CONTRIBUTIONS REGARDING THE EXERGETIC ANALYSIS OF HEAT EXCHANGERS

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***Rezumat.** Pentru a studia factorii care influenţează exergia agenţilor de lucru a unui schimbător existent trebuie determinate temperaturile agenţilor de lucru la ieşirea din schimbător. Pentru un schimbător de căldură în curent încrucişat, la care analiza exergetică se justifică datorită temperaturilor ridicate ale agentului primar, este dificilă determinarea temperaturilor agenţilor la ieşire.*

Această lucrare prezintă o metodă numerică de determinare a temperaturilor agenţilor de lucru la ieşirea dintr-un schimbător de căldură în curent încrucişat şi utilizarea acestor temperaturi în analiza exergetică a schimbătoarelor de căldură.

Se prezintă de asemenea o aplicaţie numerică, rezultatele modelării matematice şi concluziile care se desprind.

1. INTRODUCTION

The exergy method is widely used because it allows us to appreciate the thermodynamic performance, by taking into account the ireversibilities of the processes. The use of exergy method at heat exchanger is useful because it develops some important irreversibilities, especially due to heat transfer at the finite difference temperature. For applying exergy analysis at cross-flow heat exchangers, (used in energetic industry) the outlet agents' temperatures must be known.

This paper presents a method for the calculus of the outlet temperatures of the working agents. This analysis is performed for an existing heat exchanger for which working parameters are modified.

2. DETERMINATION OF THE OF THE WORKING AGENTS TEMPERATURES AT THE OUTPUT OF THE HEAT EXCHANGERS

For current flow and co-current flow heat exchangers, in the specific literature there exist certain relations for the calculus of outlet temperatures of working agents depending on inlet temperatures, specific heat, the overall heat transfer coefficient and the surface heat transfer.

$$t_f = f\,(t_i, \Delta t_i, C_1, C_2, \mathrm{k}, S) \qquad (1)$$

where: t_f – outlet temperature [°C]; t_i – inlet temperature [°C]; C_1 – thermal capacity [J/K]; C_2 – thermal capacity [J/K]; k – overall heat transfer coefficient [W/m^{2}K], S – surface heat transfer [m^2].

For a cross-flow heat exchanger, it is very difficult to determine the final temperatures; in these conditions it is very important to develop a numerical method for the calculus of output temperatures for an existing heat exchanger for which functional parameters are modified. For example, if the fluid velocity is changed, the overall heat transfer coefficient will be changed, the thermal flux, the mean logarithmic temperature and the outlet temperatures will be changed. We assume that the following dimensions are known: geometry of the heat exchanger, inlet temperatures of the primary fluid t_{11}, the inlet temperature of the secondary fluid t_{21}, heat exchange surface.

The relation for the calculus of thermal flux on the basis of the heat transfer equation is:

$$Q = k \cdot S \cdot \Delta t_m \ \text{[kW]} \qquad (2)$$

where: $\Delta t_m = \varepsilon \cdot \Delta t_{mcc}$ (3)

ε – correction factor; Δt_{cc} – mean logarithmic temperature for co-current flow;

$$\varepsilon = f_1\left(\frac{t_{22} - t_{21}}{t_{11} - t_{21}}, \frac{C_2}{C_1}\right) \qquad (4)$$

C_1, C_2 are thermal capacities of the fluids.

$$\Delta t_{mcc} = \frac{\Delta t_{max} - \Delta t_{min}}{\ln \dfrac{\Delta t_{max}}{\Delta t_{min}}} \qquad (5)$$

On the other hand, the thermal flux can be written as a function of fluid temperature difference of the working agents, by using calorimetric relation:

$$Q = C_1 \cdot (t_{11} - t_{12}) = C_2 \cdot (t_{22} - t_{21}) \ \text{[kW]} \qquad (6)$$

The relations no (2) and (6) are used for outlet temperatures of the working agents determination (t_{12} and t_{22}). It can be notice that in the equation no (2) the Δt_m is a function of t_{12} and t_{22} so it's necessary to resolve the equation system (2)(3)(4)(5)(6).

While resolving this system, the instability of the solution appears because of equations no linearity. In order to eliminate this disadvantage, the surface heat transfer S has been adopted as a variable and it was assumed (for the beginning) one of the temperatures t_{12} or t_{22} known and the others result on the basis of relation no. (6). By using this methodology the equations (2)....(5) allowed the determination of a hypothetic surface heat transfer S_x; for that surface, the temperatures of the working agents will have the imposed values. For example if the temperature t_{12} is imposed it results:

$$Q_c = C_1 \cdot (t_{11} - t_{12}) \ \text{[kW]}; \quad t_{22} = t_{21} + \frac{Q_c}{C_2} \qquad (7)$$

By knowing the temperatures and the thermal capacities, the Δt_{mcc} for co-current flow can be determined with relation no (5). If knowing the temperatures and the thermal capacities the Δt_{mcc} can be determined (on the basis of the relation no 5).

The parameters R and P are calculated with relations (according to{1}):

$$R = \frac{C_2}{C_1}; \quad P = \frac{t_{22} - t_{21}}{t_{11} - t_{21}} \qquad (8)$$

Depending on the heat exchanger type, ε is approximated by 3-th grade equation:

$$\varepsilon = a + b \cdot P + c \cdot P^2 + d \cdot P^3 \qquad (9)$$

where the coefficients a, b, c and d are obtained in function of R on the basis of diagrams existing in specific literature.

Assuming known the overall heat transfer coefficient (determined in function of the working agents velocities), from equation no (2) it is obtained:

$$S_x = \frac{Q_c}{k \cdot \Delta t_m} [m^2] \qquad (10)$$

If $S_x = S$, then the temperatures t_{11} and t_{22} represent the solutions of the problem.

If $S_x < S$, the thermal flux resulted from calorimetric equation Q_c is less than thermal flux resulted from the relation of heat transfer (2), so means that the start temperature for the calculus is too small.

If $S_x > S$, on the same judgment, it can be noticed that the start temperature for the calculus is too high.

For the last two situations, it is performed an iterative calculus, maximizing or minimizing the start temperature (t_{11} or t_{22}) until $Q_c = Q$, so that:

$$\Delta S = \frac{S - S_c}{S} < \varepsilon_s \qquad (11)$$

where ε_s must be very small (the precision of the temperatures determination depends by its value). The proposed calculus algorithm is convergent and it reaches quickly at the solution of the problems.

It has been elaborated an application. For this there have been adopted as input dimensions: thermophisical properties of the fluids: (ρ_1, ρ_2, λ_1, λ_2, υ_1, υ_2, Pr_1, Pr_2), flow speed of the agents w_1 and w_2, flow section (S_{c1}, S_{c2}), heat exchange surface S and inlet temperatures t_{11} and t_{12} of the working agents.

On the basis of the known criterion relations there have been determined α_1 and α_2 and the coefficient k. Also the fluid speed has been adopted and the thermal capacities C_1 and C_2 resulted. Using the relation no (7) there have been calculated Q_c and t_{22} and the coefficients a, b, c, d were determinated. For the heat exchanger, that analysis was performed, the values of the coefficients (a, b, c, d) are presented on the table no. 1. The program is concepted so that it chooses automatically these coefficients from the table, in function of R. The surface S_x was calculated on the basis of relation no (10) and ΔS with relation no (11). The imposed value for ε_s was 0.001.

3. WORKING AGENTS EXERGY CALCULUS

By numerical modeling of the thermal processes in cross-flow heat exchangers it's useful to determine the following: the exergy variation of the working agents at the entrance and output of the heat exchanger, the exergy losses, the exergy efficiency, in function of flow speed of the working agents (irrespective of the mass flow).

The working agents exergy at the inlet and outlet are determined with the relations (according to {2} and {3}):

$$\begin{aligned}
\dot{E}_{11} &= \dot{H}_{11} - \dot{H}_{1ma} - T_{ma} \cdot (\dot{S}_{11} - \dot{S}_{1ma}) \text{ [kW]};\\
\dot{E}_{12} &= \dot{H}_{12} - \dot{H}_{1ma} - T_{ma} \cdot (\dot{S}_{12} - \dot{S}_{1ma}) \text{ [kW]};\\
\dot{E}_{21} &= \dot{H}_{21} - \dot{H}_{2ma} - T_{ma} \cdot (\dot{S}_{21} - \dot{S}_{2ma}) \text{ [kW]}; \qquad (12)\\
\dot{E}_{22} &= \dot{H}_{22} - \dot{H}_{2ma} - T_{ma} \cdot (\dot{S}_{22} - \dot{S}_{2ma}) \text{ [kW]};
\end{aligned}$$

where the entropy variation (is calculated with the following equations:

$$\begin{aligned}
\dot{S}_{11} - \dot{S}_{1ma} &= \dot{C}_1 \cdot \ln \frac{T_{ma}}{T_{11}} \text{ [kW/K]};\\
\dot{S}_{12} - \dot{S}_{1ma} &= \dot{C}_1 \cdot \ln \frac{T_{ma}}{T_{12}} \text{ [kW/K]};\\
\dot{S}_{21} - \dot{S}_{2ma} &= \dot{C}_2 \cdot \ln \frac{T_{ma}}{T_{21}} \text{ [kW/K]}; \qquad (13)\\
\dot{S}_{22} - \dot{S}_{2ma} &= \dot{C}_2 \cdot \ln \frac{T_{ma}}{T_{22}} \text{ [kW/K]};
\end{aligned}$$

and the enthalpy variation with the relations :

$$\begin{aligned}
\dot{H}_{11} - \dot{H}_{1ma} &= \dot{C}_1 \cdot (T_{11} - T_{ma}) \text{ [kW]};\\
\dot{H}_{12} - \dot{H}_{1ma} &= \dot{C}_1 \cdot (T_{12} - T_{ma}) \text{ [kW]}; \qquad (14)\\
\dot{H}_{21} - \dot{H}_{2ma} &= \dot{C}_2 \cdot (T_{21} - T_{ma}) \text{ [kW]};\\
\dot{H}_{22} - \dot{H}_{2ma} &= \dot{C}_2 \cdot (T_{22} - T_{ma}) \text{ [kW]}.
\end{aligned}$$

The exergy balance for the heat exchangers is (according with {4}):

$$\dot{E}_{11} + \dot{E}_{21} = \dot{E}_{12} + \dot{E}_{22} + \Pi \text{ [kW]} \qquad (15)$$

while the exergy losses due to friction in the heat exchanger are neglected.

$$\begin{aligned}
\Pi &= (\dot{E}_{11} + \dot{E}_{21}) - (\dot{E}_{12} + \dot{E}_{22}) = \\
&= (\dot{E}_{11} - \dot{E}_{12}) - (\dot{E}_{22} - \dot{E}_{21}) \quad \text{[kW]}
\end{aligned} \qquad (16)$$

Table 1. Heat exchanger coefficients values

R	0.2	0.4	0.6	0.8	1	1.5	2	3	4
a	1.996	0.56	0.5	1.5	0.613	0.6	0.94	3.722	2.5
b	-5.912	2.075	2.48	-2.952	2.429	3.817	0.983	-50.11	-33.53
c	11.229	-3.125	-4	5.375	-4.75	-11	-4	276.182	241
d	-6.762	1.25	1.66	-3.75	2.083	8.333	1.667	-479.97	-566.66

The exergetic efficiency will be:

$$\eta_{ex} = 1 - \frac{\dot{\Pi}}{\dot{E}_{11} + \dot{E}_{12}} [-] \qquad (17)$$

4. THE NUMERICAL APPLICATION

In order to check the proposed model, it was performed an exergy analysis for a plate heat exchanger for the gases heat recovery rejected from a gas turbine installation with open circuit [1, pg.405].

The heat exchange surface it is realized with discontinuous fins, stainless steel with thermal conductivity $\lambda = 20$ W/mK. The geometrical characteristics of the heat exchanger are: equivalent diameter of the flow canal $d_e = 2.64$ mm, surface fins and total heat transfer surface ratio $Sn/S = 0.873$, base plate surface and total heat transfer surface ratio $S_0/S = 0.127$, total heat transfer surface and recuperator volume ratio $A/V = 1386$ m^2/m^3, fins thickness $\delta_n = 0.152$ mm, distance between plates $s = 10.52$ mm and plates thickness $\delta_n = 0.5$ mm.

The working conditions for the heat exchanger: mass flow rate of the air $m_1 = 24$ kg/s mass flow rate of the gases $m_2 = 24{,}4$ kg/s, inlet air temperature t_{21} = 177 ºC, inlet gases temperature $t_{11} = 427$ ºC.

The obtained results are presented in diagrams no. 1–4.

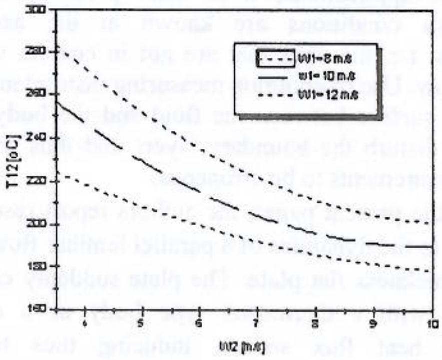

Fig. 1. Outlet temperature variation of the primary working agents as function of secondary fluid velocity.

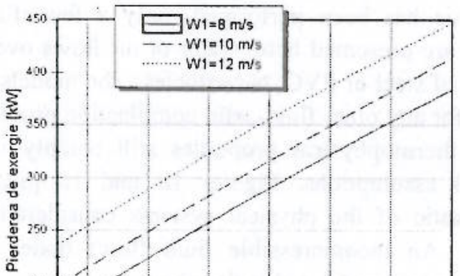

Fig. 3. Exergy losses as a function of secondary fluid velocity.

CONCLUSION

This paper presents an original method for the calculus of working agents outlet temperatures in cross flow heat exchangers. The proposed method was applied on exergetic analysis for an existing cross flow heat exchanger.

While analysis the obtaining result there have been establish the following conclusions:

- Temperatures values at the heat exchanger output on the basis of the model are: (for the air velocity of 5 m/s and for gases velocity of 10 m/s) temperature 375 ºC for air and 230 ºC for gases. In the heat exchanger example [1, pg. 405] the working agents outlet temperatures are 240 ºC for gases and 364 ºC for air. It can be notice that obtaing results on the basis of the model are very closed on the temperatures values from the example (with an error of 4%). In this conditions we can appreciate that the model is valid.
- While secondary fluid velocity increase, the exergy losses increase as well; for the primary fluid velocity of 8 m/s the exergy losses are smallest.
- The exergetic efficiency has the maximum values for low velocities values of the secondary fluids, and for the primary agent velocity of 8 m/s.

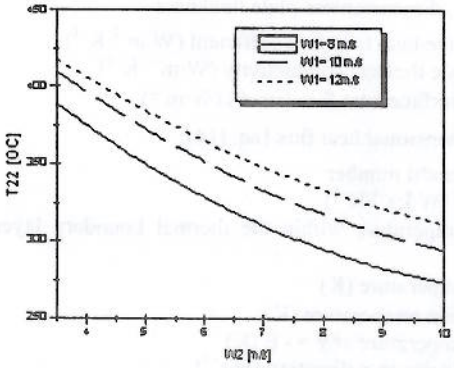

Fig. 2. Outlet temperature variation of the secondary working agent as function of secondary fluid velocity.

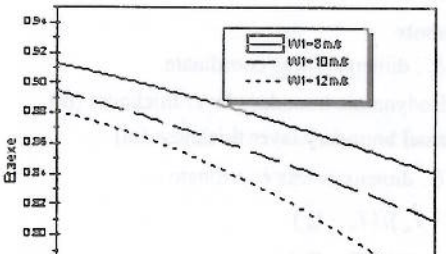

Fig. 4. Exergetic efficiency as a function of secondary fluid velocity.

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