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Selection and costing of heat exchangers

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SELECTION AND COSTING OF HEAT EXCHANGERS

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SELECTION AND COSTING OF HEAT EXCHANGERS

1. NOTATION AND UNITS

		<i>Units</i>	
		<i>SI</i>	<i>British</i>
A	heat transfer surface area of heat exchanger	m^2	ft^2
c_p	specific heat capacity	J/kg K	$\text{Btu/lb } ^\circ\text{F}$
C	cost per unit $\dot{Q}/\Delta T$	$\text{£}/(\text{W/K})$	$\text{£}/([\text{Btu/h}]/^\circ\text{F})$
E	heat exchanger effectiveness defined in Equation (6.7)	—	—
F_T	logarithmic temperature difference correction factor	—	—
h	specific enthalpy	J/kg	Btu/lb
$\dot{M}$	mass rate of flow	kg/s	lb/h
N_{TU}	number of transfer units	—	—
P	flow heat capacity ratio defined by Equation (6.6)	—	—
$\dot{Q}$	heat load of heat exchanger	W	Btu/h
$\dot{Q}_{max}$	maximum possible heat load	W	Btu/h
R	flow heat capacity ratio defined by Equation (6.9)	—	—
R_K	$(\dot{M}c_p)_h/(\dot{M}c_p)_c$	—	—
T	stream temperature	K	$^\circ\text{F}$
U	overall heat transfer coefficient	$\text{W/m}^2 \text{ K}$	$\text{Btu/h ft}^2 ^\circ\text{F}$
ΔT	temperature difference	K	$^\circ\text{F}$

Subscripts

c, h	refer to cold stream and hot stream, respectively
fg	refers to phase change between liquid and gas
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 streams, respectively

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lm	logarithmic mean value
m	mean value
1, 2	value in stream number 1 (hot stream) and stream number 2 (cold stream), respectively

2. PURPOSE AND SCOPE OF THIS DATA ITEM

ESDU has issued a number of Data Items relating to heat exchangers including Items on heat exchanger effectiveness^{6, 11}, baffled shell-and-tube exchangers⁷ and process integration with particular reference to the energy recovery in heat exchanger trains^{5, 8, 9}. Having established the overall heat recovery network the next step (preceding these detailed design considerations) is to select the type of heat exchanger. For any given duty, several types will often be suitable; in this case, the final selection will normally be based on economic factors (that is the capital and operating costs). Ideally, the operating costs (mainly maintenance and pumping costs) and capital costs should be balanced to give a minimum annual cost for the exchanger. More usually, the “allowable pressure drop” across the exchanger is specified by the process designer and this allows no scope for optimisation, the capital cost being the only consideration. In the present Data Item it has been assumed that the capital cost will be the main consideration in making the final selection.

The criteria on which selection of heat exchangers will be made are considered in Section 3. In Section 4 the main types of heat exchanger are introduced, indicating the advantages and disadvantages of each type and providing the basis for an initial selection between types following the guidelines of Section 5.

In Section 6, an approximate method of costing is presented, namely the “C-value” method, where C is the cost per unit $\dot{Q}/\Delta T_m$. Tables of values are presented for shell-and-tube, double-pipe, air-cooled, plate-fin, gasketed-plate, welded-plate and printed-circuit heat exchangers. Examples are given in Section 7 that illustrate the use of the methodology. It should be stressed that the method is a very approximate one and that, where costs are calculated to be similar for two or more feasible designs further investigation of these alternatives should be pursued. Furthermore, and particularly for tubular equipment, considerable variation of C occurs depending on the specific design details.

3. SELECTION CRITERIA

The selection process takes account of a number of factors, all of which are related to the specific application (therefore no general rules can be given). These factors include the following.

- (1) **Thermal and hydraulic requirements.** The amount of heat to be exchanged, the fluid inlet and outlet temperatures and the allowable pressure drop (or pumping power) are often specified as a result of the overall process optimisation. Clearly, any selected heat exchanger must be capable of meeting this specification.
- (2) **Compatibility with fluids and operating conditions.** The materials of construction of the selected exchanger must be able to contain the fluids on the respective sides without excessive corrosion. The exchanger must be capable of being designed and constructed to withstand the stresses due to fluid pressures and temperature differences (thermal stresses).
- (3) **Maintenance.** The characteristics of the process streams should be carefully considered to assess the requirements for cleaning (mechanical or chemical) and periodic replacement of all or part of the unit. Ease of modification could also be an important factor if process conditions are likely to change.
- (4) **Availability.** Project timescales will often dictate the use of standard designs which can be delivered rapidly. There may also be limitations in the available design methods for some of the possible units.
- (5) **Economic factors.** Obviously, if several possible heat exchanger types meet the physical requirements listed above, the final choice must be based on economics. For fixed pressure drop (pumping power), the main economic factor will be capital cost. However, it should be noted that there may be a trade-off between pumping power and capital cost which should certainly be borne in mind.
- (6) **Space and weight.** In a number of applications the volume occupied by a heat exchanger is a vital consideration while in others, the weight is a crucial factor. In some applications, both volume and weight are important. Examples of such limitations are found in the aerospace industry where limitation of payload is economically and technically important and offshore structures where there is a large incentive to reduce the size and loading of platforms.
- (7) **Fouling.** The potential for fouling with the respective fluids must be carefully assessed and the exchanger must be able to operate for the required period under the predicted conditions.

4. TYPES OF HEAT EXCHANGER

A vast range of heat exchanger types is produced and it is impossible to cover all of them in detail here. Rather, the principal types will be described, indicating their common areas of use and their main advantages and disadvantages. The types covered are listed in Table 9.1 and this table should be the focus of the initial selection process.

4.1 Shell-and-tube Heat Exchangers

The shell-and-tube heat exchanger is the most commonly used type in the process industry. In the power industry a typical case is for feed water heaters; power industry condensers (though normally tubular) have a somewhat different construction. The reason for its wide application is that it can be designed to operate

over the full range of pressures and temperatures encountered in the process industry. The shell-and-tube heat exchanger can also be constructed from a wide range of materials, there exist well established design methods and mechanical codes for the units and there have been many decades of experience in their application leading to a high degree of confidence. The most commonly used types of shell-and-tube heat exchangers are illustrated in Figure 1 and are as follows:

- (1) **Fixed tubesheet.** The construction is simple and economical and the tube bores can be cleaned mechanically or chemically. However, the outside surfaces of the tubes are inaccessible except to chemical cleaning. If large temperature differences exist between the shell and tube materials, it may be necessary to incorporate an expansion bellows in the shell, to eliminate excessive stresses caused by expansion. Such bellows are often a source of weakness and failure in operation. In circumstances where the consequences of failure are particularly grave, one of the following types should be considered.
- (2) **U-tube type.** Heat exchangers of this design are usually more expensive than the fixed tubesheet design, but the differences are often small and U-tube units can be cheaper at high pressure. The tube bundle can be removed from the shell for cleaning or maintenance. Internal cleaning of the tubes by mechanical means is difficult and it is usual to apply this type where the tube side fluids are clean. Thermal expansion is absorbed by the U-bends. The number of tubesheet joints is reduced, but tube replacement is more difficult than with straight tube exchangers since it may be necessary to remove a number of tubes before a tube in the centre of the bundle can be replaced. A disadvantage of the U-tube design is that it cannot normally have pure counter-flow and this gives a consequential penalty on effective ΔT_m . Occasionally, U-tube designs are employed in which counter-current flow is achieved by using a longitudinal baffle in the sheet. Such baffles present difficulties with leakage or, if they are welded, in bundle removal.
- (3) **Floating head type.** This type of heat exchanger is suitable for the rigorous duties associated with high temperatures and pressures. The tube bundle consists of straight tubes secured into tubesheets at each end. The tubesheet at the tubeside fluid inlet end is of larger diameter than the shell and is sealed in a similar manner to that used in the fixed tubesheet design. The other (remote) tubesheet is of slightly smaller diameter than the shell, allowing the bundle to be slid through the shell. Various methods can be used to seal the remote end; in the version shown in Figure 1, the rear end tubesheet protrudes through the end of the shell and is sealed off using a cover as shown in Figure 1. To seal the cover to the tubesheet, it is necessary to compress a gasket using bolts. These bolts pass from the cover plate through a ring which is on the other side of the tubesheet. Since the ring is of larger diameter than the shell it has to be split into two in order to bring it into position to take the bolts. Hence its name – the “split backing ring”. The cover and its mounting is termed the “floating head” since it is free to move relative to the shell. The end of the shell is sealed off in the normal way. It should be noted that the floating head is fitted to the tubesheet remote from the channel end and the cover is held in position by a split backing ring. The floating head tubesheet is only slightly smaller in diameter than the inside of the shell and the floating head can only be assembled after the tube bundle is in position in the shell. Conversely, the floating head must be dismantled before bundle removal. The tube bundle is easily removed for cleaning or maintenance and the tubes can be mechanically cleaned both inside and outside. The floating head design is more expensive (typically of order of 25% for carbon steel construction) than the equivalent fixed tubesheet design.

The types shown in Figure 1 are the three most commonly used, but many other configurations are possible and are used from time to time. The combinations are usually described in terms of a standard nomenclature devised by the Tubular Exchangers Manufacturers Association (TEMA) of the USA. The TEMA nomenclature is defined in Figure 2. It will be seen that the three types shown in Figure 1 are (in terms of the TEMA nomenclature) designated BEM, BEU and BES. The first letter describes the front end type, the second letter the shell type and the third letter the rear end type. It will also be seen from Figure 2 that there are several alternative closure schemes for floating head exchangers. In addition to the S-type described above there is also the option of sealing using packing devices (P and W types) or of having the floating head small enough to pull the whole unit through the sheet (T-type).

4.2 Double-pipe Heat Exchangers

In its simplest form, the double-pipe heat exchanger consists of a single inner tube mounted concentrically inside another tube. Heat transfer takes place between fluids passing (almost always in counter-current flow) through the inner tube and through the annular gap between the inner and outer tubes. In commercial practice, the term “double-pipe exchanger” has been extended to cover a range of modular constructed units that may have multiple tubes (sometimes called “multi-tube” exchangers) and, particularly for smaller units, may have longitudinal fins on the outer surfaces of the tubes. This type of commercial unit is typified by that shown in Figure 3, and Figure 4 shows the nomenclature for a variety of end closures and shell nozzle arrangements that can be used.

Double-pipe heat exchangers have the advantage that they offer true counter-current flow ($F_T = 1$ – see Section 6). For shell sizes of 100 mm (4 inch) and above, segmental baffles or rod-type baffles are commonly employed and these units become rather similar to a single-pass counter-flow shell-and-tube heat exchanger. They are sometimes described as jacketed U-tube exchangers. For shells with sizes less than 100 mm, there are normally no baffles, but the tubes are supported using special spider supports. Standard designs are available up to 400 mm (16 inch) nominal bore; larger units are available for special duties.

In finned double-pipe exchangers, baffles are not necessary since the tubes are inherently self-supporting within the unit. Intermeshing of the fins is prevented by wrapping strips of stainless steel or other suitable material around the outside of the fins. A further stainless steel band is wound round the outside of the bundle to hold it together and for ease of construction. With both the plain tube and finned tube multi-tubular designs, peripheral baffles or rings may be necessary to block the leakage area between the bundles and the shell. As will be seen, if axial flow is maintained in the units, then the problems of mixing and leakage associated with baffled units can be avoided. This makes the axial flow units (without baffles) particularly attractive for heat transfer at low Reynolds numbers.

The choice between plain and finned double-pipe heat exchangers is usually made ultimately on the basis of cost for a particular duty. In general, on the smaller units (< 150 mm or 6 inch shell), the finned tube type is more economic where the heat transfer coefficients are relatively high on the tubeside and where there is a need to enhance the surface on the shellside. Roughly speaking, the rule is that such units are economic when the outside coefficient is less than 25 per cent of the inside coefficient. As the unit size increases (> 100 mm or 4 inch) the bare tube, multi-tubular, units become more and more economic.

4.3 Plate-and-frame Heat Exchangers

There are a number of forms of exchanger in which the flows pass between plates which are mounted in a frame. Such exchangers include the gasketed-plate exchanger, the welded-plate pair exchanger and the brazed-plate exchanger.

The gasketed-plate heat exchanger is illustrated schematically in Figures 5 and 6. The gasketed-plate exchanger is an extremely flexible design in that it can be custom built from standard components that can be mass produced. In the normal range, the plates are made from stainless steel, but titanium plates are commonly available and other materials can be used if necessary. The plates are corrugated or embossed to improve rigidity, to control plate spacing and to improve heat transfer. Corrugation and embossing increases the pressure drop per unit length but the overall pressure drop in the exchanger is not necessarily increased for a given duty. The nature of the design is such that it can be optimised to give a highly efficient heat transfer performance, although this is often at the expense of a relatively high pressure drop. The gaskets are commonly made from ethylene-propylene or nitrile elastomers, but a variety of other compressible materials is available, selected to cope best with the corrosion or temperature problems involved. These include neoprene, Viton®, hypalon and butyl elastomer.

Another major advantage of the gasketed-plate exchanger is that it can be increased in size with little cost by simply adding further plates as plant requirements dictate.

The gasket problem can be overcome in some cases by welding pairs of plates together, thus allowing fluids that are incompatible with the gasket materials to flow between the welded plate with the other (compatible) fluid flowing between successive pairs of plates. This still leaves the problem of sealing the plates but there is a much wider choice of material for the simple O-ring seal required in this case.

In the brazed-plate exchanger, the construction is similar to that shown in Figure 5, but a brazing procedure is used in place of gaskets.

Clearly, the major limitation in the plate exchanger is in its maximum operating temperature and pressure and in the integrity of sealing, which reflects the limited range of elastomer gasket materials available.

4.4 Spiral Heat Exchangers

The principle of the spiral heat exchanger is illustrated in Figure 7. The spiral surfaces are separated by raised bosses and sealed with two end plates. In the counter-current design, the hot fluid enters at the centre of the unit and flows from the inside outwards. The cold fluid enters at the periphery and flows towards the centre.

Alternative designs of spiral heat exchanger may be used for condensing and evaporating duties as sketched in Figure 8.

- (i) A vertically mounted spiral exchanger may have condensation in downflow on one of the spirals with the coolant passing from the periphery towards the centre.
- (ii) The outer part of the spiral as described in (i) can be used to subcool the condensate. In this case, the outer part is sealed as in Figure 6.
- (iii) As in (i), a vertical mounted exchanger can be used for evaporation with steam condensing in the sealed spiral with upflow evaporation in the other spiral. Such units are useful for in-column reboilers. A similar arrangement (with cooling water in the sealed spiral) can be used for reflux condensation.

Advantages claimed for this unit are that the overall heat transfer coefficients are in general higher than those for shell-and-tube units, that the swirling of the flow scrubs the surface and reduces fouling and that the outer spiral may carry the cooler fluid and give a reduction in the insulation requirement. Spiral heat exchangers are often used for suspensions, especially of fibrous materials. The main disadvantages of spiral

heat exchangers are that chemical rather than mechanical cleaning is normally necessary and that repair is very difficult. If end gaskets are used, mechanical cleaning is possible, but this leads to problems of compatibility between gaskets and fluids.

4.5 Fully-welded-plate Heat Exchangers

The fully-welded-plate exchanger overcomes the gasket problem encountered with gasketed plate-and-frame exchangers. A variety of commercial designs are available; one such design is illustrated in Figure 9. The operating pressure of such units is normally limited to around 60 bar.

However, higher pressures (equivalent to those applicable in shell-and-tube exchangers) can be achieved by mounting the plates within a shell. Very large areas are feasible ($> 1000 \text{ m}^2$) and relatively high fluid temperatures can be accommodated.

Typically plates in such units are up to 6 m long and up to 2 m wide, and the large size compensates to some extent for the increased cost of sealing. As shown in Figure 9, several packs of plates can be put in parallel between the same headers. The specific form of welded-plate exchanger illustrated in this figure forms tube-like passages between plates for one of the streams (hence the designation “tubeside” as shown).

Costs for this form of exchanger are obviously higher than those of the gasketed-plate exchanger but, by using large plate sizes, some manufacturers claim that the increase is relatively small.

A limitation on this form of exchanger is the differential pressure between the two fluids; a typical figure given by a manufacturer for maximum differential pressure is 3.0 MPa.

An often-encountered problem with this form of exchanger is that chemical rather than mechanical cleaning of the plates is needed. However, more recent designs such as those illustrated in Figure 9 can be manufactured to allow mechanical cleaning on at least one side of the plate.

4.6 Lower Temperature Heat Exchangers for Waste Heat Recovery

Recovery of waste heat from combustion gases and other gaseous exhaust streams is of vital importance in energy conservation. The waste heat may be transferred either to another gas (for example the inlet air to a combustion process) or to a liquid (for example the feedwater in a boiler system). Gas-to-gas exchangers (typically flue gas to air) are commonly called ‘recuperators’ whereas gas-to-liquid exchangers (typically flue gas to water) are commonly called economisers. Such “economiser” heat exchangers are available in a wide variety of forms and can generally be classified into those operating at low temperatures (up to 250°C, say) and those operating at higher temperatures (up to 1000°C, say). A wide variety of heat exchangers has been used for this purpose and it is beyond the scope of this present Data Item to give a detailed discussion. The user is referred to the articles by Hewitt *et al*¹ and Hinchley³ for further information. In this present Section, low temperature economiser heat exchangers are discussed and, in Section 4.7, a brief discussion is given of high temperature economiser heat exchangers.

Low temperature waste heat recovery heat exchangers are exemplified by the units shown in Figure 10. The low temperature allows a light construction. The tubes are either plain or finned. The cold fluid can either be air or water and the units can be of the single-pass (crossflow) type or of the multi-pass type as illustrated. For the single-pass type, the effective temperature difference is reduced and the multi-pass type may be more economic, despite increased construction costs.

4.7 Higher Temperature Heat Exchangers for Waste Heat Recovery

Waste heat recovery heat exchangers operating at high ($> 250^{\circ}\text{C}$) temperature take a wide variety of forms depending on the application. High temperature gas convective banks are often constructed in site-built refractory casings but it is often cheaper to use steel units constructed in modular form as illustrated in Figure 11. Typically operating temperatures for such units range up to 1000°C . Another form of unit used is the so-called “jacketed stack” in which flue gases flow up the stack and the air flows in an annular jacket around it.

4.8 Glass-tube Heat Exchangers

Glass-tube exchangers are suitable for a variety of duties, including recovery of heat from aggressive exhaust gases at temperatures up to and around 400°C . In a typical design, the glass tubes are mounted horizontally between perforated stainless steel end-plates into which they are securely sealed. In this design, plastic spacers hold the tubes in the correct position.

The end-plates, framework and flange enclosure are fabricated from chemical-resistant stainless steel. All-glass construction is also possible.

4.9 Plastic-tube Heat Exchangers

Here, in one design, plastic tubes are mounted horizontally between perforated stainless steel end plates and plastic spacers hold the tubes in their correct position. A typical plastic used in construction would be PTFE. The units are suitable for heat recovery from aggressive exhaust gases at temperatures up to 100°C .

4.10 Heat-pipe Heat Exchangers

The heat-pipe heat exchanger is illustrated in Figure 12. Heat is transferred from the hot gas to the evaporation section. The vapour flows along the heat pipe to the condensation section, releasing its heat to the cold gas. The liquid returns along the wick to the evaporation section. A series of heat pipes can be mounted as shown in Figure 12 and, in this form, the heat-pipe heat exchanger operates in a mode close to counter-current heat exchange.

Heat-pipe heat exchangers are normally operated with the pipes in the vertical position (condensation section at the top) or with a slight inclination from the horizontal. Most heat-pipe systems are operated with water as the heat pipe fluid. Higher temperatures are possible with organic fluids, though thermal degradation of these can be a problem. Very high temperatures (up to 1500°C) are feasible if a liquid metal such as sodium is used as the fluid but this is often too high a range for most heat recovery opportunities.

4.11 Air-cooled Heat Exchangers

Although air-cooled heat exchangers are essentially heat rejection rather than heat exchange devices, they have great importance in process plant and information on their cost is important in considering the overall thermal integration of such plant. Note that air-cooled heat exchangers are often primarily crossflow units and hence have lower thermal effectiveness¹¹ than counter-current flow exchangers. The penalty for crossflow can be minimised by using multiple tubeside passes¹¹. With air cooling, the possibility of process fluid leakage into the cooling water circuit is eliminated. Ideally, they should be designed so that the minimum process side temperature is no less than 15 K over the maximum summer temperature. However, a design air temperature of 20 to 25°C is often employed since plant can usually tolerate short periods at particularly high temperatures; this approach can reduce the cost of such units considerably.

Typical air-cooled heat exchanger units are illustrated in Figure 13 for forced-draught and induced-draught operation, respectively. The induced-draught mode of operation tends to give a uniform flow distribution. Circulation fans, which have to be designed to operate in a warm air stream, give a consequent increase in required power.

Another design is the A-frame configuration as shown in Figure 14. This design is often used in vacuum steam condensation. The air-cooled heat exchanger is often economic as an alternative to water cooling, where water costs are high. The perceived environmental impact is lower than that of large cooling towers with their associated vapour plumes. However, fan noise can sometimes be a problem.

Typical air-cooled heat exchanger tubing would be 1 inch (25.4 mm) outside diameter carbon steel tube wound with 2 inch (57.1 mm) outside diameter aluminium fins, with 11 fins per inch (433 fins per metre). For such units, the bundle costs represent around 60 per cent of the total unit cost.

4.12 Plate-fin Heat Exchangers

Plate-fin exchangers are noted for small size and weight. Typically, these units have a total surface area of $500 \text{ m}^2/\text{m}^3$ of volume; this compares, for instance, with a shell-and-tube unit where the surface area per unit volume is of the order of 40 to $70 \text{ m}^2/\text{m}^3$. Plate-fin heat exchangers with surface areas of $1000 \text{ m}^2/\text{m}^3$ are sometimes employed in the process industry and ones with surface areas of $1800 \text{ m}^2/\text{m}^3$ are used in the aircraft industry. For these reasons plate-fin heat exchangers find applications in aircraft, automobiles, rail transport, offshore platforms, *etc.* Their ability to carry several streams, normally up to 7 but occasionally up to 12 or more, allows process integration in certain industrial processes, establishing them firmly in air separation processes and other cryogenic systems.

The process streams pass between metal plates which are held together by corrugations (fins) that provide extended surfaces and also enhance the heat transfer coefficient. The edges of the plates are sealed by edge bars. The construction is shown in Figure 15. The material of construction is typically aluminium but stainless steel and nickel alloy units are well established in the aircraft industry and these are now entering the process industry.

Various forms of corrugation are used and a range of more common types is shown in Figure 16. Such heat exchangers have a very large surface area per unit volume and show best advantage when low temperature differences apply as typically found in cryogenic systems and hydrocarbon dewpoint control systems where temperature difference is linked to compressor power.

Plate fins are susceptible to blockage by particulate matter and it is recommended that filters should be placed at the inlet of such units to remove particles greater than 0.3 mm diameter. Design pressures up to 100 bar can be accommodated in aluminium units and up to 200 bar or more in stainless steel. Single units up to $1.2 \text{ m} \times 1.2 \text{ m} \times 6.2 \text{ m}$ are manufactured in aluminium and units up to $0.6 \text{ m} \times 0.6 \text{ m} \times 2 \text{ m}$ are manufactured in stainless steel. Multiple units can be produced by manifolding.

4.13 Printed-circuit Heat Exchangers

Printed-circuit heat exchangers (PCHE's^{*}) are plate heat exchangers in which the plates are manufactured by techniques similar to those used for electrical printed circuits. Figure 17 illustrates the principle; the plates are chemically milled and then diffusion bonded to make the heat exchanger core. This allows relatively fine channels (typically 1 to 2 mm mean hydraulic diameter) to be created with great precision and allows the construction of PCHE cores that are 5 to 10 times smaller than tubular heat exchangers of

* At the present time (1992) there is only one manufacturer for these units.

equivalent performance. Heat transfer surface area per unit volume is of the order of 500 to 750 m²/m³. PCHE's operate over a wide range of pressure and temperature. Surprisingly, pressure drop in PCHE's is not often a limiting factor since the increased pressure gradient is offset by the shorter channel lengths.

Because of the small diameter of the channels, PCHE's are not suitable for duties with any significant amount of fouling.

4.14 Scraped-surface Heat Exchangers

For heat transfer in systems in which crystals or wax are formed as a result of the heat extraction, fouling of the heat transfer surface is endemic. For such systems (and also for highly viscous fluids) it is often best to use a heat exchanger in which a rotating blade traverses the surface, scraping off the deposits. Though such exchangers would only be used under rather special circumstances, they are an important possible solution to some very difficult problems.

4.15 Regenerators

The heat exchangers described above so far are often referred to as *recuperative* exchangers; they recover heat from one fluid stream by direct heat transfer (through a heat transfer surface separating the streams) to the other stream. In *regenerative* heat exchangers, heat is extracted from the hot stream into a solid matrix which is subsequently exposed to the cold stream, leading to the heating of the cold stream.

4.15.1 Cyclic regenerators

The classical form of regenerative heat exchanger is the Cowper Stove in which a matrix of bricks is first heated by flue gases from a process. After a period of time, the flue gas stream is replaced by the input air to the process which heats up by cooling the bricks. In the meantime, the flue gas is passed to a second matrix of bricks and the process is continued by periodically switching the streams. Thus, the name *cyclic regenerator* that is given to this type of unit.

4.15.2 Rotary regenerators

In this form of apparatus heat from a hot gas stream is imparted to a metallic or ceramic medium on a slowly rotating wheel (see Figure 18). The rotation of the wheel brings it into contact with a cold gas stream, to which the medium loses the heat that it has collected. This type of equipment can be substantially more compact and less heavy than the equivalent recuperative heat exchanger for the duty. Rotary regenerators are used commonly for heat recovery from flue gases in power stations, the heat being used to preheat the inlet air. These units (Ljungström type) can be very large (typically 10 m diameter). Smaller rotary regenerators ("heat wheels") are used for exchange between room and external air in air conditioning systems.

In a gas-to-gas *recuperative* heat exchanger, the heat transfer is usually governed by the gas heat transfer coefficient and there is, therefore, a large temperature difference between the hot gas and the heat transfer surface. Since the surface must often be maintained (for flue gas systems) above the acid dewpoint temperature, this limits the bulk temperature to which the gas may be cooled. In *regenerative* heat exchangers, on the other hand, the heat transfer is often limited by conduction in the solid matrix and the difference between the hot gas and surface temperature is less (though to compensate, the surface area has to be greater). Thus, the gas can be cooled to a lower bulk temperature before the acid dewpoint is reached at the surface of the matrix. As a result, lower hot gas stream exit temperatures can be achieved with minimised danger of acidic attack, an important point in favour of this form of waste heat recovery system.

A disadvantage of rotary regenerators is cross contamination between the gas streams. However, this can be greatly mitigated by means of devices such as double-labyrinth seals at the periphery and by utilising a purge unit in conjunction with air seals.

4.16 Cooling Towers

In a wet cooling tower, warm water is sprayed onto a packing and cools as it passes down the packing, both as a result of evaporative cooling and through sensible heat transfer to a current of air which passes upwards through the packing. In a dry cooling tower, cooling is by atmospheric air only. The air can either flow by natural draught (aided by a tower structure above the packing) or by mechanical draught (blown through the packing with the aid of fans) or by a combination of the two. A useful review on cooling towers is given by Singham¹⁰. Though not often used directly in process heat transfer, the economics of cooling towers are important in choosing between heat rejection via an air-cooled heat exchanger and heat rejection using a combination of, say, a shell-and-tube heat exchanger and a cooling tower. Consideration is given to such a comparison in Example 7 of Section 7.

4.17 Electric Tubular Heat Exchangers

A whole variety of electrical heating systems are used for process fluids. However, the so-called *electrical tubular heat exchanger* is mentioned here specifically because (somewhat misleadingly, perhaps) it is referred to as a heat exchanger and has a form similar to the shell-and-tube heat exchanger (see Figure 19) with the tubes being replaced by electric heater rods. Electric heaters are extremely efficient and can be operated by control systems that allow a wide range of flow rate to be covered. For most applications, the 'exchanger' is often of baffled crossflow design which produces the smallest and most efficient unit. The typical shellside velocities in such units inhibit fouling and the presence of the baffles minimises tube vibrations. However, longitudinal flows can be used where very low pressure drops are required.

In the specific design shown in Figure 19, the electric elements are sealed into the tubesheet using "bite" couplings; this facilitates changing individual elements within the heater bundle. This coupling can be used for all liquid and gas applications and has been used up to 1200 bar. The electric elements can be manufactured in most normal materials of construction with tube thickness to suit the given process application.

Electric tubular heat exchangers can now be designed to operate in hazardous areas (in conditions where spark ignition of inflammable material could present a problem, for instance) and their general flexibility and controllability have made them an increasingly popular choice for many heating applications.

4.18 Graphite Heat Exchangers

Graphite heat exchangers consist of a block of graphite through which holes are drilled so that the hot and cold fluid stream pass at right angles to each other through the block, heat being conducted through the block from one fluid stream to the other. The construction of such heat exchangers is illustrated in Figure 20. The gaskets sealing the headers to the graphite block can be arranged to obtain multiple-pass operation as in the example shown in Figure 20.

The overriding advantage of this type of heat exchanger is its very high corrosion resistance. It is especially useful, for instance, for HCl absorbers where heat is released as HCl dissolves. However, it should be noted that, though graphite itself is resistance to most chemicals, it has a porous structure and has to be impregnated with a resin (usually a furanic or phenolic based compound) to avoid interstream leakage. The resins are attacked by free halogens and by fluids with high pH (e.g. caustic solutions above 12 per cent concentration) and this limitation should be borne in mind.

The pharmaceutical and fine chemical manufacturing industries are the major users of graphite heat exchangers because of their corrosion resistance. The exchangers can be used in conjunction with glass pipework and are a compact alternative to glass heat exchangers, having also a much higher maximum pressure than the latter.

In addition to their widespread use for single-phase flow duties, graphite heat exchangers are also used for condensation with the vapour condensing in downflow through vertical holes in the block. The surface area available for condensation can be increased by using slots rather than holes in the graphite. Both co-current and counter-current (reflux) condensing modes can be accommodated.

In some applications graphite can also be used as a construction material for exchangers of the more conventional shell-and-tube form.

4.19 Conclusion

Although a large range of heat exchangers has been covered here, it was not practicable to be exhaustive. Other types include direct contact heat exchangers, jacketed vessels, and coils in stirred vessels. However, the above will give a flavour of the wide range of types available from which selection is a vital element of design.

5. INITIAL SELECTION

The range of heat exchangers considered in this present section is listed in Table 9.1. The list is not exhaustive, but it does include the more commonly used types. For the particular application being considered, the list given in the table should be checked for the following.

- (1) Maximum pressure – many exchanger types operate only at low pressure and can immediately be ruled out for particular applications.
- (2) Temperature range – many exchanger types are applicable over a limited range of temperature and, again, this rules out a number of types.
- (3) Fluid limitations – here the main emphasis is on compatibility between the fluid and the materials of construction. For instance, in gasketed plate exchangers, it may not be possible to find a gasket material that will be compatible with the fluid. Another factor is that concerned with the consequences of failure leading to interstream mixing and/or leakage. Clearly, for highly toxic or inflammable fluids, the consequences of failure are much more significant and this can often be an important consideration in deciding on the heat exchanger type.
- (4) Size range available – it is always possible to overcome the problem of maximum size limitation by having a number of heat exchangers in parallel. However, this itself gives additional costs in piping and sometimes leads to problems in the distribution of flow between the parallel units.
- (5) Fouling and cleanability – if the fluid on one side (or both sides) of the heat exchanger is susceptible to fouling, the cleaning methods need careful consideration. Fouling occurs by a whole range of mechanisms⁴; the choice of heat exchanger is affected by this. For instance, the presence of suspended solids may render the use of narrow passages impracticable. Pre-removal of suspended solids by filtration may sometimes be economic.

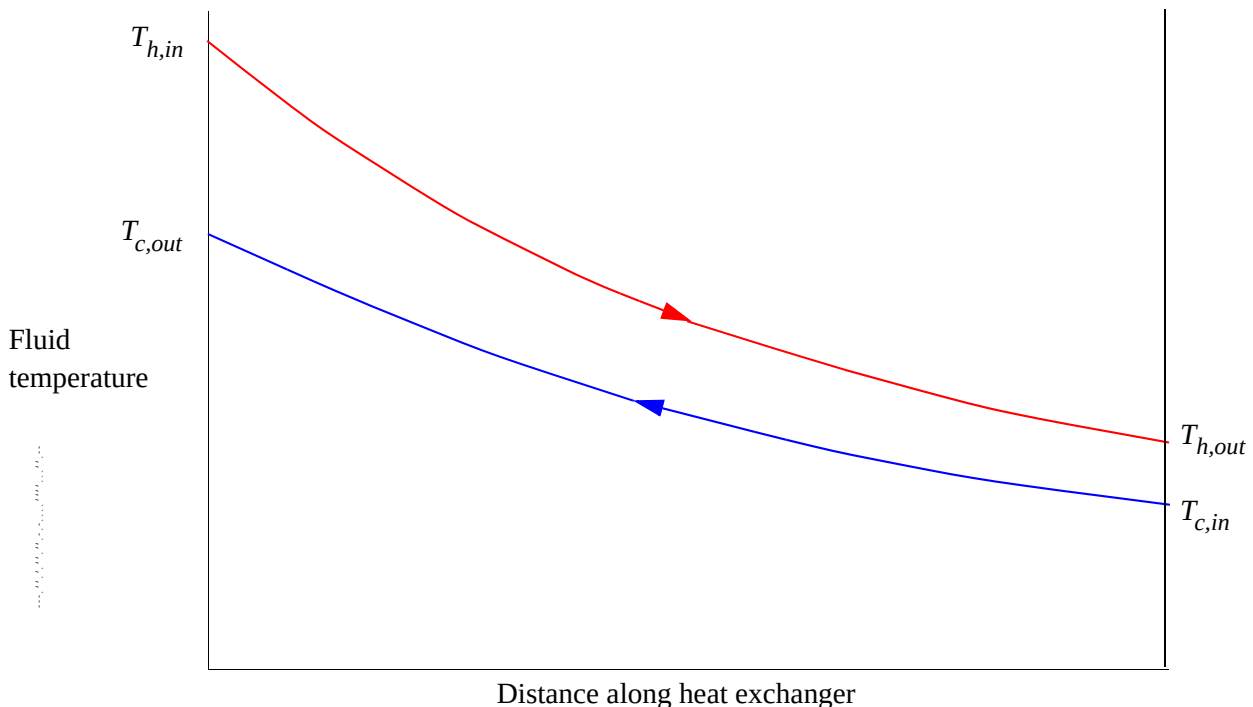
Other factors that should be considered are plot area available, design life requirement (which affects material choice) and location (which affects maintainability of the equipment). In addition, the question should be asked, "Is there a temperature cross?" (that is does the hot stream leave the heat exchanger at a temperature below the exit temperature of the cold stream). If so, units approximating counter-current flow are needed (either pure counter-flow or multi-pass units in series); if not, crossflow or mixed flow units are adequate.

Examination of Table 9.1 should lead to identification of at least one, and probably several, types of exchanger that are suitable for the particular application.

6. SELECTION BETWEEN FEASIBLE TYPES

Having obtained from Table 9.1 (and other information) a list of feasible heat exchanger types for the particular application, the next step is to obtain an indication of the cost of the heat exchanger for each of these types; clearly, this is the most important factor in the selection. Here, a method is described that was developed by Hewitt, Guy and Marsland¹. This is based on estimating the cost, C , per unit $\dot{Q}/\Delta T_m$ where $\dot{Q}$ is the heat load and ΔT_m the corrected temperature difference. In this method, it is assumed that single units are employed. However, in order to avoid inefficiencies due to temperature distribution in duties employing multi-pass exchangers, multiple exchangers in series are commonly employed. In that case, the cost estimate has to be made as the sum of the costs of the individual exchangers taking account of the calculated inter-exchanger temperatures. In order to calculate $\dot{Q}/\Delta T_m$, two approaches may be taken, namely the so-called " F_T " and "effectiveness" methods. These are described, respectively, in Sections 6.1 and 6.2. Section 6.3 gives a further introduction to the C -value method and describes the data presented here.

6.1 The F_T Method



Sketch 6.1 Temperature profiles in counter-current flow heat exchangers

For a pure counter-current heat exchanger, the temperatures vary with position as shown in Sketch 6.1. For a constant overall heat transfer coefficient and for constant specific heat capacity of the fluid streams, the mean temperature difference ΔT_m is equal to the logarithmic mean temperature difference ΔT_{lm} given by:

$$\Delta T_{lm} = \frac{(T_{h, in} - T_{c, out}) - (T_{h, out} - T_{c, in})}{\log_e \frac{(T_{h, in} - T_{c, out})}{(T_{h, out} - T_{c, in})}} \quad (6.1)$$

Heat exchangers rarely have pure counter-current flow and a common practice is to correct the mean temperature difference by a factor F_T that accounts for the deviation from pure counter-current operation:

$$\Delta T_m = F_T \Delta T_{lm}. \quad (6.2)$$

Information is available in the literature for the calculation of F_T (see for instance Reference 2). For quick calculations (often adequate at the level of accuracy of the method described here), the following approximate values of F_T , may be helpful.

Pure counter-current flow:	$F_T = 1.0$,
pure crossflow exchangers:	$F_T = 0.7$,
multi-pass exchangers:	$F_T = 0.9$,
isothermal boiling and condensation:	$F_T = 1.0$.

However, for any exchanger not having pure counter-current flow or having isothermal boiling or condensation, it is useful to check that the value of F_T is not so low as to be sensitive to small changes in stream temperature (typically, F_T should be greater than 0.8 for shell-and-tube exchangers). For these cases, therefore, it is worthwhile estimating F_T anyway. The heat load for the exchanger is calculated from

$$\dot{Q} = \dot{M}_h(h_{h, in} - h_{h, out}) = \dot{M}_c(h_{c, out} - h_{c, in}). \quad (6.3)$$

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

$$\dot{Q} = \dot{M}_h c_{p, h} (T_{h, in} - T_{h, out}) = \dot{M}_c c_{p, c} (T_{c, out} - T_{c, in}), \quad (6.4)$$

where $c_{p, h}$ and $c_{p, c}$ are the specific heat capacities of the hot and cold streams, respectively. It also follows that

$$\dot{Q} = UA \Delta T_m, \quad (6.5)$$

where A is the heat exchanger surface and where U is referred to as the *overall heat transfer coefficient*. In practice, U is not constant along the heat exchanger; this is particularly so when phase change occurs and condensation or evaporation gives rise to a variation of velocity and local parameters along the heat exchanger. Similarly, the assumptions on which the calculation of ΔT_m are based^{2, 11} are often violated. However, at the level of sophistication being used in the design selection process, it is sufficient for most purposes to assume "typical" values of U .

The procedure for determining $\dot{Q}/\Delta T_m$ using standard F_T charts is as follows.

- (1) Calculate $R_K = (\dot{M}c_p)_h/(\dot{M}c_p)_c$ and calculate the parameter P from the expression:

$$P = \frac{(T_{c, out} - T_{c, in})}{(T_{h, in} - T_{c, in})}; \quad (6.6)$$

(note that $P = E$ if the cold stream has the smaller $\dot{M}c_p$).

- (2) Read off the value of F_T from the chart for the appropriate values of R_K and P .

- (3) Calculate ΔT_{lm} from Equation (6.1).
- (4) Calculate ΔT_m from Equation (6.2) and calculate $\dot{Q}/\Delta T_m$.

6.2 The Effectiveness Method

In general, the use of normal charts of F_T needs an iterative solution if the outlet temperatures of the heat exchanger are not known. This led ESDU^{6, 11} to prefer the effectiveness – N_{TU} approach. Furthermore the cited ESDU Data Items cover a much wider range of configurations than is available elsewhere in the literature. Here, effectiveness E is defined as:

$$E = \frac{\dot{Q}}{\dot{Q}_{max}} = \frac{(T_{in} - T_{out})_{larger}}{(T_{h, in} - T_{c, in})}, \quad (6.7)$$

where $\dot{Q}_{max}$ is the maximum heat transfer that can be achieved if the outlet temperature of one of the streams reaches the inlet temperature of the other stream. The number of transfer units N_{TU} is defined as:

$$N_{TU} = \frac{UA}{(\dot{M}c_p)_{smaller}}, \quad (6.8)$$

where $(\dot{M}c_p)_{smaller}$ is the product of the flow rate and the specific heat capacity, c_p , of the stream having the lower product of $\dot{M}$ and c_p . The heat capacity ratio R is defined as:

$$R = \frac{(\dot{M}c_p)_{smaller}}{(\dot{M}c_p)_{larger}}. \quad (6.9)$$

From Equations (6.5) and (6.8)

$$\frac{\dot{Q}}{\Delta T_m} = (\dot{M}c_p)_{smaller} N_{TU}. \quad (6.10)$$

The ESDU plots of E against N_{TU} are typified by the plot for a two-pass shell-and-tube heat exchanger given in Figure 21 and by that for an unmixed crossflow exchanger given in Figure 22.

Note:-

- (i) The subscripts “larger” and “smaller” refer to the magnitude of the terms and not to one or other of the streams.
- (ii) The definitions of E , N_{TU} and R can vary between sources and great care must be taken in transferring from one publication to another.

The procedure for determining $\dot{Q}/\Delta T_m$ using the effectiveness N_{TU} charts is as follows.

- (1) Calculate R from Equation (6.9) and E from Equation (6.7).
- (2) Using the appropriate chart, read off N_{TU} for the calculated values of R and E .

- (3) Calculate the value of $\dot{Q}/\Delta T_m$ from Equation (6.10).
 Alternatively, ΔT_m may be related directly to E and N_{TU} by the expression:

$$\Delta T_m = F_T \Delta T_{lm} = \Delta T_{lm} \log_e [(1 - ER)/(1 - E)] / [N_{TU}(1 - R)]$$
 and $\dot{Q}/\Delta T_m$ calculated.

6.3 The C-value method

The overall heat transfer coefficient, U , will vary with fluid enthalpy (particularly in multiphase systems) and with fluid velocity. There is a fundamental objection, therefore, to specifying a given value for a particular duty and configuration. Order-of-magnitude estimates can often be made for U to check for gross errors in heat exchanger size calculations. Clearly, the heat transfer performance is related directly to the allowable pressure drop and the implication is that the latter must also follow conventional practice. Typical values of pressure drop are 1 per cent of inlet pressure for gases, 1 bar for liquids and 0.1 to 0.5 bar for condensers. Often, in the heat transfer text books, ranges of typical values are given. In the present section, single values are given as being typical of a particular duty and configuration. There is no implication that the assigned value is particularly reliable; it is given merely for the purpose of quick order of magnitude assessment. This procedure is usually a precursor to detailed design. Where a very large range of conditions is covered in one unit, the unit can be considered as a series of connected units, each with its own representative conditions.

The values of U (and also, by inference, C) presented here are those for in-service conditions after fouling has taken place, allowing for the mitigating effect of cleaning.

If the appropriate value of U (including proper allowance for fouling) can be estimated approximately, then the heat exchanger area A may be calculated from

$$A = \frac{1}{U} \left(\frac{\dot{Q}}{\Delta T_m} \right). \quad (6.11)$$

The quotient $(\dot{Q}/\Delta T_m)$ is characteristic of the heat exchanger duty being carried out and the cost of the exchanger to perform the duty is often estimated by multiplying A by a cost per unit area. A difficulty arises here over the definition of area, particularly when extended surfaces and complex geometries are employed. However, from the point of view of the process designer, the key question is the overall cost for the particular duty, specified in terms of $(\dot{Q}/\Delta T_m)$. Here, a cost factor C is defined that represents the cost in pounds sterling per unit $(\dot{Q}/\Delta T_m)$; C has the units £/(W/K). For a particular duty and configuration, therefore, values of C may be estimated and are given in addition to U values in Tables 9.2 to 9.8. The cost of the heat exchanger may be estimated by simply multiplying C by $(\dot{Q}/\Delta T)$.

The cost of a heat exchanger per unit surface area, and hence the C value, decreases with increasing heat exchanger size. In the information presented here, C is given at specified values of $\dot{Q}/\Delta T$ and its value at intermediate values of $\dot{Q}/\Delta T$ may be estimated by logarithmic interpolation.

It should be noted that the cost values given do not normally include site installation. While the normal piping and valving associated with a heat exchanger is often a relatively small item, this is not always true, particularly for hot gas applications, where ducting, perhaps refractory lined and, possibly, dampers capable of working at high temperatures are needed. Note that there is a relation between installation cost and size and weight and this may affect the exchanger choice.

The costs are based on current costs in 1992. Costs will vary with time (approximately according to cost estimation indices for process plant) but the relative costs are likely to vary more slowly.

To summarise, the procedure for evaluation of the alternative feasible types of heat exchanger identified in the initial selection procedure is as follows.

- (1) Estimate the heat load, $\dot{Q}$, from a heat balance (Equation (6.3)).
- (2) If the F_T method is being used, estimate the mean temperature difference, ΔT_m , using standard F_T charts together with the logarithmic mean temperature difference calculated from Equation (6.1). If the effectiveness method is being employed, calculate $\dot{Q}/\Delta T_m$ directly through Equation (6.7) to (6.10).
- (3) Calculate the quotient $\dot{Q}/\Delta T_m$ for each proposed configuration (note that, for the particular duty required, the values of $\dot{Q}/\Delta T_m$ will vary between exchanger types since the temperature difference correction factor or N_{TU} value varies).
- (4) From the tables provided for each exchanger type, read off the value of C , interpolating logarithmically (see Examples) between the levels of $\dot{Q}/\Delta T_m$ given in the tables.
- (5) Calculate the cost of each configuration for the specified duty by multiplying $\dot{Q}/\Delta T_m$, by C and compare the costs, bearing in mind possible differences in installation and pumping costs.
- (6) If one configuration is much cheaper than the others (by a factor of 1.5 to 2.0 or more, say) that design should be selected and detailed calculation and estimates carried out. If there are several designs at around the same cost, the performance of all of the designs should be estimated in greater detail. Process industry experience shows that, particularly for the more conventional designs such as shell-and-tube exchangers and air-cooled heat exchangers, wide variations in quoted costs exist reflecting the suppliers current order position in addition to the standard costs of production. For tubular exchangers, the actual costs will also vary considerably with the configuration (type of shell, lengths of unit, tube diameter, etc., for shell-and-tube exchangers; number of tube rows, tube length, etc., for air-cooled exchangers).

Tables of U and C values for various heat exchangers are given in Tables 9.2 to 9.8 as follows:

Table 9.2: Shell-and-tube heat exchangers

Table 9.3: Double-pipe heat exchangers

Table 9.4: Gasketed-plate exchangers

Table 9.5: Printed-circuit heat exchangers

Table 9.6: Plate-fin heat exchangers

Table 9.7: Air-cooled heat exchangers

Table 9.8: Welded-plate exchangers

It should be noted that the costs given in these tables are for the materials stated. Costs would differ if other materials are used.

7. EXAMPLES OF SELECTION PROCESS

7.1 Selection of Double-pipe Heat Exchanger

A heat exchanger is required for the cooling of a high viscosity oil with a specific heat of 2 kJ/kg K flowing at a rate of 0.5 kg/s and at a pressure of 30 bar (3.0 MPa). The oil enters the exchanger at 70°C and leaves at 30°C. It is cooled by water flowing at 2 kg/s and entering at 20°C. The specific heat capacity of water is 4.2 kJ/kg K. The oil stream is subject to moderate fouling. Evaluate the alternative heat exchangers for this duty and select between the feasible types on the basis of cost estimated from Tables 9.2 to 9.8.

Method. The heat exchanger types are first reviewed and suitable alternatives selected. For this case, the appropriate types are the shell-and-tube and double-pipe. The heat load, $\dot{Q}$, and water outlet temperature are calculated from a heat balance. Dealing first with the shell-and-tube heat exchanger, values of E and R are calculated from Equations (6.7) and (6.9), respectively. From Figure 21 N_{TU} is read off for these values of E and R . From Equation (6.10), $\dot{Q}/\Delta T_m$ is then estimated. For the double-pipe heat exchanger case, a pure counter-current flow may be assumed, so that $\Delta T_m = \Delta T_{lm}$. A value of ΔT_{lm} is calculated and thus $\dot{Q}/\Delta T_m$ for this case without the need for the calculation of R and E . The C -value tables, Tables 9.2 and 9.3, are then used to estimate the cost of the alternative.

Solution. For this case, the pressure is too high for a gasketed-plate exchanger, and the plate-fin, welded-plate and printed-circuit heat exchangers are eliminated on the grounds of fouling. The options considered are the shell-and-tube and double-pipe heat exchangers. The heat load, $\dot{Q}$, is given by:

$$\dot{Q} = \dot{M}_h c_{p,h} (T_{h,in} - T_{h,out}) = 0.5 \times 2.0 \times 10^3 \times (70 - 30) = 40\,000 \text{ W.}$$

The water outlet temperature is given by:

$$T_{c,out} = T_{c,in} + \frac{\dot{Q}}{\dot{M}_c c_{p,c}} = 20 + \frac{40000}{2 \times 4.2 \times 10^3} = 24.8^\circ\text{C.}$$

Assuming that the shell-and-tube exchanger is multi-pass (the most widely used) then the value of $\dot{Q}/\Delta T_m$ is calculated using the procedure given in Section 6. The values of E and R are calculated as follows (Equation (6.7)):

$$E = \frac{(T_{in} - T_{out})_{larger}}{(T_{h,in} - T_{c,in})}.$$

The oil has the larger value of $(T_{in} - T_{out})$ and thus:

$$E = \frac{(70 - 30)}{(70 - 20)} = 0.8.$$

The values of $\dot{M}_c c_p$ are calculated for the oil and for the water as follows:

$$\begin{aligned} (\dot{M}_c c_p)_{oil} &= 0.5 \times 2.0 = 1.0 \text{ kW/K} \\ (\dot{M}_c c_p)_{water} &= 2.0 \times 4.2 = 8.4 \text{ kW/K.} \end{aligned}$$