5.1 PRINCIPLES OF OPERATION OF A HEAT EXCHANGER

In a heat exchanger, the fluid flows can be performed in multiple arrangements.

One can easily show that thermodynamically, the most efficient heat exchanger is the counter-flow heat exchanger (Figure 5.1.1), but other concerns than the thermodynamic effectiveness are taken into account when designing a heat exchanger: the maximum permissible temperatures in one fluid, or more often considerations of size, weight or cost.

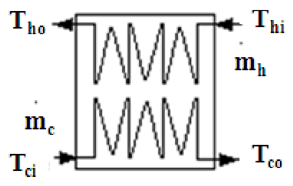

Figure 5.1.1: Heat exchanger sketch

It follows that the configurations of exchangers that are encountered in practice are relatively numerous.

However, we can gather these configurations in three main geometries:

- counter-flow, in which fluids flow in parallel and in opposite directions;
- parallel-flow, in which fluids flow in parallel and in the same direction;
- cross-flow, in which fluids flow in perpendicular directions.

We will denote the hot fluid by index h, and the cold fluid by index c. Besides the geometric configuration, exchanger sizing or performance depends on many parameters and variables:

- mass flows $\dot{m}_h$ and $\dot{m}_c$ passing through them;
- inlet temperatures T_{hi} and T_{ci} and outlet temperatures T_{ho} and T_{co} of both fluids;
- heat exchange coefficients U_h and U_c of each fluid;
- thermal resistance of the wall $\frac{e}{\lambda}$;
- the surface A of the exchanger;
- the pressures of both fluids, almost constant;
- the thermophysical properties of fluids, which are used to determine coefficients U_h and U_c. It is essentially the heat capacity c_p, the density ρ, thermal conductivity λ, and viscosity μ.

In what follows, we will assume that the heat exchange coefficients U_h and U_c and thermophysical properties of fluids maintain a constant value at any time in the entire heat exchanger.

If this assumption is not verified, then in order to study the performance of the exchanger, it is necessary divide it into small volume elements, in which these properties can be considered constant. The calculations are then much more cumbersome.

Finally, we always assume that the heat exchanger is globally adiabatic, that is to say there is no heat exchange with the surroundings.

5.1.1 HEAT FLUX EXCHANGED

It can be shown, where fluid flow is pure (parallel or counter-flow), that the heat flux exchanged between the two fluids is given by the formula:

$$\phi = UA\ \Delta T_{ml} \tag{5.1.1}$$

$$\text{with } \Delta T_{ml} = \frac{\Delta T_0 - \Delta T_L}{\ln \dfrac{\Delta T_0}{\Delta T_L}} \tag{5.1.2}$$

A being the exchange surface, U the overall heat exchange coefficient (see next section), ΔT_{ml} the logarithmic mean temperature difference, and ΔT_0 and ΔT_L the differences in fluid temperatures respectively at one end and at the other end of the exchanger, with the convention: $\Delta T_0 > \Delta T_L$.

Care must be taken that we speak of temperature differences between the two fluids at the entrance and exit of the exchanger (dimensions 0 and L), and not differences of inlet temperatures ($T_{hi} - T_{ci}$) and output temperatures ($T_{ho} - T_{co}$) of the fluids.

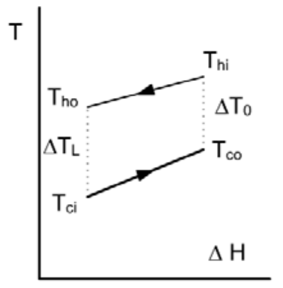

Figure 5.1.2 : LMTD notations

In the example in Figure 5.1.2, we have:

$\Delta T_0 = T_{hi} - T_{co}$

$\Delta T_L = T_{ci} - T_{ho}$

When the flow configuration is more complex, we introduce a correction factor F (less than 1), which is given by charts calculated or determined experimentally, most often provided by manufacturers.

Equation (5.1.1) becomes:

$$\phi = UA\ F\ \Delta T_{ml} \tag{5.1.3}$$

This formula shows that to transfer a given flux ϕ, if we wish to reduce the irreversibilities and therefore ΔT_{ml}, we need the product UA to be the largest possible, which can be done either by increasing the surface, but this directly affects the cost, or by increasing the U value, which is an aim sought by all heat exchanger designers.

In this chapter, the design of the heat exchanger will be limited to determining the product UA of the exchanger surface A by the overall heat transfer coefficient U, the accurate estimation of the latter depending on the internal constructive details of the exchanger.

In general, the design of heat exchangers is a compromise between conflicting objectives, whose two main ones are:

- a large exchange area is desirable to increase the effectiveness of heat exchangers, but it results in high costs;
- small fluid flow sections increase the values of the heat exchange coefficients U_h and U_c defined in the next section, and thus reduce the area, but they also increase pressure drops.

5.1.2 HEAT EXCHANGE COEFFICIENT U

We only deal in Chapter 2 of Part 5 of the precise determination of U and the study of pressure drops in heat exchangers, not that these difficult issues lack interest, quite the contrary, but because, as for other components, we are not currently looking to complete their detailed internal design: we limit ourselves to dealing with their integration into systems that are energy technologies. We will only present here the principles governing the calculation of U.

Thermal science teaches us that for a flat plate heat exchanger, the overall heat transfer coefficient U is such that its inverse, called thermal resistance, is the sum of thermal resistances between the two fluids:

$$\frac{1}{U} = \frac{1}{h_h} + \frac{e}{\lambda} + \frac{1}{h_c} \qquad (5.1.5)$$

h_h being the heat transfer coefficient between the hot fluid and the wall, h_c the heat transfer coefficient between the cold fluid and the wall, e the wall thickness and λ the thermal conductivity of this wall.

If the wall is composed of several layers of different materials, or if other deposits have covered the wall, this formula becomes:

$$\frac{1}{U} = \frac{1}{h_h} + \sum_{k=1}^{n} \frac{e_k}{\lambda_k} + \frac{1}{h_c} \qquad (5.1.6)$$

The overall thermal resistance is the sum of (n + 2) thermal resistances, the largest of them being the one that slows down the more heat that is exchanged.

For example, in the case of a gas/liquid heat exchanger, the convection coefficient can be 10 to 100 times lower on the gaseous side than on the liquid side. The thermal resistance of the wall and the liquid side are generally negligible, and U is approximately equal to h gas side.

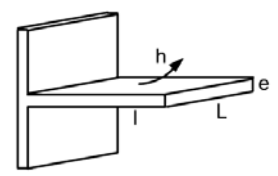

Figure 5.1.3: Rectangular fin

In this case, we seek to increase the gas side exchange surface by using fins. The previous formulas are complicated, and as the surfaces of cold and hot sides are not the same, we introduce a warm side global exchange coefficient U_h, such that its inverse is:

$$\frac{1}{U_h} = \frac{1}{\eta_{0,h}\, h_h} + \frac{e}{\frac{A_w}{A_h}\lambda} + \frac{1}{\frac{A_c}{A_h}\eta_{0,c}\, h_c} \qquad (5.1.7)$$

with A_h and A_c total exchange areas warm and cold sides, A_w exchanger wall surface, and $\eta_{0,h}$ and $\eta_{0,c}$ overall fin effectiveness on warm and cold sides (see section 5.1.3).

There is also a cold side global exchange coefficient U_c, with of course:

$U_c A_c = U_h A_h$

5.1.3 FIN EFFECTIVENESS

For a thin rectangular fin (e << l, Figure 5.1.3), we can show that the fin effectiveness, defined as the ratio of heat actually exchanged to the heat that would be exchanged if the fin was at the temperature of the base, is:

$$\eta_a = \frac{\tanh(ml)}{ml}$$

$$\text{with } m = \sqrt{\frac{2\,h}{\lambda\,e}} \qquad (5.1.8)$$

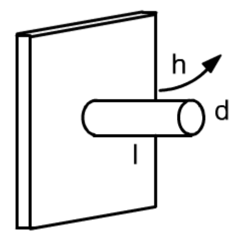

Figure 5.1.4 : Needle fin

For fins of different shapes, the value of m changes. For example, for circular needle fins (Figure 5.1.4):

$$m = \sqrt{\frac{4\,h}{\lambda\,d}} \qquad (5.1.9)$$

For other geometries, we refer to the literature (Incropera, Dewitt, 1996).

The total flux exchanged is equal to the sum of the fluxes exchanged on the one hand by the fins (area A_a) and on the other hand by the wall between the fins (area A - A_a). This defines an overall effectiveness η_0 equal to the ratio of the flux actually exchanged to the flux that would be exchanged if the entire surface was at the temperature of the base:

$$\eta_0 = \frac{h\,A_a\,\eta_a + h\,(A - A_a)}{h\,A} = 1 - \frac{A_a}{A}(1 - \eta_a) \qquad (5.1.10)$$

5.1.4 VALUES OF CONVECTION COEFFICIENTS h

The values of convection coefficients h_h and h_c depend on fluid thermophysical properties and exchange configurations. Further explanations will be given in chapter 2 of Part 5 on technological design.

These coefficients can be estimated from correlations giving the value of Nusselt number $Nu = \frac{h\,d_h}{\lambda}$, as a function of Reynolds $Re = \frac{\rho\,V\,d_h}{\mu}$ and Prandtl $Pr = \frac{\mu\,c_p}{\lambda}$ numbers.

The hydraulic diameter d_h is equal to the ratio of four times the flow area S to the wetted perimeter p: $d_h = 4\,S/p$ (if there are isolated walls, the perimeter of heat exchange must be considered).

Inside tubes, the formula most used is that of Mac Adams:

$$Nu = 0{,}023\ Re^{0{,}8}\ Pr^{0{,}4} \qquad (5.1.11)$$

For flow perpendicular to tubes, several options exist, such as Colburn:

$$Nu = 0{,}33\ Re^{0{,}6}\ Pr^{0{,}33} \qquad (5.1.12)$$

In general, the exponent of the Reynolds number is between 0.5 and 0.8, and that of the Prandtl number between 0.33 and 0.4. Bontemps et al. (1998) give values of correlations for very special finned geometries.