

Chapter 2

Design Computation

2.1 Introduction

The design computation consists of determining the surface S of the heat exchanger or the tube bank to obtain a certain result.

To that extent, note that for thermal balance we can write that

$$q = M'' c''_{pm} (t''_2 - t''_1) = \eta_e M' c'_{pm} (t'_1 - t'_2) \quad (2.1)$$

In (2.1) q is the heat transferred to the heated fluid in the time unit in W, M' and M'' are the mass flow rates of the heating fluid and the heated fluid, respectively, in kg/s, t'_1 and t'_2 are the inlet and outlet temperatures of the heating fluid, t''_1 and t''_2 are the inlet and outlet temperatures of the heated fluid in °C, c'_{pm} and c''_{pm} are the mean isobaric specific heat of both the heating and the heated fluid in J/kgK, and η_e is the actual or assumed efficiency of the heat exchange.

In addition, we know (from Chap. 1) that

$$q = US\Delta t_m. \quad (2.2)$$

For the design computation, once M' , M'' , t'_1 , t'_2 , η_e are known, we may wish to obtain the exchange of a certain heat q ; from (2.1) we obtain the temperatures t'_2 and t''_2 , given that the two mean specific heat depend on the four temperatures in question. It is possible instead to impose temperature t'_2 or temperature t''_2 (2.1); still leads to the other unknown temperature and to heat q .

In any case, in the end we have the value of q and the four temperatures.

At this point, if the fluids are in parallel flow or in counter flow we compute the value of Δt_m , corresponding to the mean logarithmic temperature difference, as we shall see later on. If this not the case, we compute the actual mean temperature difference by multiplying the logarithmic one by a corrective factor; in any case we obtain the value of Δt_m .

Once the overall heat transfer coefficient U is computed, we obtain the necessary surface S through (2.2).

As far as the computation of U we indicate which criterion should be followed in our view to compute α' and α'' (see Chap. 1)

For the computation of the heat transfer coefficient of the heated fluid it is best to refer to the arithmetic average of both inlet and outlet temperatures, whereas for the computation of the heat transfer coefficient of the heating fluid, it is generally best to refer to the logarithmic average of the two temperatures above, the necessity to refer to film temperature when it is required for the computation of α , notwithstanding.

2.2 Fluids in Parallel Flow or in Counter Flow

If we examine two fluids in parallel flow or in counter flow, the pattern of the temperatures t' and t'' is shown in both Fig. 2.1 and Fig. 2.2.

M' and M'' are the mass flow rates of both fluids, and c'_{pm} and c''_{pm} refer to the mean specific isobaric heat. The overall heat transfer coefficient U is assumed to be constant.

The heat transferred through the elementary surface dS is given by:

$$dq = U dS (t' - t''). \quad (2.3)$$

On the other hand, given that t' decreases with the increase surface and by introducing the exchange efficiency η_e , the same value dq is equal to

$$dq = -\eta_e M' c'_{pm} dt'. \quad (2.4)$$

If the exchange occurs with parallel flow, given that t'' increases with S , from Fig. 2.1 we see that

$$dq = M'' c''_{pm} dt''. \quad (2.5)$$

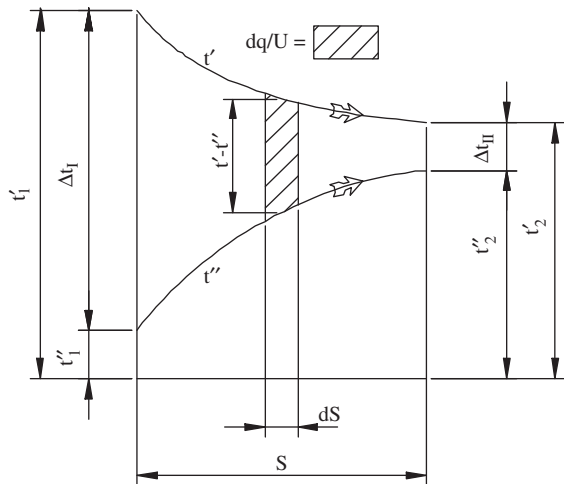
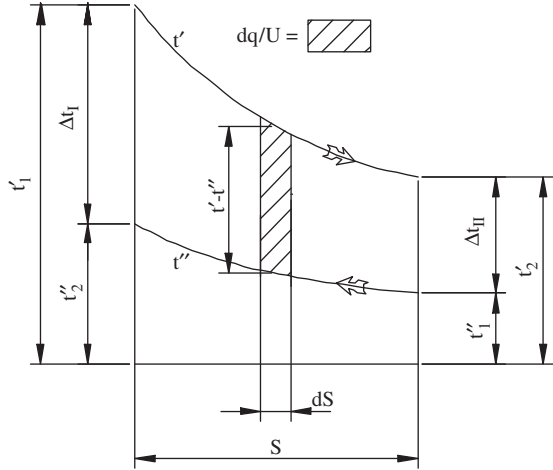


Fig. 2.1 Parallel flow

Fig. 2.2 Counter flow

Viceversa, Fig. 2.2 relative to heat transfer during counter flow shows that

$$dq = -M''c''_{pm}dt''. \quad (2.6)$$

Therefore,

$$d(t' - t'') = -dq \left(\frac{1}{\eta_e M' c'_{pm}} \pm \frac{1}{M'' c''_{pm}} \right); \quad (2.7)$$

and recalling (2.3)

$$d(t' - t'') = -UdS(t' - t'') \left(\frac{1}{\eta_e M' c'_{pm}} \pm \frac{1}{M'' c''_{pm}} \right). \quad (2.8)$$

Here the plus sign indicates parallel flow and the minus sign indicates counter flow.

On the other hand

$$q = M''c''_{pm}(t''_2 - t''_1) = \eta_e M' c'_{pm}(t'_1 - t'_2). \quad (2.9)$$

Thus, with parallel flow

$$\frac{1}{\eta_e M' c'_{pm}} + \frac{1}{M'' c''_{pm}} = \frac{1}{q}(t'_1 - t''_1 - t'_2 + t''_2), \quad (2.10)$$

and with counter flow

$$\frac{1}{\eta_e M' c'_{pm}} - \frac{1}{M'' c''_{pm}} = \frac{1}{q} (t'_1 - t''_2 - t'_2 + t''_1) \quad (2.11)$$

The term on the right of the equal sign of both (2.10) and (2.11) (Figs. 2.1 and 2.2) is equal to:

$$\frac{\Delta t_I - \Delta t_{II}}{q}. \quad (2.12)$$

(2.8) can therefore be written as follows:

$$\frac{d(t' - t'')}{t' - t''} = -\frac{U dS}{q} (\Delta t_I - \Delta t_{II}); \quad (2.13)$$

and through integration we obtain:

$$|-\log_e (t' - t'')|_I^II = \frac{US}{q} (\Delta t_I - \Delta t_{II}); \quad (2.14)$$

then

$$\log_e \frac{\Delta t_I}{\Delta t_{II}} = \frac{US}{q} (\Delta t_I - \Delta t_{II}). \quad (2.15)$$

Finally,

$$q = US \frac{\Delta t_I - \Delta t_{II}}{\log_e \frac{\Delta t_I}{\Delta t_{II}}}. \quad (2.16)$$

The following quantity is the mean logarithmic temperature difference Δt_{ml} :

$$\boxed{\Delta t_{ml} = \frac{\Delta t_I - \Delta t_{II}}{\log_e \frac{\Delta t_I}{\Delta t_{II}}}} \quad (2.17)$$

then

$$q = US \Delta t_{ml}. \quad (2.18)$$

The resulting equation is quite similar to (1.1) where instead of the constant difference in temperature between the heating fluid and the heated one, we have the mean logarithmic temperature difference given by (2.17) (of course, U represents U_o and U_i , respectively, depending on whether S is the outside or inside surface of the tubes [see (1.2) and (1.3)]).

Another way to proceed is suggested by the fact that, if the ratio $\Delta t_I / \Delta t_{II}$ is not too high, Δt_{ml} does not considerably differ from the mean arithmetic temperature difference equal to:

$$\Delta t = \frac{\Delta t_I + \Delta t_{II}}{2}. \quad (2.19)$$

Therefore, we can write that

$$\Delta t_{ml} = \chi \frac{\Delta t_I + \Delta t_{II}}{2}. \quad (2.20)$$

Based on (2.17) and (2.20), the corrective factor χ is given by

$$\chi = \frac{2 (\Delta t_I - \Delta t_{II})}{(\Delta t_I + \Delta t_{II}) \log_e \frac{\Delta t_I}{\Delta t_{II}}}. \quad (2.21)$$

The value for χ obtained from Fig. 2.3 clearly shows the influence of $\Delta t_I / \Delta t_{II}$ on the reduction of Δt_{ml} with respect to the mean arithmetic temperature difference.

Note that the use of this diagram combined with (2.20) leads to the exact computation of Δt_{ml} .

In the case of fluids in parallel flow, the value of the ratio $\Delta t_I / \Delta t_{II}$ is higher than with fluids in counter flow, thus the value of both χ and Δt_{ml} is smaller. Based on (2.18), it follows that a greater surface with equal transferred heat is needed.

The assumption so far was that the value of U is constant.

In fact, the heat transfer coefficients of both fluids vary with temperature, and so does the value of U . Therefore, it is a question of defining which value of U must be introduced in (2.18).

It is customary to consider the values of the heat transfer coefficients of both fluids corresponding to the average between the inlet and the outlet temperature, and to compute the overall heat transfer coefficient U based on these values of α .

This is the only recommendable (conservative) criterion for heated fluid, even though the behavior of the temperature is not linear. As far as the heating fluid, given the behavior of temperature, it is generally advisable to adopt the

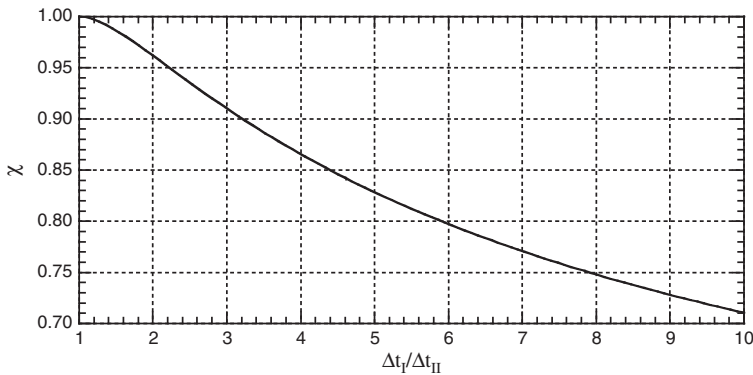


Fig. 2.3

logarithmic average between the inlet and outlet temperatures as reference temperature. Naturally, if the film temperature must be adopted for the computation of the heat transfer coefficient, the temperature of reference must be the average between the temperature mentioned earlier and the wall temperature.

The mean logarithmic temperature of the heating fluid is given by

$$t'_{ml} = \frac{t'_1 - t'_2}{\log_e \frac{t'_1}{t'_2}} \quad (2.22)$$

We will come back to this topic when discussing the verification computation.

2.3 The Mean Difference in Temperature in Reality

In real instances the behavior of the fluids, with the exception of fluids with cross flow which are a case in itself, is usually close to the behavior of fluids in parallel flow or counter flow. In general, the most logical methodology to obtain the actual value of Δt_m is to refer to the mean logarithmic difference in temperature in parallel flow or counter flow, and to introduce a corrective factor by which to multiply this difference to obtain Δt_m .

To that extent we introduce three dimensionless factors, the same we will use for the verification computation.

They are:

$$\psi = \frac{t'_2 - t''_1}{t'_1 - t''_1}; \quad (2.23)$$

$$\beta = \frac{\eta_e M' c'_{pm}}{M'' c''_{pm}}; \quad (2.24)$$

$$\gamma = \frac{US}{\eta_e M' c'_{pm}}. \quad (2.25)$$

Since this is a design computation, the inlet and outlet temperatures of both fluids are known, and as a result so is the value of ψ .

Moreover, the value of β is also known.

If we consider the fluids in parallel flow, there is precise connection between the three indicated factors. In fact, based on (3.14) factor γ which is indicated by γ_p , is given by

$$\gamma_p = \frac{1}{1 + \beta} \log_e \frac{1}{(1 + \beta)\psi - \beta}. \quad (2.26)$$

If we consider the fluids in counter flow instead, and if $\beta \neq 1$, based on (3.23) factor γ indicated with γ_c is given by

$$\gamma_c = \frac{1}{1 - \beta} \log_e \left(\frac{1 - \beta}{\psi} + \beta \right). \quad (2.27)$$

If $\beta = 1$ instead, from (3.28) we obtain

$$\gamma_c = \frac{1}{\psi} - 1. \quad (2.28)$$

In real instances the value of γ meant to satisfy the imposed value of ψ , is close plus or minus from the value of γ_p or γ_c .

Based on Sects. 2.1 and 2.2, the transferred heat is equal to

$$q = \eta_e M' c'_{pm} (t'_1 - t'_2) = US \Delta t_m = \eta_e M' c'_{pm} \gamma \Delta t_m; \quad (2.29)$$

then

$$\Delta t_m = \frac{t'_1 - t'_2}{\gamma}. \quad (2.30)$$

Given that t'_1 and t'_2 are fixed values, we establish that Δt_m is inversely proportional to γ .

If we consider the fluids in parallel flow, instead of (2.30) we must write that

$$\Delta t_{ml(p)} = \frac{t'_1 - t'_2}{\gamma_p} \quad (2.31)$$

where $\Delta t_{ml(p)}$ is the mean logarithmic temperature difference referred to fluids in parallel flow, and γ_p is obtained through (2.26).

Therefore, by introducing the corrective factor χ_p , we may write

$$\boxed{\chi_p = \frac{\Delta t_m}{\Delta t_{ml(p)}} = \frac{\gamma_p}{\gamma}} \quad (2.32)$$

In other words, if the reference is to fluids in parallel flow, after computation of γ_p with (2.26) based on imposed values of ψ and β , the case in question is examined and the real value of γ required to obtain the requested value of ψ is calculated; this way the value of corrective factor χ_p is computed through (2.32).

Thus is possible to compute the value of the actual mean temperature difference Δt_m starting from the value of $\Delta t_{ml(p)}$ relative to the fluids in parallel flow.

The procedure is similar in reference to fluids in counter flow. In that case

$$\boxed{\chi_c = \frac{\Delta t_m}{\Delta t_{ml(c)}} = \frac{\gamma_c}{\gamma}} \quad (2.33)$$

where $\Delta t_{ml(c)}$ is the mean logarithmic temperature difference referred to fluids in counter flow, and γ_c is obtained through (2.27) or (2.28).

Note that with reference to fluids in parallel flow, for the situation to actually be possible we must have

$$\psi > \frac{\beta}{1 + \beta}. \quad (2.34)$$

If the reference is to fluids in counter flow instead, and $\beta > 1$, for the situation to actually be possible we must have

$$\psi > \frac{\beta - 1}{\beta}. \quad (2.35)$$

The described process allowed us to build a series of Tables which are included in Appendix A. We refer the reader to this section to make the comparisons discussed in the text.

We did not consider the instances where $\gamma > 6$ since they are unlikely and not advisable. In addition, we neglected those cases where the difference plus or minus between the actual mean temperature difference and the logarithmic one is under 1%, thus to be considered rather insignificant.

In the Tables of Appendix A the missing values to the left of those included correspond to impossible cases or to those where $\gamma > 6$. The missing values to the right of those included correspond to cases where the difference between Δt_m and $\Delta t_{ml(p)}$ or $\Delta t_{ml(c)}$ is less than $\pm 1\%$; for those we can assume the mean logarithmic temperature difference for Δt_m .

2.3.1 Fluids in Cross Flow

The behavior of fluids in cross flow (Fig. 2.4) is closer to that of fluids in counter flow compared to fluids in parallel flow.

So we computed the values of χ_c to include them in the Table A.1.

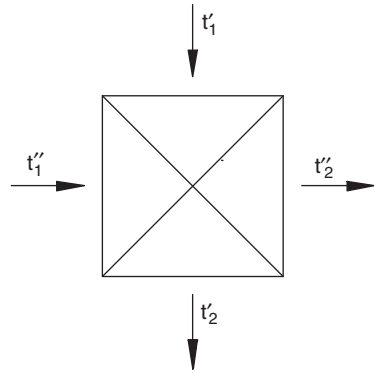


Fig. 2.4 Cross flow

2.3.2 Heat Exchangers

2.3.2.1 Heat Exchangers with Two Passages of the Internal Fluid

We consider heat exchangers with two passages of the fluid inside the tubes shown in Fig. 2.5.

As you see, there are four possible combinations indicated by the letters A, B, C and D.

If the number of passages of the fluid external to the tubes is odd, as shown in Fig. 2.5, types A and B which are apparently different from one another, have in fact the same behavior and have the same value of χ .

This depends on the fact that each has one of the two peculiar characteristics of fluids in parallel flow. In fact, in type A the internal fluid enters the tubes in the same location in which the external fluid enters the exchanger; in type B the fluid exits the tubes in the same location in which the external fluid exits the exchanger; this makes their behavior absolutely identical and similar to that of fluids in parallel flow.

If the number of passages of the fluid external to the tubes is even instead, the just described situation occurs for types A and D.

Similar considerations are true for types C and D if we consider an odd number of passages of the external fluid, as described in Fig. 2.5.

Each one has one of the peculiar characteristics of fluid in counter flow. In fact, in type C the internal fluid exits the tubes in the same location in which the external fluid enters the exchanger. In type D the internal fluid enters the tubes in the same location in which the external fluid exits the exchanger. This makes their behavior absolutely identical and similar to that of fluids in counter flow.

If the number of passages of the external fluid is even instead, the just described situation occurs for types B and C.

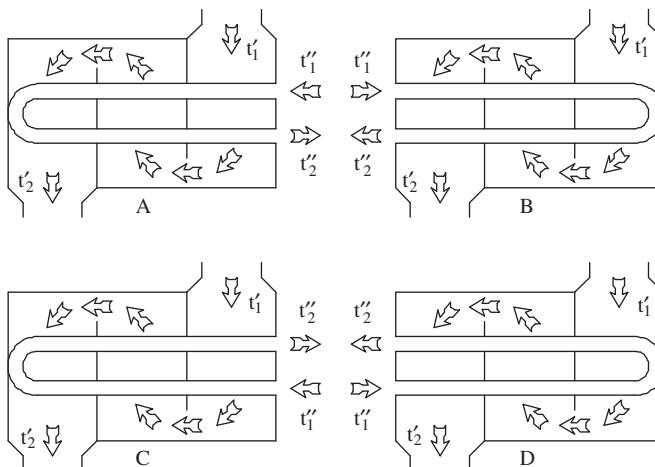


Fig. 2.5 Heat exchangers with two passages of internal fluid

Therefore, for types A and B (or A and D) it would be logical to calculate the value of the corrective factor χ_p , thus referring the requested mean difference in temperature Δt_m to the mean logarithmic difference relative to fluids in parallel flow. Nonetheless, to be able to compare them with types C and D (or B and C) we preferred to compute χ_c ; for types C and D (or B and C) the logical solution is undoubtedly that to compute the corrective factor χ_c , thus referring Δt_m to the mean logarithmic difference in temperature relative to fluids in counter flow.

The computation of the values of χ_c is based on a few schemata and assumptions. First of all, the position of the baffles must be such that the exchange surface is divided in equal sections for the various passages of the fluid outside the tubes. Moreover, we assume that the differences in temperature of the different threads of the external fluid annul each other, due to the mixture of the threads occurring with the reversal of the direction of the flow. Thus, the temperature of the external fluid is uniform at the entrance of the new passage.

The analysis was conducted (and this is true for all Tables in Appendix A) by considering β variable between 0.1 and 3.0 and considering ψ variable between 0.04 and 0.96.

The values of χ_c for types A and D or for types B and C with two passages of the external fluid are shown in Tables A.2 and A.3.

The values of χ_c for types A and B or for types C and D with three passages of the external fluid are shown in Tables A.4 and A.5.

The values of χ_c for types A and D or for types B and C with four passages of the external fluid are shown in Tables A.6 and A.7.

Finally, the values of χ_c for types A and B or for types C and D with five passages of the external fluid are shown in Tables A.8 and A.9.

A single passage of the fluid outside the tubes is not considered because in that case the exchanger is reduced to a coil with two sections; its behavior is implied by the section on coils to follow later on.

Analysis of the Tables leads us to interesting considerations.

First of all, it is not surprising that, β and ψ being equal, the value of χ_c and thus of Δt_m for types A and B (or A and D) with reference to Tables A.2, A.4, A.6 and A.8 is always lower than that for types C and D (or B and C) with reference to Tables A.3, A.5, A.7 and A.9.

In addition, the increase in the number of passages of the fluid outside the tubes in types A and B (or A and D) is matched by an increase of Δt_m , whereas in types C and D (or B and C) it decreases. The difference in behavior between types A and D and types B and C which is rather noticeable with 2 passages of the external fluid diminishes with the increase in passages of the external fluid.

Through five passages of the external fluid the values of χ_c relative to the various types of exchangers get considerably closer.

It is rather unlikely that the number of passages of the external fluid is greater than 5; if this should be the case, through, and considering the outcome registered above due to caution we recommend to adopt the values of χ_c included in Table A.8 for all types.

2.3.2.2 Heat Exchangers with Three Passages of the Internal Fluid

If there is just one passage of the external fluid, the exchanger is reduced to a coil with 3 sections. We refer you to the section on coils.

The various types of exchangers with three passages of fluid inside the tubes are shown in Fig. 2.6 and indicated by E, F, G and H.

If the number of passages of the fluid outside the tubes is even, as shown in Fig. 2.6, types E and F have a characteristic in common with the fluids in parallel flow.

In fact, in type E both the internal and the external fluid enter the exchanger in the same position; in type F both fluids exit the exchanger in the same position instead.

Always considering an even number of passages of the external fluid, types G and H have a characteristic in common with the fluids in counter flow.

In fact, if we consider type G we notice that the internal fluid exits from the tubes in the position in which the external fluid enters into the exchanger.

In type H the internal fluid enters the tubes in the position in which the external fluid exits from the exchanger instead.

With an even number of passages of the external fluid type E behaves like type F and type G behaves like type H.

If the number of passages of the external fluid is odd, it is necessary to consider types E and G individually; type E has both characteristics relative to the inlet and outlet of the fluids in common with the fluids in parallel flow; type G has both characteristics relative to the inlet and outlet of the fluids in common with fluids in counter flow.

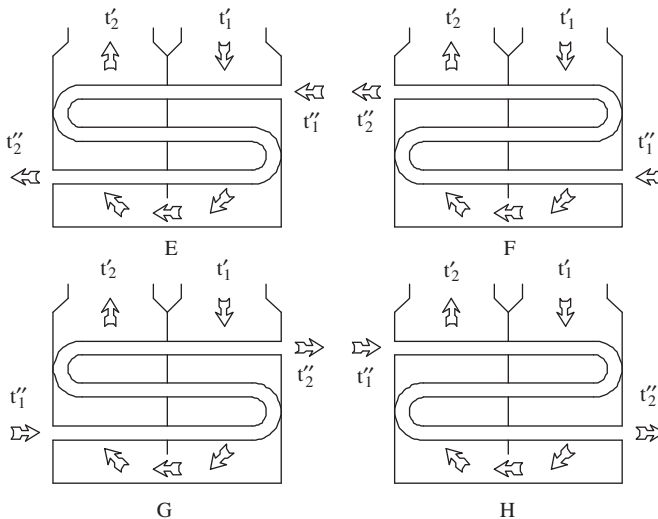


Fig. 2.6 Heat exchangers with three passages of internal fluid

The values of χ_c for types E and F or for types G and H with two passages of the external fluid are shown in Tables A.10 and A.11.

The values of χ_c for type E or for type G with three passages of the external fluid are shown in Tables A.12 and A.13.

The values of χ_c for types E and F or for types G and H with four passages of the external fluid are shown in Tables A.14 and A.15.

Finally, the values of χ_c for type E or for type G with five passages of the external fluid are shown in Tables A.16 and A.17.

The analysis of the values of χ_c in the various Tables shows the following.

For types E and F, moving from 2 to 4 passages of the external fluid (Tables A.10 and A.14), the value of the corrective factor generally slightly decreases.

For types G and H, moving from 2 to 4 passages of the external fluid (Tables A.11 and A.15), the value of the corrective factor generally slightly increases.

The opposite occurs for types E and G with an odd number of passages of the external fluid.

In fact, for type E, moving from 3 to 5 passages of the external fluid (Tables A.12 and A.16) the value of the corrective factor increases; viceversa, for type G, always moving from 3 to 5 passages (Tables A.13 and A.17), the value of the corrective factor decreases.

Finally, note that for type E with an odd number of passages of the external fluid the corrective factor is considerably smaller in comparison with an even number of passages. This is not surprising, given that with an odd number of passages the behavior of the exchanger closely resembles that of fluids in parallel flow.

Similarly, for type G with an odd number of passages of the external fluid the corrective factor is considerably higher compared to an even number of passages; in fact, with an odd number of passages the behavior of the exchanger closely resembles that of fluids in counter flow.

In the unlikely case that the number of passages of the external fluid is greater than 5, given the modest variations taking place after variations in the number of passages, the recommendation is to refer to Tables A.14, A.15, A.16 and A.17, depending on the situation.

2.3.2.3 Heat Exchangers with Four Passages of the Internal Fluid

Now we consider exchangers with 4 passages of the fluid inside the tubes (Fig. 2.7).

If there is just one passage of the external fluid, the exchanger is reduced to a coil with 4 sections. We refer you to the section on coils.

If the number of passages of the fluid outside the tubes is ≥ 3 , for some types of exchangers the behavior is quite similar to that of exchangers with 2 passages of the fluid inside the tubes.

Specifically, for types I and L with 3 passages of the external fluid, it is possible to use Table A.4; for types I and N with 4 passages of the external fluid it is possible to use Table A.6; finally, for types I and L with 5 passages of the external fluid it is possible to use Table A.8. The potential errors occurring through this simplification do not exceed 1%. The situation is entirely different, in the case of types I and N

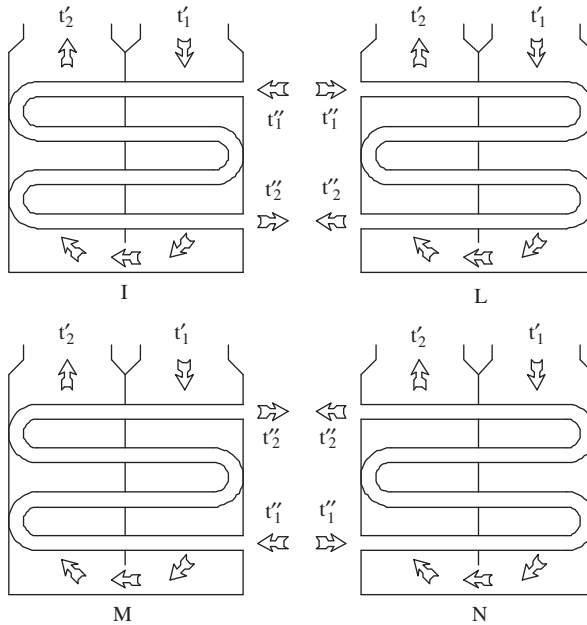


Fig. 2.7 Heat exchangers with four passages of internal fluid

with respect to types A and D, if the exchanger has 2 passages of the external fluid, as shown in Fig. 2.7.

The behavior of the exchanger with 4 passages of the fluid inside the tubes is considerably different from that of an exchanger with 2 passages. Table A.2 cannot be used; for the value of one must refer to Table A.18.

The values of χ_c for types L and M with 2 passages of external fluid are shown in Table A.19.

The values of χ_c for types M and N with 3 passages of the external fluid are shown in Table A.20.

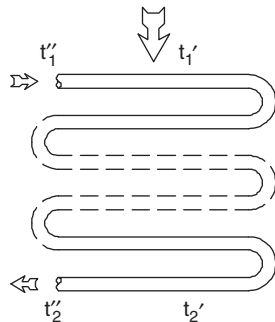
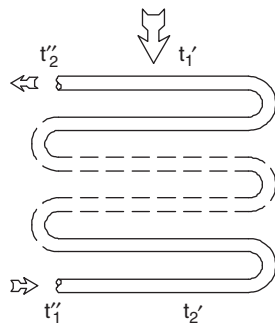
The values of χ_c for types L and M with 4 passages of the external fluid are shown in Table A.21.

Finally, the values of χ_c for types M and N with 5 passages of the external fluid are shown in Table A.22.

As far as the influence of the number of passages of the external fluid on the values of the corrective factor, the same considerations made in reference to the exchangers with 2 passages of the fluid inside the tubes apply. Thus, if the number of passages of the external fluid should be greater than 5, it is recommended to apply caution and refer to Table A.8.

2.3.3 Coils

In the case of coils in Figs. 2.8 and 2.9 it would not be possible to speak of fluids in parallel flow or counter flow. In fact, each section of the coil is hit by the fluid

Fig. 2.8 Coils – parallel flow**Fig. 2.9** Coils – counter flow

outside the tubes in such a way to be considered cross flow. Therefore, the coil is the sum of elements in which the fluxes are in cross flow.

Usually, though, if the internal fluid enters the coils in correspondence of the inlet in the coil of the external fluid (Fig. 2.8), it is customary to speak of fluids in parallel flow. If the inside fluid enters the coils in correspondence of the outlet of the external fluid (Fig. 2.9), it is customary to speak of fluids in counter flow.

At this point we would like to analyze the topic in-depth both for coils with fluids in parallel flow and those with fluids in counter flow.

2.3.3.1 Coils with Fluids in Parallel Flow

With respect to fluids in real parallel flow they show differences in heat transfer that we would like to highlight. Based on the premises, the corrective factor χ_p is logically calculated.

The considered range is, as for heat exchangers, as follows: $\beta = 0.1 - 3.0$ and $\psi = 0.04 - 0.96$.

Tables A.23, A.24 and A.25 show the values of χ_p relative to coils with 2, 3 and 4 sections, respectively.

We establish that the value of χ_p is always greater than one. This means that the heat transfer occurs with more favourable characteristics compared to those relative to fluids in parallel flow, given that the value of Δt_m is greater than $\Delta t_{m(p)}$.

As the number of section increases, the value of χ_p gets close to unity. If there are 4 sections there are few instances where $\chi_p > 1.02$; if the number of sections is ≥ 4 , giving up the little advantage represented by $\chi_p > 1$, we recommend to adopt the mean logarithmic difference in temperature referred to fluids in parallel flow as value of Δt_m .

2.3.3.2 Coils with Fluids in Counter Flow

Now we consider the coils in Fig.2.9.

Naturally, in this case we calculated the values of χ_c .

The analyzed range is, as usual, as follows: $\beta = 0.1 - 3.0$ and $\psi = 0.04 - 0.96$.

The values of χ_c for a number of sections equal to 2, 3, 4, 6, 8 and 10 are shown in Tables A.26, A.27, A.28, A.29, A.30 and A.31.

As expected, we establish that the values of χ_c are all below unity. This means that the heat transfer is less favorable in comparison with fluids in counter flow, given that Δt_m is smaller than $\Delta t_{ml(c)}$.

The phenomenon is particularly noticeable when the number of sections is small, while it decreases when their number is high.

If the number of sections is ≥ 10 , the situations where $\chi_c < 0.98$ are rare and unlikely. Therefore, it is possible to conclude that in reality if the number of sections is ≥ 10 , the coil may be treated as if the fluids were in fact in counter flow by adopting for Δt_m the value of $\Delta t_{ml(c)}$.

In any case, for those situations outlined in Table A.31 where the value of the corrective factor is considerably far from one, it is possible to refer to this Table, even for a number of sections greater than 10.

2.3.4 Tube Banks with Various Passages of the External Fluid

We consider a tube bank consisting of a series of straight tubes; a fluid flows inside the tubes, while another fluid hits the bank outside with a series of passages created through dividing baffles. If there is only one passage of the fluid outside the tubes, these are fluids in cross flow, and we refer the reader to the appropriate section.

The classic device of this type is the recuperative air heater at the end of a steam generator. From now on we will refer to this device but keeping in mind that this type of exchanger can be used even with other fluids, generally gaseous ones. In air heaters the flue gas is generally located inside the tubes while the air hits the bank outside, but nothing stands in the way of the opposite solution.

The external fluid can enter the heater in correspondence of the inlet to the tubes of the internal fluid, or viceversa with the external fluid entering the heater in correspondence of the exit from the tubes of the internal fluid. Figure 2.10 represents an air heater of the first kind with three passages of the external fluid. Figure 2.11 represents an air heater of the second kind instead.

Clearly, with the first kind the behavior of the fluid through the heater recalls the typical behavior of fluids in parallel flow, whereas the second type is similar to that of fluids in counter flow.

Fig. 2.10 Tube bank with several passages of the external fluid – parallel flow

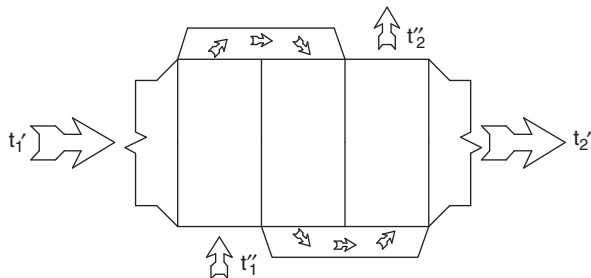
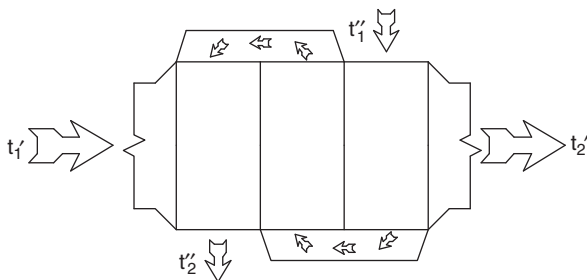


Fig. 2.11 Tube bank with several passages of the external fluid – counterflow



In fact, with these devices it is customary to speak of fluids in parallel flow or in counter flow even this is not exactly true. This topic requires in-depth analysis.

Therefore, we will refer to χ_p for the first type and to χ_c for the second one.

We could consider using the values of χ_p and χ_c already obtained for the coils.

In fact, if the fluid flowing through the tubes of the heater were compared to the fluid hitting the coil, and the fluid hitting the tubes of the heater with the fluid flowing through the coil, the analogy is evident. Still, we must consider that while the temperature of the internal fluid is unique in any position along the coil, the temperature of the fluid hitting the tube bank varies not only depending on the direction of the flux, but also transversally to it. Then the values of the factors cited in relation with the coils are only approximated values.

Another very simple procedure could be as follows. If we assume that the heater is represented by a series of sections where the motion of the fluids is in cross flow, the values of χ_c relative to cross flow can be used for every section, and in the end a global value of χ_p or χ_c is reached to solve the problem. Even this method, though, contains an error. The values of χ_c in Table A.1 are based on uniform temperatures at the inlet of both fluids, while those at the outlet are the average temperatures of the various threads at the exit. In our case we can hypothesize that the temperature of the external fluid is uniform at the inlet of every passage, given the mixture of the threads occurring with the reversal of the flux, but this is certainly not true for the fluid flowing in the tubes. For the latter the division in sections is purely formal because every tube is in one piece where the fluid takes on its own temperature condition which changes from tube to tube.

In view of this, even this method leads to values of χ_p or χ_c yielding only approximated computation.

To obtain more realistic values of χ it is therefore necessary to do a more in-depth analysis to factor in these facts. This is what was done leading to the values in Tables A.32, A.33, A.34, A.35, and A.36.

2.3.4.1 Tube Banks with Fluids in Parallel Flow

We considered the usual range: $\beta = 0.1 - 3.0$ and $\psi = 0.04 - 0.96$.

For the tube bank in parallel flow in Fig. 2.10 the Tables A.32 and A.33 list the values of χ_p for 2 and 3 passages of the external fluid. Of course, they are greater than unity, given that the heat transfer is more favourable than in the case of fluids in parallel flow. We establish that the values of χ_p with 2 passages are greater compared to those with 3 passages. Therefore, the solution with 3 passages is less favourable. Finally, if the passages are ≥ 4 our advice is to give up the modest advantage represented by the fact that in some cases $\chi_p > 1$ by adopting the mean logarithmic temperature difference relative to fluids in parallel flow for Δt_m .

2.3.4.2 Tube Banks with Fluids in Counter Flow

Tables A.34, A.35 and A.36 show the values of χ_c relative to the tube bank in counter flow in Fig. 2.11, respectively, with 2, 3 and 4 passages of the external fluid. The examined range include: $\beta = 0.1 - 3.0$ and $\psi = 0.04 - 0.96$.

Of course, the factors χ_c are below unity since the heat transfer is less favourable compared to the one with fluids in counter flow.

We see that even with 4 passages the difference between Δt_m and the mean logarithmic temperature difference may even be considerable (about up 10%) and it is advisable to take this fact into account.

We did not pursue the investigation any further by examining even solutions with a number of passages greater than 4 given that they are unlikely. In case solutions of this type were adopted, we recommend to conservatively refer to the values of χ_c listed in Table A.36.

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