ASSESSMENT OF REAL HEAT TRANSFER COEFFICIENTS THROUGH SHELL AND TUBE AND PLATE HEAT EXCHANGERS

Dan CONSTANTINESCU[*)], *Horia PETRAN*[1)]

ABSTRACT

The purpose of this paper is to present a procedure used in the assessment of the real heat transfer characteristic of shell and tube and plate heat exchangers. The theoretical fundamentals of the procedure are introduced as well as the measured data collection and processing. The theoretical analysis is focused on the adoption of criterial equations which, subjected to certain verification criteria presented in the paper, provide the most credible value of the convection heat transfer coefficients inside the circular and flat tubes. In the end two case studies are presented, one concerning a shell and tube heat exchanger operational at INCERC Thermal Substation and the other concerning a plate heat exchanger tested on the Laboratory Stand of the Department of Building Services and Efficient Use of Energy in Buildings of INCERC Bucharest.

Key-words: forced convection, turbulent and transitory flow, dimensionless number, heat flow-rate, momentum eddy diffusivity, thermal eddy diffusivity

REZUMAT

Scopul articolului de faţă este de a prezenta o procedură aplicabilă pentru evaluarea caracteristicii reale de transfer de căldură proprie schimbătoarelor de căldură de tip tub în tub şi a celor compacte de tip plăci plane. Se prezintă fundamentarea teoretică împreună cu validarea empirică bazată pe măsurarea şi prelucrarea datelor măsurate. Analiza teoretică se concretizează prin adoptarea ecuaţiilor criteriale a căror rezolvare conduce la determinarea coeficientului de transfer de căldură prin convecţie forţată în interiorul tuburilor cilindrice, respectiv al traseelor de curgere plane. Lucrarea prezintă două studii de caz proprii unui schimbător de căldură de tip tub în tub din dotarea Punctului Termic INCERC, respectiv a unui schimbător de căldură compact cu plăci plane, testat în Laboratorul Departamentului de Instalaţii din INCERC Bucureşti.

Cuvinte cheie: convecţie forţată, curgere tranzitorie şi turbulentă, flux termic, difuzie turbulentă a impulsului, difuzie turbulentă a căldurii

1. INTRODUCTION

The assessment of the heat exchangers real thermal characteristic is necessary both in the activity related to their sizing and in the usual operational activity.

Currently, the thermal substations of the Romanian district heating systems are equipped with shell and tube as well as plate heat exchangers. To know the global heat transfer coefficients means to have the possibility to make two important decisions, as follows:

- in the case of shell and tube heat exchangers, based on the thermal efficiency indicator, it is decided to preserve or decommission them, either for repairs or for replacement by compact plate heat exchangers;
- in the case of plate heat exchangers, the necessity of cleaning them off the substances deposited on the plate surfaces.

In both cases the decision is made subsequent to a comparison between the real and the theoretical thermal performance. The main indicators are the global heat transfer coefficient and the thermal resistance of the organic and inorganic matters deposits on the surface of the heat transfer units.

[*)] Corresponding author: D. Constantinescu, e-mail: dan_const_home@yahoo.com

[1)] National Institute for Research and Development in Construction, Urban Planning and Sustainable Spatial Development "URBAN-INCERC", INCERC Bucharest Branch, Romania

An important phase is the selection of the correct criterial equations based on which will be assessed the values of the global heat transfer coefficients among the heat carriers flown inside the heat exchangers. In the case of the thermal substations, both the primary heat carrier (hot) and the secondary one (cold) is water.

The analysis presented in the paper concerns the transitory and turbulent flow conditions, as they are specific to the heat exchangers operation.

The activity of designing thermal systems (space heating and domestic hot water producing) implies the selection and sizing of heat exchangers, based on the use of certain software specific to each equipment. The calculation relations used in design cannot be used in the verification of the heat exchangers thermal performance and therefore it is necessary to identify the real criterial equations specific to each equipment that is tested.

2. NOMENCLATURE

A_0	heat transfer area (m^2)
D	casing diameter (m)
d	tube diameter (m)
d_{ech}	equivalent thermal diameter (m)
G_v	heat carrier volumetric flow-rate (m^3 / s)
t	temperature (°C)
R	thermal resistance (m^2K / W)
U	global heat transfer coefficient (W / m^2K)
h	heat transfer coefficient through forced convection (W / m^2K)
Q	heat flow-rate (W)
k	water thermal conductivity (W / mK)
a	water thermal diffusiveness (m^2 / s)
N	number of measurements
N_0	number of tubes
Nu	Nusselt number
Re	Reynolds number
Pr	Prandtl number
St	Stanton number
f_a	friction coefficient

Greek Letters

ν	kinematic viscosity (m^2 / s)
ε_M	momentum eddy diffusivity (m^2 / s)
ε_T	thermal eddy diffusivity (m^2 / s)
η	thermal efficiency indicator (-)

Subscripts

f	fluid
w	wall
i	inside
e	outside
0	nominal
P	primary
S	secondary
T	inlet
R	outlet
t	turbulent

3. ANALYSIS OF CRITERIAL EQUATIONS OF THE HEAT TRANSFER INSIDE AND OUSIDE THE TUBES

37 criterial equations proposed by various authors were studied and for the analysis only the equations specific to the transitory and turbulent flow referring to water were retained. The analysis was based on the following essential criteria:

– validity range accuracy;

– precision of the author and of the experiment conditions;

– mutual confirmations of various authors;

– phenomenological continuity of criterial equations of the Nu = f(Re) type.

3.1. Forced convection in conditions of transitory and turbulent flow in circular pipes and channels

16 criterial equations were selected which, in terms of the correlation Nu = f(Re) observe at least one of the above mentioned criteria.

Figure 1 presents the Nu = f(Re) correlation curves for which the equations that generates them observe all the previously mentioned selection criteria. The criterial equations are devised by Miheev, Hansen, Timofeev, Ramm, Mc Adams, Dittus and Boetler, Sieder-Tate and Petuhov and refer to water transitory flow by $2300 \leq \mathrm{Re} \leq 10^4$ and in turbulent conditions $10^4 \leq \mathrm{Re} \leq 10^6$. The top limitation of the Re value for turbulent conditions is accounted for strictly by the normal operational conditions of the heat exchangers. As concerns the analysis of the equations specific to turbulent flow, the selected benchmark equation is the empirical relation proposed by Colburn [1] theoretically based

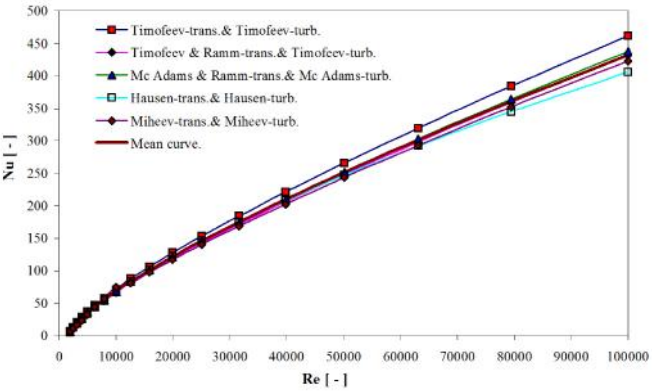

FIg. 1. Nusselt Number Comparison from Different Criterial Equations. Transition to Turbulent and Turbulent Fully Developed Flow inside of Circular Tubes – Pr = 5

on the Chilton-Colburn analogy [2]. We underline that the relations proposed for the turbulent flow analysis are specific to the fully developed flow in smooth circular tubes. All the analyzed equations, except for the Sieder-Tate equation, refer to moderate temperature differences between the pipe walls and the circulated fluid. The Nu number is calculated for water average temperature, namely the arithmetic mean of water temperature at the inlet and outlet of the circular tubes.

Figure 1 presents the criterial equations devised by Hansen, Miheev, Timofeev, Ramm and Mc Adams for transitory flow and by Timofeev, Mc Adams, Hansen and Miheev for turbulent flow. These equations specifically confirm each other and have a phenomenological connecting characteristic of the analysed flow conditions. A compact cluster of curves results which was used in establishing the curve presenting the standard deviation, minimum as against all the curves. The resulting curve actually overlaps the curve representing the criterial equation proposed by Timofeev, both in transitory and in turbulent conditions:

$$\mathrm{Nu} = 0.0263 \cdot \mathrm{Re}^{0.80} \cdot \mathrm{Pr}^{0.35} \cdot (1 - 6 \cdot 10^{5} \cdot \mathrm{Re}^{-1.8}) \tag{1}$$

for transitory conditions, and

$$\mathrm{Nu} = C \cdot \mathrm{Re}^{0.80} \cdot \mathrm{Pr}^{n} \tag{2}$$

cooling: $C = 0.0263; \quad n = 0.35$

heating: $C = 0.0209; \quad n = 0.45$

for turbulent conditions.

Relations (1) and (2) were retained for further calculations.

3.2. Forced convection in turbulent flow along a cluster of tubes inside an envelope

The analysis focused on the criterial equations specific to the flow in the case of a cluster of tubes not equipped with baffles as well as in the case of one equipped with baffles. The aim of this analysis was to eliminate the criterial equations that lead to absurd results (*Nu*–without baffles > *Nu*–with baffles). The *Nu* values are assessed for the equivalent thermal diameter of the flow line and *Re* number for the equivalent hydraulic diameter. The result of the analysis emphasized on one hand the grouping of the $Nu = \mathrm{f}(Re)$ curves specific to the turbulent flow on a baffled line and imposed the acceptation of only one criterial equation for the non-baffled line of flow. In a general form, the three equations are written as follows:

$$\mathrm{Nu} = C \cdot \mathrm{Re}^{m} \cdot \mathrm{Pr}^{0.33} \tag{3}$$

where *C* and *m* are established by the thermal identification presented in this paper. The following is specified for the sizing calculations:

– heat exchangers without baffles:

$$C = 1.16 \cdot d_{ech}{}^{0,6} \cdot \left(\frac{\mathrm{Pr}_f}{\mathrm{Pr}_w}\right)^{0,14}, \quad m = 0.60 \tag{3.1}$$

– heat exchangers with baffles:

$$C = \left(0.22 \div 0.25\right) \cdot \left(\frac{\mathrm{Pr}_f}{\mathrm{Pr}_w}\right)^{0,14}, \quad m = 0.60 \tag{3.2}$$

3.3. Forced convection between flat plates

Four criterial equations were analysed for turbulent flow conditions, of which three can be written as

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

and the fourth is generated by the integration of the energy equation specific to the thermal boundary layer. The solution obtained by Von Karman [2] for the turbulent Prandtl number ($\mathrm{Pr}_t = \varepsilon_M / \varepsilon_T$), $\mathrm{Pr}_t = 1$ is:

$$\mathrm{St} = \frac{0.50 \cdot f_a}{1 + 5 \cdot \left\{0{,}5 \cdot f_a \cdot \left\{(\mathrm{Pr}-1) + \ln\left[1 + \frac{5}{6} \cdot (\mathrm{Pr}-1)\right]\right\}\right\}^{0.50}} \tag{5}$$

For different values of the f_a friction coefficients relation (5) ranges in the rather disperse group of the values obtained by the use of criterial equation (4). Therefore for turbulent flow between flat plates the option is the use of the general character criterial equation (4). According to the technological solution chosen in carrying out the flat plates and to the plate heat exchangers operational conditions, the numerical coefficients of equation (4) are modified.

4. IDENTIFICATION OF THE REAL THERMAL CHARACTERISTIC OF A SHELL AND TUBE HEAT EXCHANGER – THEORETICAL SUBSTANTIATION

The thermal characteristic is defined by the product UA_0, in real operational conditions of the heat exchanger.

A relation identical with that of the flat wall expresses the global heat transfer coefficient, conventionally:

$$U=\left(\frac{1}{h_p}+\frac{1}{h_s}+R_0\right)^{-1} \tag{6}$$

In the case of the heat exchangers used for the producing of the heat carrier necessary in space heating, the primary heat carrier (hot) flows inside the tubes and the secondary one (cold) flows between them. The following hypotheses are devised:

1. The heat transfer area is an invariant of the equipment;
2. The thermal resistance between the flowing lines of the two heat carriers is an invariant and is equal to the value used in the design calculations.
3. Deposits do not affect interior of pipes.

Neither of these hypotheses is in accordance with the physical reality, but eliminate any supposition concerning the real number of operational tubes or the thermal resistance of the deposits on the external part of the tubes. Inside the tubes the water flown is treated, so the hypothesis concerning the lack of deposits is justified.

The previous elements prove that all the imperfections of the heat transfer are taken over, from the mathematical point of view, by the h_e heat transfer coefficient.

By processing the criterial equations (1) and (2) is obtained the approximate relation, unique for any type of flow (transitory and turbulent), for the assessment of the h_i coefficients:

$$h_P=0.032669\cdot\left(\frac{G_v}{N_0}\right)_P\cdot d^{-1.80}\cdot\psi_i\cdot\left(\bar{t}_P\right) \tag{7}$$

where:

$$\psi_i\cdot\left(\bar{t}_P\right)=\left(\frac{k_f}{\nu_f^{0.45}\cdot a_f^{0.35}}\right)_P \tag{7.1}$$

By carrying out j measurements we obtain j values of the U_j global heat transfer coefficient:

$$U_j=\frac{Q_s}{A_0}\cdot\frac{\ln\dfrac{t_{T_{Pj}}-t_{T_{Sj}}}{t_{R_{Pj}}-t_{R_{Sj}}}}{t_{T_{Pj}}-t_{T_{Sj}}-(t_{R_{Pj}}-t_{R_{Sj}})} \tag{8}$$

Equation (6) provides values of h_{ej} coefficient:

$$h_{S_j}=\left(\frac{1}{U_j}-\frac{1}{h_{P_j}}-R_0\right)^{-1} \tag{9}$$

with the values h_{S_j}, real, assessed taking into account the characteristics of the flow and of the heat exchanger geometry as well, are established the Nu_{Sj}, Re_{Sj} and Pr_{Sj} values

$$\mathrm{Nu}_{S_j}=\frac{h_{S_j}\cdot\left(D^2-N_0\cdot d_e^2\right)}{k_{s_j}\cdot N_0\cdot d_e} \tag{10}$$

$$\mathrm{Re}_{s_j}=1.27324\cdot\frac{G_{v_{Sj}}}{\nu_{s_j}\cdot\left(D+N_0\cdot d_e\right)} \tag{11}$$

$$\mathrm{Pr}_{s_j}=\left(\frac{\nu}{a}\right)_{s_j} \tag{12}$$

The criterial equation specific to the flow among tubes is written as

$$\overset{*}{\mathrm{Nu}}=C\cdot\mathrm{Re}_s^m \tag{13}$$

where

$$\overset{*}{\mathrm{Nu}}=\mathrm{Nu}_s\cdot\mathrm{Pr}_s^{-0,33} \tag{14}$$

By applying equation (13) is obtained the “ln” operator:

$$\ln \overset{*}{\mathrm{Nu}} = m \cdot \ln \mathrm{Re}_s + \ln C \qquad (15)$$

By noting $A = m$; $B = \ln C$; $Y = \ln \overset{*}{\mathrm{Nu}}$ and $X = \ln$ Re, relation (15) becomes:

$$Y = A \cdot X + B \qquad (16)$$

with unknown values of A and B which do not vary depending on values Nu_S, Re_S and Pr_S (this condition is met by reaching the steady-state flow and heat transfer).

For the j measurements performed we obtain the lots $\{X\}$ and $\{Y\}$ and values A and B are obtained by applying the method of the least squares. The final result is:

$$\begin{cases} A \equiv m \\ C = \exp B \end{cases} \qquad (17)$$

The numerical values m and C obtained reflect the real condition of the heat exchanger in terms of heat transfer. The practical usefulness is represented by the testing of the efficiency of the heat transfer within the equipment and the correcting of the thermal adjustment curves of the primary heat carrier with a view to reach thermal comfort in the heated spaces. Practically, for testing the heat transfer efficiency, the lots of values $\{t_{T_P}\}$, $\{t_{R_S}\}$, $\{G_{v_P}\}$ and $\{G_{v_S}\}$ are assessed. The outlet functions are formed of the lots of values $\{i_{R_P}\}$ and $\{i_{R_S}\}$.

The thermal balance equations specific to the tested device are non-linear algebraic equations that are solved by a well-known method (e.g. Kani). The non-linearity of the equations system is accounted for by the logarithm function in the temperature mean difference as well as by the dependency of coefficients h_i and h_e of the heat carriers average temperatures. In terms of values, the resulted unknown values and are compared to the similar values characteristic to a new heat exchanger. For this case is used the same thermal balance equations system where and are obtained based on the criterial equations used in the heat exchanger sizing (equation (2) for the calculation of and equation (3) with the explanation (3.1) for the calculation of). Values and are assessed. The following indicator is generated:

$$\eta_j = \frac{t_{T_{P_j}} - i_{R_{P_j}}}{t_{T_{P_j}} - t_{R_{P0j}}} = \frac{t_{T_{S_j}} - i_{R_{S_j}}}{t_{T_{S0j}} - t_{R_{S_j}}} \le 1 \qquad (18)$$

reflecting the heat transfer depreciation rate in the tested heat exchanger.

5. IDENTIFICATION OF THE REAL THERMAL CHARACTERISTIC OF A PLATE HEAT EXCHANGER – THEORETICAL SUBSTANTIATION

The criterial equation (4) is specific to the heat transfer in both loops so that the convection heat transfer coefficients h_P and h_S are written as:

$$h_P = A \cdot M_P \cdot \left(\frac{G_{v_P}}{v_P} \right)^m \qquad (19)$$

$$h_S = A \cdot M_S \cdot \left(\frac{G_{v_S}}{v_S} \right)^m \qquad (20)$$

where

$$M_P \approx 0{,}95 \cdot k_P \cdot Pr_P^n \qquad (21)$$

$$M_S \approx 1{,}06 \cdot k_S \cdot Pr_S^n \qquad (22)$$

The expression of the global heat transfer coefficient U_{Th} (Th-theoretical) is written:

$$U_{Th} = \left[\left(A \cdot M_P\right)^{-1} \cdot \left(\frac{G_{v_P}}{v_P} \right)^{-m} + \left(A \cdot M_S\right)^{-1} \cdot \left(\frac{G_{v_S}}{v_S} \right)^{-m} + R \right]^{-1} \qquad (23)$$

where values k, v, Pr are assessed according to the average temperatures of the two heat carriers. For each measurement performed in heat transfer and flow steady-state conditions, the real coefficient U_j is determined:

$$U_j = \frac{Q_j}{A_0} \cdot \frac{\ln \dfrac{t_{T_{P_j}} - t_{T_{S_j}}}{t_{R_{P_j}} - t_{R_{S_j}}}}{t_{T_{P_j}} - t_{T_{S_j}} - \left(t_{R_{P_j}} - t_{R_{S_j}} \right)} \qquad (24)$$

where Q_j represents the arithmetic mean of the heat flow-rates specific to the primary and secondary heat carrier loops in acceptability conditions specific to the measurement activity by delimitation of the differences between and at maximum 3 %.

The error between the theoretically assessed value and the experimentally assessed value U_j is established by the relation:

$$\Delta\varepsilon_j = \left(1 - \frac{U_{Th_j}}{U_j} \right) \cdot 100 \ [\%] \qquad (25)$$

The condition is imposed that the sum total of deviations for all the measurements should be null, as this is a condition necessary for the thermal identification procedure.

$$\sum_j \Delta\varepsilon_j = \sum_j \left(1 - \frac{U_{Th_j}}{U_j}\right) \cdot 100 = 0 \qquad (26)$$

In the first phase the condition $R = R_0$ has to be met, specific to the new equipment. Then a value $n_0 \in [0.10; 0.50]$ is imposed. Equation (26) provides one pair of values m_k and A_k that fulfills the equation. Therefore for the established R_0 and $n_{0,}$ distinct values m_k and A_k result. With each pair $\{m_k, A_k\}$ is established the standard deviation of the errors for the N measurements performed:

$$\sigma_{k_{n0},\,R0} = \sqrt{\frac{\sum_{1=1}^{N} \left(\Delta\varepsilon_j\right)^2}{N \cdot (N-1)}} \qquad (27)$$

Value R_0 is preserved unmodified and the new value $n_1 = n_0 + \Delta n$ (Δn = 0,05) is used; the solution of equation (26) provides new pairs of values $\{m_k, A_k\}$. The following set of values is determined and the calculation continues until the range of values tested for coefficient n is completed. Two new sets of values $\{n\}$ and are obtained, where the notation is used. Each value n is associated with a value and is considered and selected as valid the value associated with where . Therefore for $R = R_0$ resulted two values n and m that lead to the minimum standard deviation between the calculated and measured U values. The procedure is resumed by modifying value $R_1 = R_0 + \Delta R$ and in general $R_q = R_{q-1} + \Delta R$.

The final solution is obtained for the group of values n, m and R leading to the minimum standard deviation defined by relation (27). From the mathematical point of view the identification of the real thermal characteristic of a plate compact heat exchanger is synonym with a conditioned minimum problem. The function to be minimized is the standard deviation (27) and the condition is imposed by equation (24). The independent variables are m and n numerical coefficients and the value of R thermal resistance, with the condition $R - R_0 \geq 0$ met.

6. CASE STUDIES

The paper further presents two applications of the thermal identification procedures described in the previous chapters.

6.1. Identification of the thermal characteristic of the shell and tube heat exchanger in the INCERC – Bucharest Thermal substation

The equipment that is the object of this case study is used for the heating of the secondary heat carrier flown in the central heating system of the laboratory of the institute. It is made in Romania and has been operational for 17 years. Inside a part with a diameter of 0.257 m are placed 55 steel tubes, each with an inside diameter of 0.016 m. The measurements were performed in the current operational conditions of the Thermal substation. The equipment used for taking over the thermodynamic parameters is formed of an ultrasonic flow-meter Parametrics DF 868 with two measurement channels, four RTD PT 100 and an automatic Data Logger DT 50 connected to a portable computer. The measured parameters were recorded once every minute by averaging the read values every 10 seconds.

Figures 2 and 3 present the variation of the heat carriers volume flow-rates namely of the heat carriers temperatures. Based on the analysis of the measured data, 9 distinct operational set of conditions were

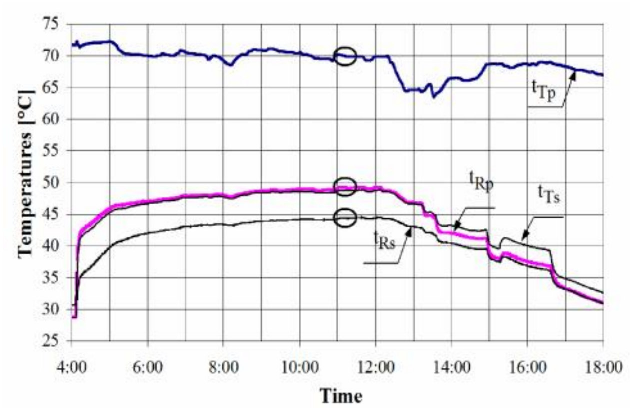

Fig. 2. INCERC Thermal Substation – Measured Temperatures

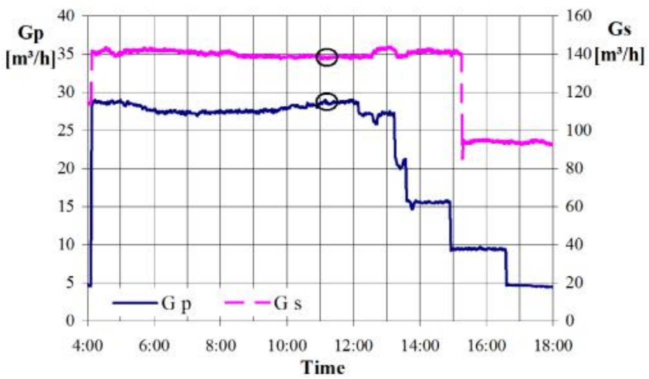

Fig. 3. INCERC Thermal Substation – Measured Flow-Rates

selected for which the parameters average values were assessed, included in table 1.

The processing of the measured data leads to the real and theoretical U values, as well as U_{Th} included in table 2. The statistical analysis provided values of the numerical coefficients that are specific to the criterial equation specific to the convection heat transfer in the space between the envelope and the tubes.

The following is obtained:

$$\mathrm{Nu} = 0.9601 \cdot \mathrm{Re}^{0.346} \cdot \mathrm{Pr}^{0.33} \tag{28}$$

while the correlation rate of the linear regression represented by relation (16) is $r^2 = 0.992$.

Table 1.

Meas. No.	G_P [m³/h]	G_S [m³/h]	t_{Tp} [°C]	t_{Rp} [°C]	t_{Ts} [°C]	t_{Rs} [°C]
1	11.48	74.16	69.46	37.25	40.36	35.33
2	11.65	73.56	72.48	38.43	41.74	36.51
3	23.91	72.89	72.70	48.76	50.07	42.07
4	29.00	73.44	72.61	51.97	52.90	44.68
5	29.43	131.03	73.60	50.20	49.60	44.35
6	28.99	132.59	72.45	50.46	50.08	45.19
7	12.28	113.18	71.96	39.90	42.09	38.67
8	27.30	137.72	70.38	47.99	47.75	43.23
9	28.92	139.00	70.22	49.26	48.82	44.40

Table 2.

Meas. No.	Q_P [kW]	Q_S [kW]	Q_{ave} [kW]	Error [%]	U [W/m²K]	U_{Th} [W/m²K]
1	419.4	429.3	424.34	– 2.3%	568.9	1,188.5
2	449.5	442.1	445.8	1.7%	574.7	1,207.9
3	648.6	667.8	658.2	– 2.9%	673.9	1,669.3
4	678.4	690.6	684.5	– 1.8%	734.3	1,812.7
5	779.9	787.7	783.8	– 1.0%	816.7	2,257.1
6	722.5	742.9	732.7	– 2.8%	829.7	2,255.3
7	446.1	445.3	445.7	0.2%	664.8	1,404.7
8	693.2	713.9	703.5	– 2.9%	821.9	2,199.5
9	687.6	704.7	696.1	– 2.5%	835.7	2,270.8

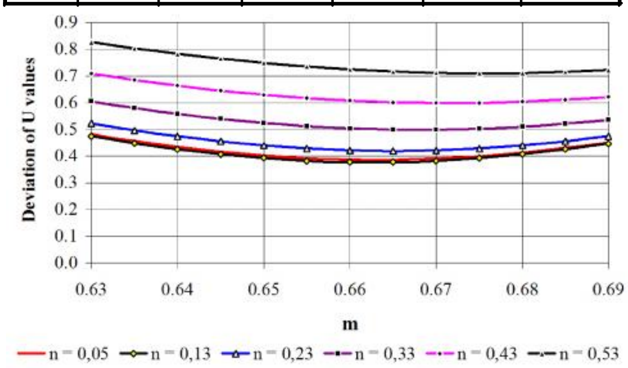

Fig. 4. Thermal Identification of a Plate Heat Exchanger (M6-MFG) – INCERC Laboratory Stand

The heat transfer efficiency indicator in nominal operational conditions ($t_{T_{P0}} = 150°C$, $t_{R_{S0}} = 75°C$, $G_{v_{P0}} = 40.75\ \mathrm{m^3/h}$, $G_{v_{S0}} = 132.67\ \mathrm{m^3/h}$) has the value $\eta_0 = 0.766$ attesting a serious damaging of the heat transfer between the heat carriers.

6.2. Identification of the thermal characteristic of the compact plate heat exchangers

The procedure presented in the paper was used in identifying the values of the numerical coefficients of the criterial equation and of the thermal resistance between the primary and secondary heat carriers, specific to a plate heat exchanger. The tested equipment is of the M6-MFG type. The experimental analysis was performed on the laboratory stand of the Department of Building Services and Efficient Use of Energy in Buildings – INCERC Bucharest.

The processing of the measured data led to the assessment of the U_j values, $j \in [1, 15]$. The analysis of standard deviation minimizing was performed for two values of the thermal resistance, namely $R_0 = 2.7 \cdot 10^{-5}\ \mathrm{m^2K/W}$ and $R_1 = 12.7 \cdot 10^{-5}\ \mathrm{m^2K/W}$ (scale deposits with a thickness of $2 \cdot 10^{-3}$ m).

The variation σ_k according to m and n for the two values R_q leads to the minimum values $\dot{\sigma}_{k,R_0} = 0.385$, respectively $\dot{\sigma}_{k,R_1} = 0.81$.

Therefore $R_0 = 2.7 \cdot 10^{-5}\ \mathrm{m^2K/W}$. For this value of the thermal resistance Figure 4 presents variation $\sigma_k = \sigma_k(m, n)$. The final values of the identification operation result:

$$n = 0.13$$
$$m = 0.665$$
$$A = 56.02197$$

as well as the expressions of the convection heat transfer coefficients:

$$h_P = 53.22087 \cdot k_P \cdot \mathrm{Pr}_P^{0.13} \cdot \left(\frac{G_{v_P}}{\nu_P}\right)^{0.665} \tag{29}$$

$$h_S = 59.38329 \cdot k_S \cdot \mathrm{Pr}_S^{0.13} \cdot \left(\frac{G_{v_S}}{\nu_S}\right)^{0.665} \tag{30}$$

The verification of the identification operation is obtained by comparing the real and theoretical values of the U_j and U_{Thj} global heat transfer coefficients. The result is presented in the diagram in Figure 5. The value of the maximum deviation (j = 13) is $\Delta\varepsilon_{j=13}$ = – 3.86 % and the other values range within [– 1.36 %; 1.76 %]. Therefore the thermal identification operation is considered correct, as the deviations range between acceptable limits.

7. CONCLUSION

The identification of the real thermal characteristic of the operational heat exchangers consists in the accurate selection of the criterial equation based on which are assessed the real coefficients of forced convection heat transfer h_P and h_S.

At the same time in the case of the plate heat exchangers is also assessed the real value of the thermal resistance between the flow lines of the two heat carriers.

The study establishes the representative type of criterial equations of $Nu = f(Re)$ type for transitory and turbulent flow conditions.

The assessment of the numerical coefficients specific to the criterial equations for flow in the space between the tubes and the envelope in the case of shell and tube heat exchangers is based on the statistical processing of the measured data, in the form of a regression straight line and the angular coefficient and the free term lead to the values to be established, according to relation (17). In the case of plate heat exchangers, the problem of assessing the coefficients of the criterial equations and of the thermal resistance between the flow lines of the heat carriers is one of conditioned minimum. The minimized function is the standard deviation between the theoretical and measured global heat transfer coefficients.

The study includes two case studies aimed to explain by real cases the procedures presented in the chapters devoted to the theoretical substantiation. The errors obtained between the global heat transfer coefficients obtained by processing the measured data and by applying the calculation relations ranges under 4 %. It can therefore be stated that both the procedures presented and the criterial equations resulted are accurate enough to allow the estimation of the heat transfer real characteristic of the existing heat exchangers as well as of the η indicator of heat transfer efficiency. At the same time the criterial equations resulted allow the modeling of the heat transfer processes at the level of the cogeneration heating systems with a view to adapt the operational parameters to the thermal comfort conditions [3].

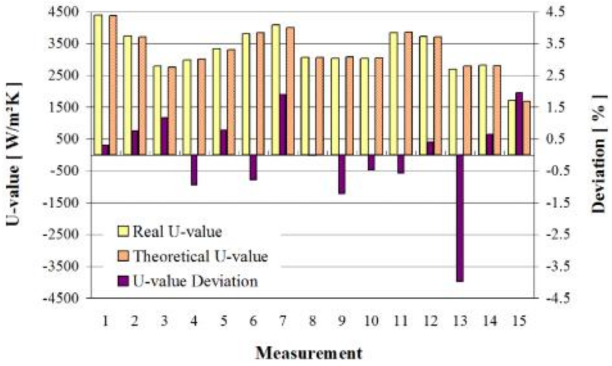

Fig. 5. Comparison Between Theoretical and Real U-values. M6-MFG Heat Exchanger – INCERC Laboratory Stand

REFERENCES

[1] Incropera, P. F., DeWitt, P.D. – *Fundamentals of Heat Transfer*, John Wiley & Sons Inc., p. 406, 1981

[2] Bejan, A. – *Convection Heat Transfer*, John Wiley & Sons Inc., p. 235, p. 251, 1984

[3] Constantinescu, D., Petran, H. – *Evaluation of Bucharest District Heating System Efficiency*, UTCB Contr. No. 191 / 2001 (in Romanian), 2001