THE ANALYSIS OFTHE MATRIX HEAT EXCHANGER EFFECTIVENESS

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The goal of this paper is to establish the optimal operating regime of the observed perforated plate air/water heat exchanger in a wide range of parameters. The experimental investigation was carried out in a package of three perforated plates which were placed in the experimental chamber and heated by hot water. A fan with the variable air volume flow was connected to the experimental chamber, so the air flow rates varied from 100 to 300 m^3/h, while the water flow varied from 0.03 to 0.06 m^3/h. The thermocouples were attached to the surface of the middle perforated plate in the package along its upwind and downwind sides, as well as at the inlet and outlet of the chamber and between the perforated plates. During each experiment, the readings of thermocouples were recorded alongside with the air and water volume flow and temperatures of water at the inlet and outlet of the chamber. In order to predict the performance of the observed perforated plate heat exchanger, NTU-Effectiveness analysis was performed on the basis of the experiment results and analytical relations. Experimental results showed that the effectiveness of the perforated plate heat exchanger can be calculated the same as for the concentric tube counter flow. At the end of the paper, the optimal operating point in the range of varied parameters was determined.

Keywords: effectiveness, matrix heat exchanger, NTU, perforated plate

Introduction

One of the most important properties of heat exchangers, apart from having high effectiveness, is the need to be very compact *i.e.*, they must accommodate a large surface to volume ratio. This helps in controlling the heat exchanger exposure to the surroundings by reducing the exposed surface area, which also means a smaller heat inertia. This requirement is particularly important for small refrigerators operating at the liquid helium temperature.

The need for attaining high effectiveness and high-level of compactness together in one unit led to the invention of the perforated plate heat exchanger, or matrix heat exchangers (MHE) by McMation et al. [1]. A matrix heat exchanger consists of a package of perforated plates with a multitude of flow passages aligned in the direction of flow, allowing a high heat transfer in a proper design unit. These exchangers can have up to 6000 m^2/m^3 surface to volume ratio, much higher than for a typical shell-and-tube heat exchanger [2,3]. A schematic of a MHE is presented in Figure 1.

Besides their high surface-to-area density, MHE has better fluid flow characteristics, low axial heat conduction and manufacturing advantages over many other types of compact heat exchangers. MHE have become attractive for cryogenic applications [4,5,6]. Perforated plates are also used as solar collectors absorbers or fins for electronic equipment cooling [7,8]. Although MHE does not represent a novelty, still they are not widely adopted as a heat exchanger device, especially at ambient temperatures. In this paper, a specific design of MHE was tested [9,10]. In order to predict its performance, measurements and analytical analysis were carried out.

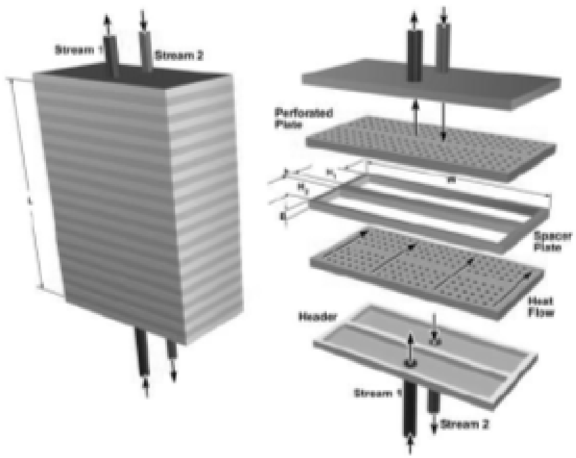

Figure 1. Matrix heat exchanger schematic [2,3]

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Experimental

A package of three plates, sized 740x145 mm, 2 mm thick, with square arranged perforations with a diameter of 2 mm was tested in the experiment. The plate has been divided into two sections: the central section through which the water flows and the peripheral section through which the air flows. The sections were separated by a gasket (Figure 2). The package was placed in the channel of the experimental chamber, at which the entrance was a thrust fan. The schematic of the installation is presented in Figure 3.

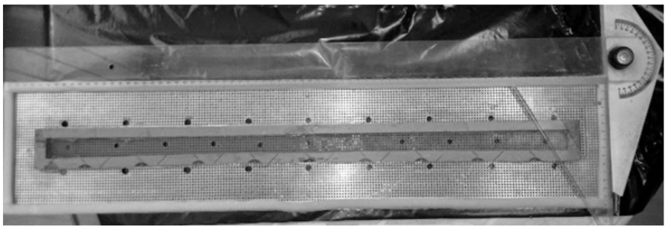

Figure 2. Perforated plate with a gasket

Water was used as a heating fluid. Water enters the collector and flows through the plate. Along the water flow, heat is transferred from the water to the plate. The exchanged heat is further transferred by conduction through the plate towards the edge of the plate, where it comes into contact with the air stream. The heat is then transferred by means of convection from the plate to the cooler air stream.

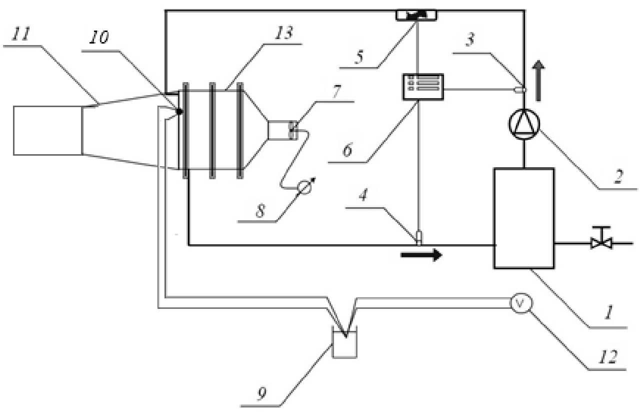

Figure 3. Measuring installation: 1 -boiler, 2 -circulation pump, 3,4 - pt100 probes, 5 - water flow meter, 6 - Kamstrup acquisition unit, 7 - fan, 8 - fan regulation, 9 - cold end, 10 - thermocouples, 11 - Alnor Balometer air flow measuring unit, 12 - Voltmeter Hewlett-Packard 3468A, 13 - air chamber with the perforated plates package

For the needs of the experiment the thermocouples were set in the middle perforated plate. In total, 11 thermocouples were placed, 5 on each side of the plate (Figure 4) and one as control thermocouple for error estimation. Heads of thermocouples were covered with thermally conductive paste in order to ensure a thermal contact between thermocouples and the plate. Thermocouples were calibrated before the experiment.

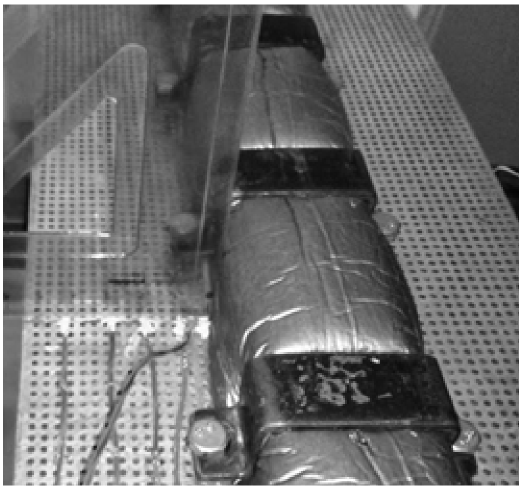

Figure 4. Thermocouples position on the perforated plate

The cold end of thermocouples was obtained as a mixture of water and ice. During each experiment, the air flow, water flow, temperatures of the perforated plate, as well temperatures of air and water at the inlet and the outlet of the chamber and temperatures of air between the plates were measured. The measurements were conducted when the thermal equilibrium was achieved, *i.e.* the change of the water temperature was less than 0.1 K in 10 minutes [11].

For the aims of the research, the parameters varied in the following range:

- Volume flow rate of the water $\dot{V}_w$=(0.03-0.06) m^3/h,
- Volume flow rate of the air $\dot{V}_L$=(100-300) m^3/h.

Throughout measurements, the temperature of water at the inlet was ≈52 ºC, while the temperature of the air was ≈25 ºC

Error estimation

The convective heat transfer rate $\dot{Q}_w$ of the water side is equal to

$$\dot{Q}_w=\rho_w \dot{V}_w c_w \Delta T_w \quad (1)$$

and similarly the heat transfer rate of air side is equal to

$$\dot{Q}_L=\rho_L \dot{V}_L c_L \Delta T_L \quad (2)$$

The heat transfer rate of the MHE was calculated as the average value of water and air heat transfer rates

$$\dot{Q}=\frac{\dot{Q}_w+\dot{Q}_L}{2} \quad (3)$$

From the eq. (1) and (2) the error of the heat rate for the water and air side is [12]

$$\delta\dot{Q}_w=\frac{\partial\dot{Q}_w}{\delta\rho_w}\delta\rho_w+\frac{\partial\dot{Q}_w}{\partial\dot{V}_w}\delta\dot{V}_w+\frac{\partial\dot{Q}_w}{\partial c_w}\delta c_w+\frac{\partial\dot{Q}_w}{\partial\Delta T_w}\delta\Delta T_w \quad (4)$$

$$\delta\dot{Q}_L=\frac{\partial\dot{Q}_L}{\delta\rho_L}\delta\rho_L+\frac{\partial\dot{Q}_L}{\partial\dot{V}_L}\delta\dot{V}_L+\frac{\partial\dot{Q}_L}{\partial c_L}\delta c_L+\frac{\partial\dot{Q}_L}{\partial\Delta T_L}\delta\Delta T_L \quad (5)$$

In the eq. 4 and 5 $\delta\rho$, $\delta\dot{V}$, δc, $\delta\Delta T$ represent deviations of the fluid density, volume flow rate, heat capacity and temperature difference between the inlet and outlet respectively, for water (indexed as *w*) and air (indexed as *L*). The error parameters were derived from the accuracy of the measuring equipment and fluid properties.

Table 1. Error parameters

Unit	*Dimension*	*Error*
$\delta\rho_w$	kgm^{-3}	± 5
$\delta\dot{V}_w$	lh^{-1}	± 1,5 (2,5%)
δc_w	Jkg^{-1}K^{-1}	± 3
$\delta\Delta T_w$	K	± 0,1 (1,1%)
$\delta\rho_L$	kgm^{-3}	± 0,01
$\delta\dot{V}_L$	m^{3}h^{-1}	± 15 (5%)
δc_L	Jkg^{-1}K^{-1}	± 1
$\delta\Delta T_L$	K	± 0,10

The difference between the control and measuring thermocouple on the perforated plate was up to 0.6 ºC, which is generally caused by the paste thermal resistance, while in the case of the air temperature measurements, the difference was 0.1 ºC.

According to the data presented in Tab. 1, the estimated error of the heat flux rate of the water side is up to 5.6% and for the air heat flux it is 4.1%, which is generally acceptable.

Heat exchanger performance determination

The Number of Transfer Units - *NTU* method is used to calculate the rate of the heat transfer in heat exchangers. *NTU* is commonly defined as

$$NTU=\frac{UA}{C_{min}} \quad (6)$$

i.e. it could be calculated as

$$NTU=\frac{Q}{C_{min}\Delta\Theta} \quad (7)$$

The unit C_{min} is smaller from the product of the mass flow rate and heat capacity of the water and air - $\dot{m}_w c_w$ and $\dot{m}_L c_L$, while the unit $\Delta\Theta$ represents a log-mean temperature difference of water and air.

The thermal effectiveness of the heat exchanger is commonly defined as

$$\varepsilon=\frac{q}{q_{max}} \quad (8)$$

where quantity q_{max} is the maximum heat that could be transferred between the fluids.

$$q_{max}=C_{min}(T_{w,in} - T_{L,in}) \quad (9)$$

For any heat exchanger the effectiveness is in the function of *NTU* and C_r (Figure 5)

$$\varepsilon=f(NTU,C_r) \quad (10)$$

or

$$NTU=f(\varepsilon,C_r) \quad (11)$$

and C_r represents the heat capacity rate of C_{min} and C_{max} (higher from the product of mass flow rate and heat capacity - $\dot{m}_w c_w$ and $\dot{m}_L c_L$,)

$$C_r=\frac{C_{min}}{C_{max}} \quad (12)$$

According to Venkatarathnam, the equation for a conventional concentric tube counter flow heat exchanger could be also applied to *MHE* [14]

$$\varepsilon=\frac{1-\exp(-NTU(1-C))}{1-C_r\exp(-NTU(1-C))} \quad (13)$$

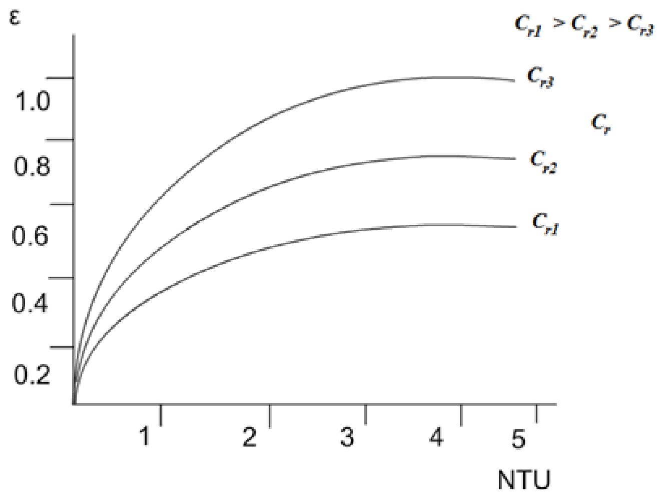

Figure 5. Heat exchanger effectiveness in the function of Cr and NTU

In order to calculate *NTU*, first the value *UA* was determined. *UA* represents the product of the heat transfer coefficient *U* and the heat transfer area *A* as

$$UA=\frac{\dot{Q}}{\Delta\Theta} \quad (14)$$

The results are presented in Figure 6 as the product *UA* in the function of the air flow. During the measurements, it was noted that the volume flow rate of the heating fluid - water does not have a significant influence on the heat transfer in the range of varied parameters (0.03-0.06) m^3/h.

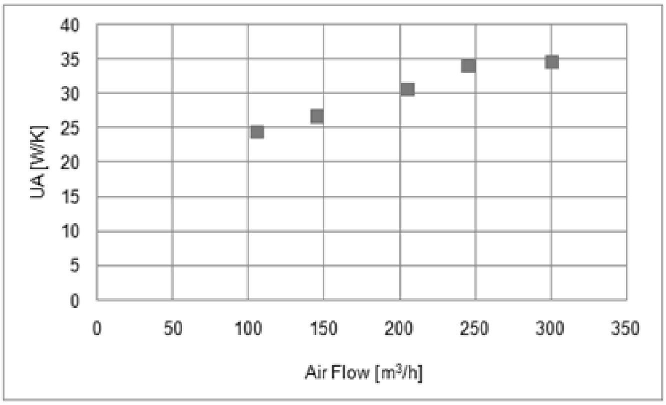

Figure 6. The product of U and A in the function of the air flow

After the *UA* was obtained, the *NTU* could be calculated according to eq. 6. The results in Figure 7 represent the *NTU*-Effectiveness relationship obtained by laboratory measurement. The results were compared to the equation for the concentric tube counter flow heat exchanger and the maximal difference was found to be less than 3% [13].

Further, in Figure 8 the obtained *NTU*-Effectiveness-*Cr* relation was presented. It could be noted that the results generally corresponded to the theory presented in Figure 5.

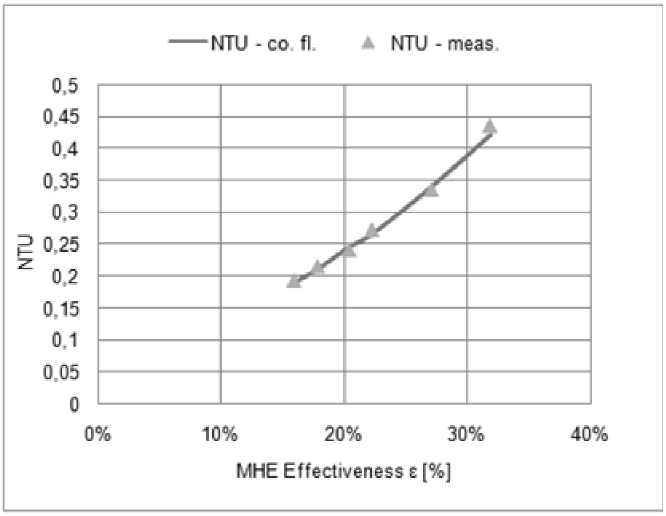

Figure 7. The comparison of the measured and calculated NTU- Effectiveness relation

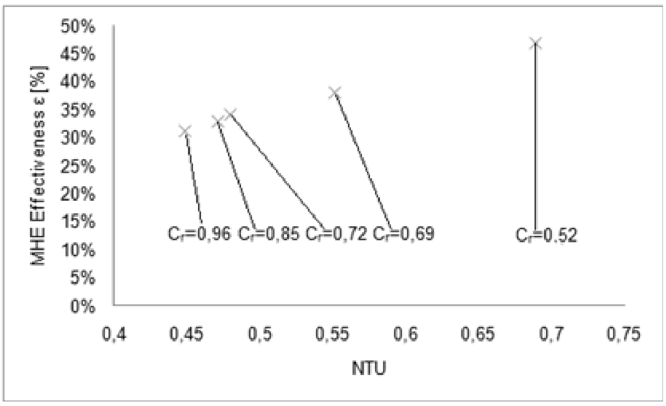

Figure 8. MHE efficiency in the function of NTU and Cr

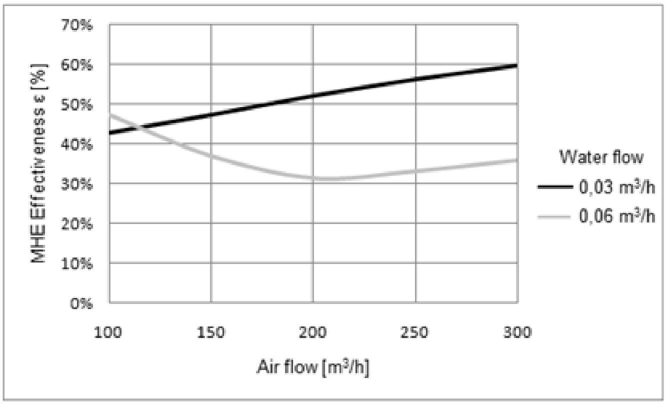

Figure 9. MHE effectiveness in the function of the air and water flow

On the basis of the results which are presented in Figures 6, 7, 8 and eq. 10, 15 and 18, the performance of the observed *MHE* could be predicted. The analysis of effectiveness and NTU are presented in Figures 9 and 10.

From the results in Figures 9 and 10 it could be concluded that the higher effectiveness in the range of varied parameters for the observed MHE was achieved in the case of the lower water flow - 0.03 m^3/h (Figure 9). Also, the maximal effectiveness of the observed MHE is 60% for the air flow of 300 m^3/h.

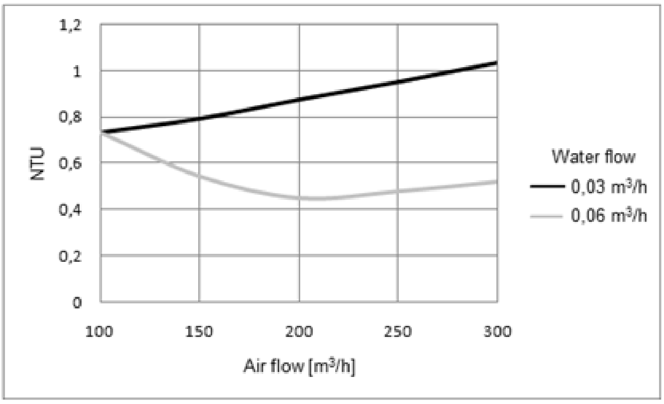

Figure 10. NTU in the function of the air and water flow

Similarly, the highest value of NTU of 1.1 could be obtained for the water flow of 0.03 m^3/h and the air flow of 300 m^3/h.

Conclusions

In this paper the results of measurements and the analysis of the compact perforated plate heat exchanger performance are presented. For the analysis of the results, NTU- Effectiveness relationship was used. The results for the NTU from the experiments were compared with the equation for the concentric tube counter flow heat exchangers and the experimental data agree reasonably well with the predicted. On the basis of the results and NTU-Effectiveness relation, the analysis of the performance was done. The results have shown that in the range of varied parameters, the highest effectiveness and NTU values could be obtained for the water flow of 0.03 m^3/h and the air flow of 300 m^3/h.

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Izvod

ANALIZA EFIKASNOSTI RAZMENJIVAČA TOPLOTE SA PERFORIRANIM PLOČAMA

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Cilj ovog rada je da se utvrdi optimalni radni režim posmatranog razmenjivača toplote vazduh/voda u opsegu variranih parametara. Eksperimentalno istraživanje je sprovedeno nad paketom od tri perforirane ploče, grejanih toplom vodom i smeštenih u kanalu eksperimentalne komore. Ventilator sa podesivim protokom vazduha je povezan sa eksperimentalnom komorom, pri čemu je protok vazduha variran u opsegu od 100 do 300 m^3/h, dok je protok vode variran u opsegu od 0.03 to 0.06 m^3/h. Termoparovi su bili postavljeni po površini srednje ploče u paketu duž njene uzvodne i nizvodne strane, kao i na ulazu i izlazu iz komore i između perforiranih ploča. Tokom svakog eksperimenta beležena su njihova očitavanja, kao i protoci i temperature vazduha i vode na ulazu i izlazu iz komore. Kako bi se odredile perforamanse razmenjivača sa perforiranim pločama izvršena je analiza NTU i efikasnosti na osnovu eksperimentalnih rezultata i analitičkih izraza. Eksperimentalni rezultati su pokazali kako se efikasnost razmenjivača toplote sa perforiranim pločama može računati kao kod razmenjivača toplote tipa cev-u-cevi. Na kraju rada je određena optimalna radna tačka u opsegu variranih parametara.

Ključne riječi: efikasnost, razmenjivači toplote, NTU, perforirane ploče