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About the overall heat transfer coefficient of plate heat exchangers

Despre coeficientul de transfer termic global al schimbătorului de căldură cu plăci

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Abstract. *The present work seeks to establish some practical working relationships for determining the values of overall heat transfer coefficient, k, related to plate heat exchangers. This problem is experimentally solved by the manufacturers of such heat exchangers in the world, so our work has more of an educational-research character, which can serve to some users of plate heat exchangers in district heating systems or indoor central heating installations.*

Key words: *plate heat exchanger, overall heat transfer coefficient*

1. Introduction

For some years, plate heat exchangers are the most used heat exchangers in district heating systems, as well as in indoor heating systems, as they are equipment with high energy performance and a high degree of compactness. As it is known, there are various constructive plate heat exchanger types, regarding the plates from which the exchanger is made up, the coupling of the plates being done either in a removable system (with gaskets) or in a fixed, non-removable system. The energy performance of these heat exchangers is dependent on their overall heat transfer coefficient, which is the object of our concern in this work.

An important influence on the value of the overall heat transfer coefficient is of course the type of circulation of thermal fluids within the heat exchanger, mainly counter-current or co-current. It is known that the energy performances of the counter-current heat exchanger are superior to the co-current one. In the present work, the case of a counter-current type plate heat exchanger will be analysed.

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Another important influence on the value of the overall heat transfer coefficient is due to the dependence on temperature for the physical parameters of thermal fluids (density ρ, specific heat c, thermal conductivity λ and kinematic viscosity ν).

2. Description of the problem. Calculation relations

The basic relationship used to evaluate the global thermal transfer coefficient of the heat exchanger is:

$$\frac{1}{k} = \frac{1}{\alpha_1} + \frac{\delta_p}{\lambda_p} + \frac{1}{\alpha_2} \tag{1}$$

In order to determine the convective heat transfer coefficients related to the two thermal fluids circulating in counter-current through the heat exchanger, relationships based on the values of the Nusselt criteria (Nu_1 and Nu_2) were used. Therefore:

$$Nu_1 = \frac{\alpha_1 \cdot l_c}{\lambda_f};\ \ Nu_2 = \frac{\alpha_2 \cdot l_c}{\lambda_f} \tag{2}$$

From relations (2) it follows:

$$\alpha_1 = \frac{Nu_1 \cdot \lambda_f}{l_c};\ \ \alpha_2 = \frac{Nu_2 \cdot \lambda_f}{l_c} \tag{3}$$

For plate heat exchangers, a relationship used for the Nusselt criterion is:

$$\mathrm{Nu} = C \cdot \mathrm{Re}^n \cdot \mathrm{Pr}^m \cdot \left(\frac{\mu}{\mu_p}\right)^p \tag{4}$$

with coefficients $m = 1/3$ and $p = 0.17$. Some values of the coefficients C and n are presented in table 1 (as presented in [1], [2]), for different plate chevron angles β.

β	Re	*C*	*n*
≤ 30°	> 10	0.348	0.663
45°	> 100	0.300	0.663
50°	> 300	0.130	0.732
60°	> 400	0.108	0.703
≥ 65°	> 500	0.087	0.718

Table 1. Values of coefficients *C* and *n* for plate heat exchangers

In the present paper, we considered a plate heat exchanger with a chevron angle β = 60°, and a Reynolds number Re > 400. Neglecting the effect of viscosity, one obtains:

$$Nu = 0{,}108 \cdot \mathrm{Re}^{0{,}703} \cdot \mathrm{Pr}^{0{,}333} \tag{5}$$

where we have the following expressions for the Reynolds and Prandtl numbers:

$$\mathrm{Re}=\frac{w\cdot l_c}{\nu};\quad \mathrm{Pr}=\frac{\nu}{a};\quad a=\frac{\lambda_f}{\rho\cdot c} \tag{6}$$

The physical parameters of thermal fluids (water) were considered to be dependent on temperature [3], their expressions in the specific interval for heating systems being:

$$\begin{aligned}
&\rho=-0,0031\cdot t_m^2-0,1098\cdot t_m+1001\\
&c=0,0134\cdot t_m^2-1,2129\cdot t_m+4203,7\\
&\lambda_f=-10^{(-5)}\cdot t_m^2+0,0024\cdot t_m+0,5533\\
&\nu=10^{(-6)}\cdot\begin{pmatrix}-2,0406\cdot 10^{(-10)}\cdot t_m^5+8,3863\cdot 10^{(-8)}\cdot t_m^4-\\ -1,3650\cdot 10^{(-5)}\cdot t_m^3+1,1548\cdot 10^{(-3)}\cdot t_m^2-\\ -5,7371\cdot 10^{(-2)}\cdot t_m+1,7857\end{pmatrix}
\end{aligned} \tag{7}$$

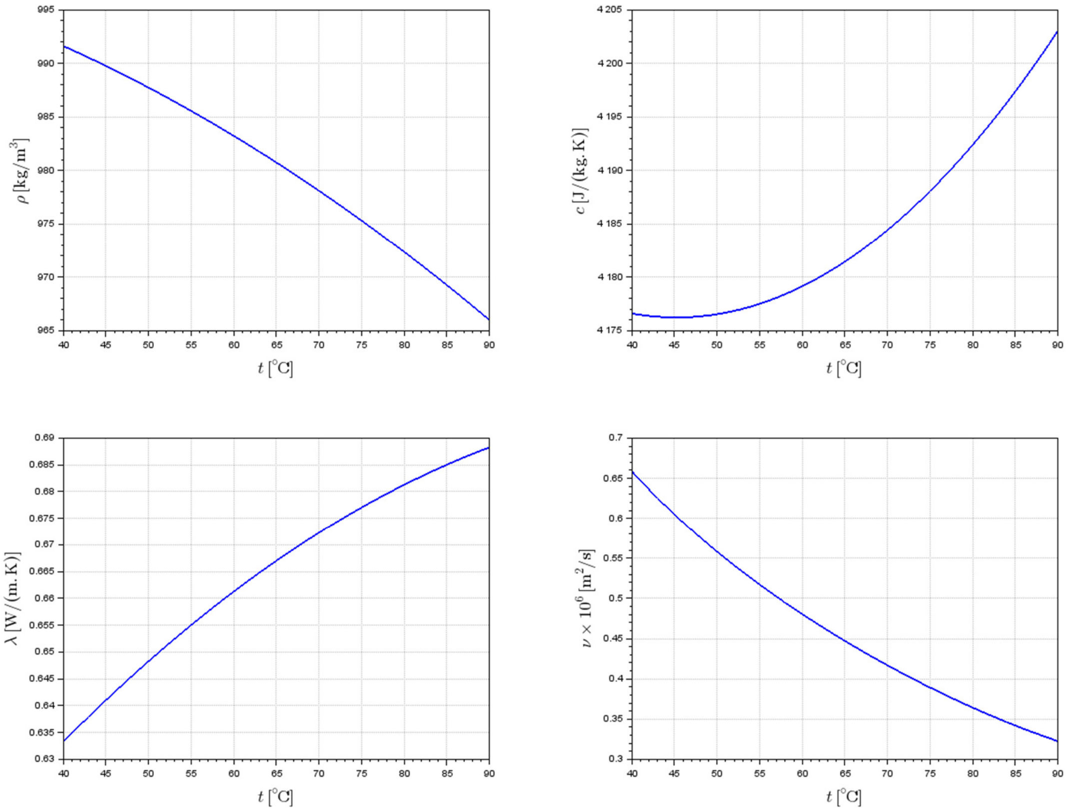

Fig. 1. The correlations for water density ρ, specific heat c, thermal conductivity λ and kinematic viscosity ν

It follows that for the evaluation of the convective thermal transfer coefficients α_1, α_2 it is necessary to evaluate the criteria Nu_1, Nu_2 and therefore of the criteria Re_1, Re_2 and Pr_1, Pr_2.

This involves the evaluation the physical parameters of the thermal fluids corresponding to the average temperatures of the thermal fluids. According to the theory presented at large in [4] in the chapter of heat exchangers, the determination of the average values of the temperatures of the thermal fluids for counter-current heat exchangers can be done using the relations:

$$\begin{aligned} t_{m1} &= C_1 + C_2 \cdot F \\ t_{m2} &= C_1 + y \cdot C_2 \cdot F \end{aligned} \qquad (8)$$

where the constants C_1 and C_2 have the expressions:

$$\begin{aligned} C_1 &= \frac{-y \cdot E}{1 - y \cdot E} \cdot t_{11} + \frac{1}{1 - y \cdot E} \cdot t_{21} \\ C_2 &= \frac{1}{1 - y \cdot E} \cdot t_{11} - \frac{1}{1 - y \cdot E} \cdot t_{21} \end{aligned} \qquad (9)$$

and F is the heat transfer co-module, which, together with the heat transfer module, E, have the expressions:

$$\begin{aligned} y &= \frac{G_1}{G_2} = \frac{a_1}{a_2} \\ NTU_1 &= \frac{k \cdot S}{G_1 \cdot \rho \cdot c} = \frac{k}{a_1 \cdot \rho \cdot c} \\ E &= \exp\left(-NTU_1 \cdot (1 - y)\right) \\ F &= \frac{1 - E}{-\ln(E)} \end{aligned} \qquad (10)$$

In order to carry out the assessments presented, an iterative procedure is proposed, which starts with an initially proposed value for the global heat transfer coefficient for the heat exchanger, k_0. The nominal values of sizing parameters t_{110}, t_{210}, t_{120} and t_{220} are also considered known. Based on them, the values of the specific volumic flow rates of the thermal fluids are established:

$$\begin{aligned} a_1 &= \frac{k_0}{\rho \cdot c} \cdot \frac{\Delta t_{ml0}}{\left(t_{110} - t_{120}\right)} \\ a_2 &= \frac{k_0}{\rho \cdot c} \cdot \frac{\Delta t_{ml0}}{\left(t_{220} - t_{210}\right)} \end{aligned} \qquad (11)$$

The iterative procedure involves the determination of the values *NTU*, *y*, *E*, *F*, and the average temperatures of the thermal fluids on the two circuits, t_{m1} and t_{m2}.

The values of convective heat transfer coefficients, α_1, α_2 are obtained, according to those presented, as well as the value of the overall heat transfer coefficient, *k*. With the new value obtained for *k*, the points in the procedure are followed again and so on, until a certain maximum imposed error is attained. Basically, the values of the inlet temperatures of the thermal fluids of the heat exchanger are proposed and the corresponding value of the overall heat transfer coefficient, *k*, is determined iteratively.

The circulation velocities of the thermal fluids between the heat exchanger changer plates were considered to have the same ratio given by the value of *y*, respectively 0.4 m/s on the primary circuit and 0.6 m/s for the secondary circuit.

Three values for the distance between the heat exchanger plates were considered for this analysis, namely $\delta = 0.002$, 0.004 and 0.006 m. The obtained results proved that this is an important constructive parameter.

3. Obtained results

In order to carry out this analysis, an automatic calculation tool was built in SCILAB, that allow for a couple of values (t_{11}, t_{21}) to quickly provide the values of the global heat transfer coefficient of the heat exchanger, *k*.

It was observed that the values of the overall heat transfer coefficient, *k*, depend quite a lot on the average value of the couple of input temperature values, which is why an attempt was made to correlate the output values, *k*, with the input ones, (t_{11}, t_{21}), more precisely, the correlation between *k* and $t_{med} = (t_{11} + t_{21})/2$ and $dt = t_{11} - t_{21}$.

Some of the obtained results are presented below.

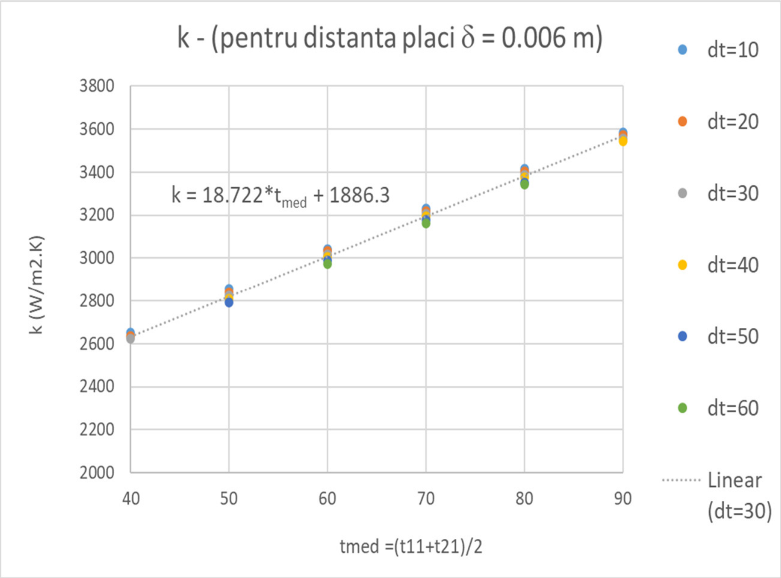

Fig. 2. The correlation for *k* ($\delta = 0.006$ m)

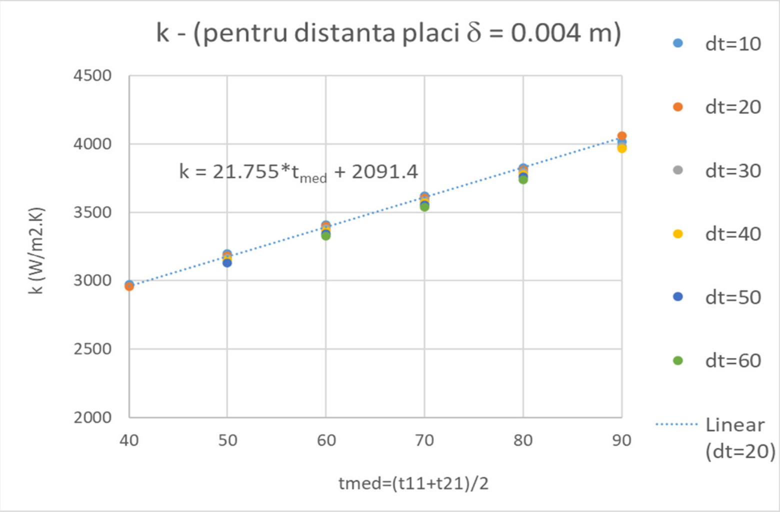

Fig. 3. The correlation for *k* (δ = 0.004 m)

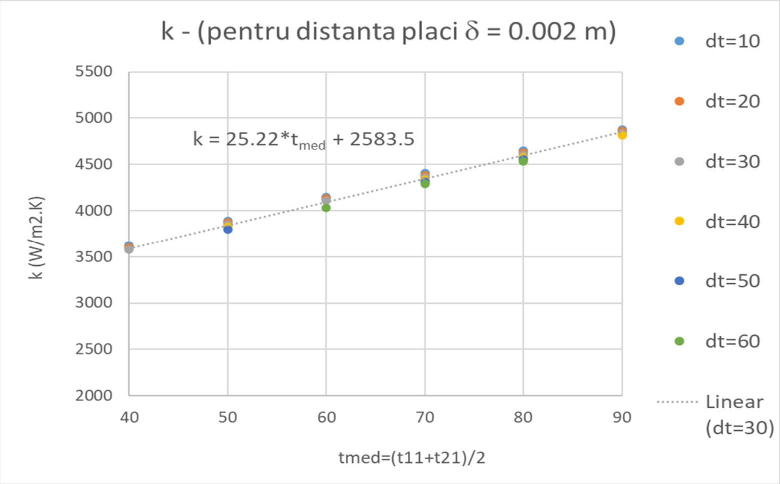

Fig. 4. The correlation for *k* (δ = 0.002 m)

The 3 straight lines from the figures 2, 3 and 4, placed on the same diagram, give us the opportunity to see the importance of the distance between consecutive plates.

As a conclusion, for a heat exchanger with a known distance between the plates, a straight-line equation connecting *k* and t_{med} can be determined, as:

$$k = m \cdot t_{med} + n \tag{12}$$

To determine the coefficients *m* and *n*, the diagram in Fig. 6 can be used. Using the expressions of coefficients *m* and *n* from fig. 6, relation (12) can be written in a more developed way:

$$k = \left(-1624.5 \cdot \delta + 28.397\right) \cdot t_{med} + \left(2969.5 \cdot e^{-78.63 \cdot \delta}\right) \tag{13}$$

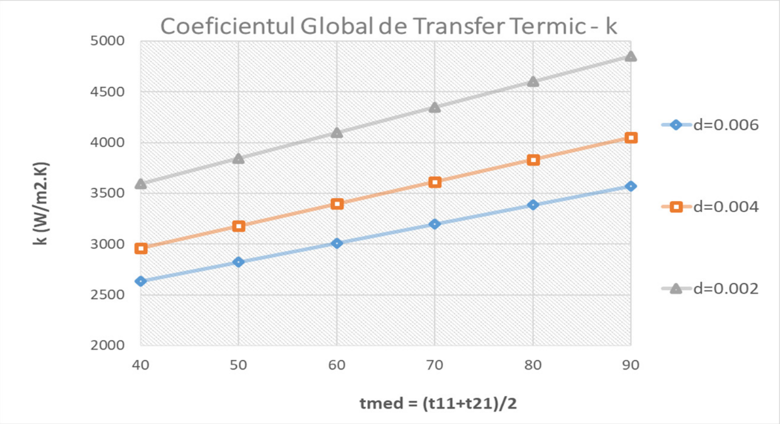

Fig. 5. The comparison of the correlations for *k*

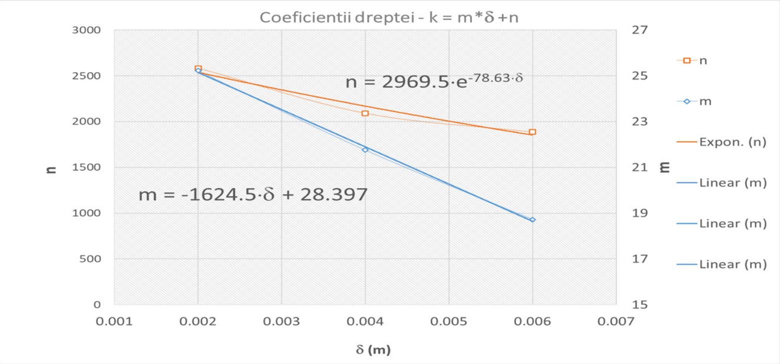

Fig. 6. The coefficients for the *k* correlation

Conclusions

The simplified procedure for determining the overall heat transfer coefficient for the analysed counter-current heat exchanger assumes establishing the distance between the consecutive plates of the heat exchanger, δ, then the coefficients *m* and *n* according to the diagram in fig. 6 and of course the average temperature t_{med}.

The presented diagrams and consequently the simplified procedure refer exclusively to the counter-current plate heat exchanger. The logical flowchart for the whole calculation procedure is presented in Figure 7.

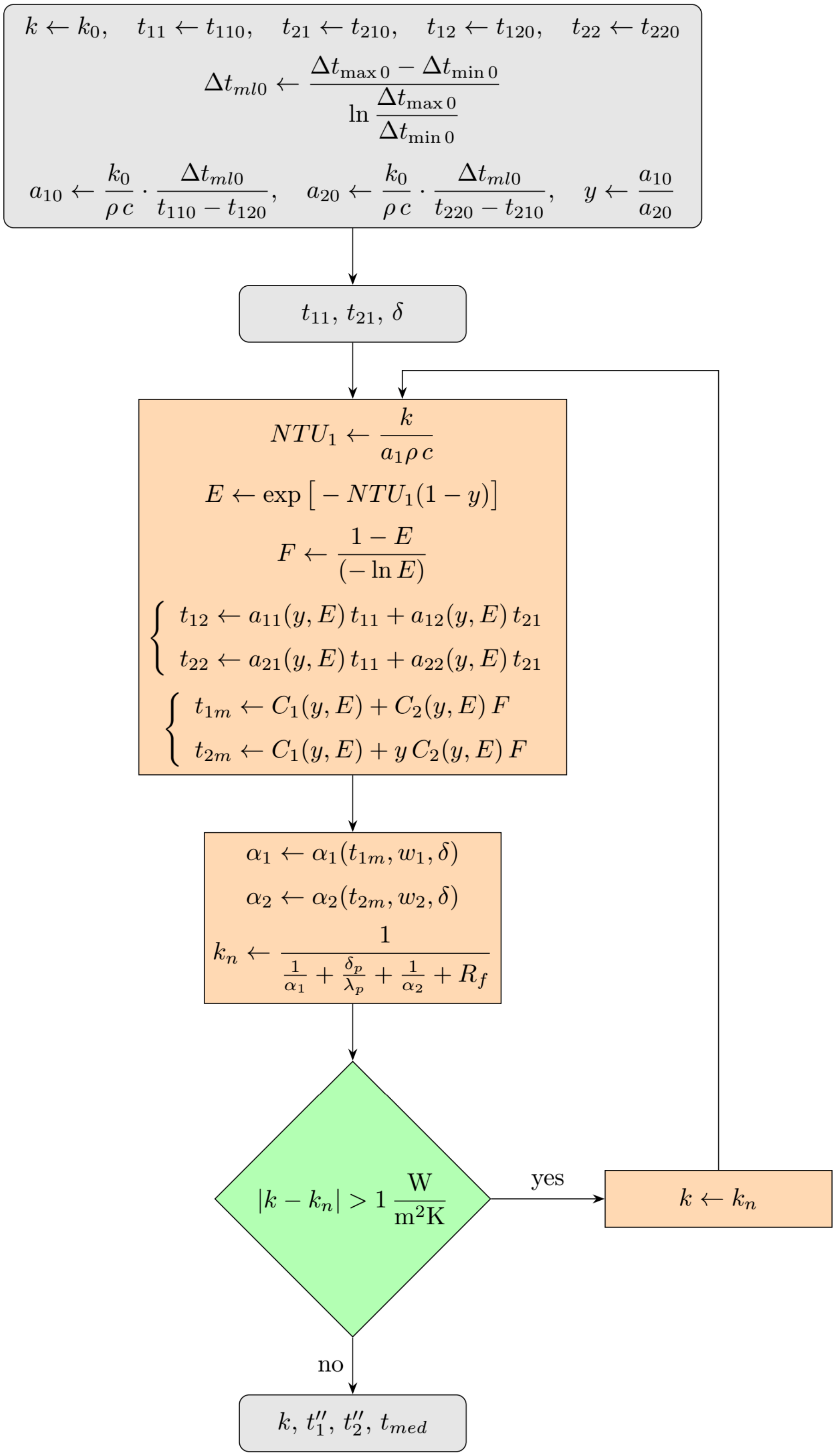

Fig. 7. The logical flowchart for the calculation procedure

Notations

t_{11} – the temperature of the primary thermal fluid at the entrance to the heat exchanger, ^{o}C;
t_{12} – the temperature of the primary thermal fluid at the exit from the heat exchanger, ^{o}C;
t_{21} – the temperature of the secondary thermal fluid at the entrance to the heat exchanger, ^{o}C;
t_{22} – the temperature of the secondary thermal fluid at the exit from the heat exchanger, ^{o}C;
t_{m1}, t_{m2} – the average temperature of the thermal fluid on the primary/secondary circuit, ^{o}C;
Δt_{ml} – the logarithmic mean temperature difference, K;
t_{med} – the arithmetic mean of the input temperatures of the thermal fluids, ^{o}C;
dt – the difference in the input temperatures of the thermal fluids, ^{o}C;
k – overall heat transfer coefficient of the heat exchanger, W/m^2.K;
α_1 – convective heat transfer coefficient for the primary fluid, W/m^2.K;
α_2 – convective heat transfer coefficient for the secondary fluid, W/m^2.K;
δ - the distance between the plates of the heat exchanger, m;
δ_p – the thickness of the stainless steel plate of the heat exchanger, m;
λ_p – the thermal conductivity of the stainless steel plate of the heat exchanger, W/m.K;
λ_f – the thermal conductivity of thermal fluids, W/m.K;
l_c – the characteristic length in the flow of thermal fluids, m;
a – the thermal diffusivity of thermal fluids, m^2/s;
ν - the kinematic viscosity of thermal fluids, m^2/s;
ρ - the density of thermal fluids, kg/m^3;
c – the specific heat of thermal fluids, J/kg.K;
w – the velocity of a thermal fluid, m/s;
G_1, G_2 – the volume flow rates of primary and secondary fluids, L/h;
a_1, a_2 - the specific volume flow rates of primary and secondary fluids, L/(h.m^2);
$\rho \cdot c$ – the volumetric specific heat, 1.163 W.h/(L.K);
NTU_1 – the number of (heat) transfer units, -;
E, F – the heat transfer module and co-module, $0 < E < F < 1$, -;
Nu – Nusselt number, -;
Re – Reynolds number, -;
Pr – Prandtl number, -;
0 – index for nominal values.

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