the plates.

4. DESIGN METHODS

The process specification for a plate-and-frame heat exchanger would consist of information about the process streams (temperatures, fluid physical properties, pressures) and about the target for heat transfer between the streams (usually expressed in terms of a heat load, $\dot{Q}$). The first step is to select the appropriate configuration (and in particular the type of plate to be used - see Section 3.2) and the design proceeds to match the specified available pressure drop to achieve the lowest cost exchanger to meet the heat transfer requirements. This is done by adjusting the size and number of plates and the plate configuration (plate spacing, corrugation angle, etc.). It is beyond the scope of this present Data Item to go into detail on this process. However, there is one aspect of the design process which will be mentioned here because of its widespread use within the industry producing this form of exchanger. The *number of transfer units*, N_{TU} , for the hot and cold streams can be defined by the following equations (for single phase fluids)

$$(N_{TU})_h = \frac{UA}{\dot{M}_h c_{p,h}} = \frac{(T_{h,in} - T_{h,out})}{\Delta T_{lm}} \quad (4.1)$$

and

$$(N_{TU})_c = \frac{UA}{\dot{M}_c c_{p,c}} = \frac{(T_{c,out} - T_{c,in})}{\Delta T_{lm}}, \quad (4.2)$$

where U is the overall heat transfer coefficient, A the heat exchanger surface area (based on the projected areas of the plates), $\dot{M}_h$ and $\dot{M}_c$ the mass rates of flow of the hot and cold streams respectively, $c_{p,h}$ and $c_{p,c}$ the mean specific heat capacities of the respective streams, $T_{h,in}$ and $T_{h,out}$ the inlet and outlet temperatures of the hot stream, $T_{c,out}$ and $T_{c,in}$ the outlet and inlet temperatures of the cold stream and ΔT_{lm} the logarithmic mean temperature difference given by:

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

Note that in the ESDU Items on the Effectiveness – N_{TU} method, N_{TU} is defined as $UA / (\dot{M}c_p)_{smaller}$, where $(\dot{M}c_p)_{smaller}$ is the smaller of the $\dot{M}c_p$ products for the hot and cold streams respectively. In this Item, the practice of plate heat exchanger manufacturers is followed whereby separate N_{TU} 's are defined for each stream.

Heat exchanger duties which have high N_{TU} values are often in the industry called "thermally long" or "high theta" duties. Commonly, the range of N_{TU} values used in plate-and-frame heat exchangers is 0.5 to 3, with 2 being a reasonable average value. High N_{TU} duties can be met using modern, high efficiency plates which can reach N_{TU} values typically up to 7; this minimises the number of cases where it is necessary to have multiple pass operations which can also be used to deal with high N_{TU} duties.