EXPERIMENTAL AND ANALYTICAL RESEARCH OF THE HEAT TRANSFER PROCESS IN THE PACKAGE OF PERFORATED PLATES

by

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Original scientific paper
DOI: 10.2298/TSCI16S5251Ž

The need for compact heat exchangers has led to the development of many types of surfaces that enhance the rate of heat transfer, among them the perforated plate heat exchangers, also known as matrix heat exchangers. The perforated plate heat exchangers consist of a series of perforated plates that are separated by a series of spacers. The present study investigates the heat transfer characteristics of the package of perforated plates. Perforated plates were 2 mm thick, with holes with 2 mm in diameter and porosity of 25.6%. The package of one, two, and three perforated plates was set in the channel of the experimental chamber at which entrance was a thrust fan with the ability to control the flow rate. The fluid flow rates, the temperatures of the fluids at the inlet and outlet of the chamber and the temperature of the air between the plates, were measured at the predefined locations in the package and the experimental chamber. Based on the measurements, heat transfer coefficients for the individual plates, as well as for the packages of perforated plates were determined. In further research, an iterative analytical procedure for investigation of the heat transfer process and the overall heat transfer coefficient for the package of perforated plates were developed. Based on these analytical and experimental results, conclusions were drawn about the heat transfer in a package of perforated plates.

Key words: *heat transfer, package, perforated plate, matrix heat exchanger*

Introduction

One of the most important properties of heat exchangers, a part of having a 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. A small mass means also a smaller heat inertia. This requirement is particularly important for small refrigerators operating at liquid helium temperature. The need of attaining high effectiveness and a high level of compactness together in one unit led to the invention of matrix heat exchangers (MHE) by McMation *et al.* [1]. Matrix heat exchanger consists of a package of perforated plates with a multitude of flow passages aligned in the direction of flow, allowing high heat transfer in a proper design unit. This exchanger can have up to 6000 m^2/m^3 surfaces to volume ratio [2-4].

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In 1966 an extensive experimental study of convective heat transfer and flow friction based on transient technique was published for eight different perforated surfaces [5]. In the report, the authors concluded that the perforations of perforated plate heat exchangers apparently disturb the thermal boundary layer to a much higher degree than the hydrodynamic boundary layer. According to the authors, by using perforated materials, an improvement in heat transfer was made. A proper literature review could be found in the papers of Venkatarathnam and Sarangi [3] and Ragab [6].

The goal of this paper is to investigate thermal processes on the air side of an air/water perforated plate heat exchanger on the basis of the steady-state method. The convective heat transfer characteristics of any heat exchanger surface can be determined using the steady-state, periodic test and transient test techniques [2]. For a steady-state method, the temperatures of hot and cold fluids entering and leaving the heat exchanger, as well as flow rates are measured and when steady-state is achieved it is possible to determine heat flux, thus the overall heat transfer coefficient. In the transient technique method, after the steady-state is achieved, the temperature of the fluid entering the heat exchanger is suddenly changed. The heat transfer coefficient can be determined from temperature-time history data. The periodic test techniques represent a variation of the transient method in which the temperature of the fluid entering the heat exchanger is continuously varied. Due to its simplicity, the steady-state method has been used in the experimental research.

The research was conducted in two directions: experimental research of the package of perforated plates with one, two and three plates and an analytical research of the package of perforated plates.

Experimental set-up

In this experiment, an MHE consisting of perforated plates with a porosity of 25.6%, 2 mm thick, with square arranged 2 mm in diameter perforations was tested. Each plate was divided into two sections: central section through which water flows, and peripheral section, through which the air flows. Sections were separated by a gasket, fig. (1). The plates were placed in the channel of the experimental chamber, at which entrance was a thrust fan with the airflow from 100 to 300 m^3/h. The distance between the plates in the package was set to 5 mm in order to provide the access for the measuring equipment, fig. (2).

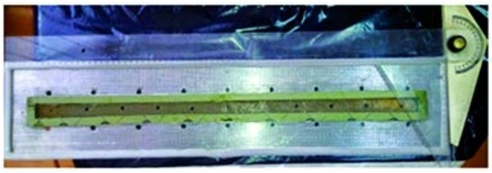

Figure 1. A perforated plate with a gasket

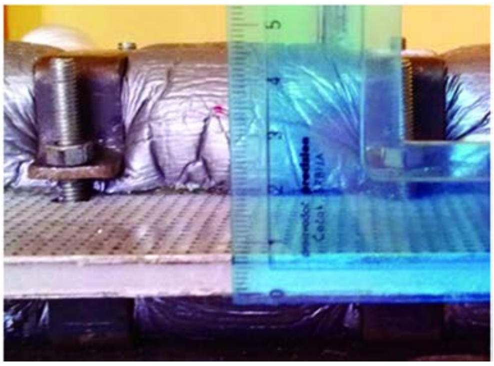

Figure 2. A package of two plates

As a heating fluid, water was used, while the heat source was the boiler with adjustable power. Hot water enters the collector and flows through the central part of the plate while the heat is transferred from the water to the plate.

Exchanged heat is further transferred by conduction through the plate towards the edge of the plate, where it comes into the contact with the cooler air stream. The heat is then transferred by means of convection from the plate to the cooler air stream.

For the needs of the experiment, the thermocouples were set on the perforated plate. In total eleven thermocouples were placed, five on each side of the plate, fig. (3), and one as control thermocouple for the error estimation. Heads of thermocouples were covered with thermally conductive paste in order to ensure thermal contact between thermocouples and plate. The thermocouples were calibrated before the experiment. Also, the temperatures of air at the inlet and the outlet of the chamber were measured by thermocouples.

The cold end of thermocouples was obtained as a mixture of water and ice. During each experiment, the air flow, water flow, temperatures at the inlet and outlet of the chamber, plate temperatures, and temperatures of air between the plates were measured. Measurements were conducted when the thermal equilibrium was achieved, *i. e.* when the change of the water temperature was less than 0.1 K during 10 min period [7].

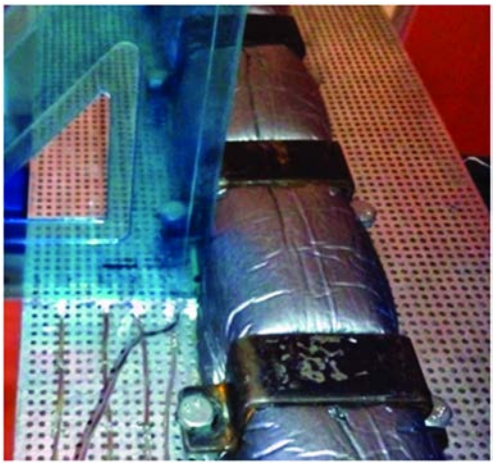

Figure 3. Thermocouples positions on the perforated plate

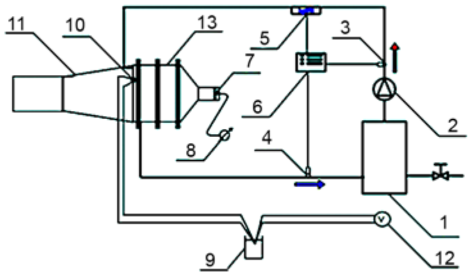

Figure 4. Experimental set-up;
1 – boiler, 2 – pump, 3, 4 – PT probes, 5 – ultrasonic water flow meter, 6 – acquisition unit, 7 – fan unit, 8 – fan speed control, 9 – cold end of thermocouples, 10 – thermocouples, 11 – Alnor balometer, 12 – millivoltmeter, 13 - chamber

The scheme of the experimental set-up is presented in the fig. 4. During the experimental research the following assumptions were made.

– The first plate in the package of two plates acts as the first plate in the package of two or more plates.
– The second plate in the package of two plates acts as the last plate in the package of two or more plates.
– The second plate in the package of three plates acts as the inner plates in the package of three or more plates.

According to these assumptions the packages of one, two, and three plates were formed. The convective heat transfer rate $\dot{Q}_\mathrm{w}$ from the water side is equal to:

$$\dot{Q}_\mathrm{w} = \rho_\mathrm{w} \dot{V}_\mathrm{w} c_\mathrm{w} \Delta T_\mathrm{w} \tag{1}$$

Similarly, the heat transfer rate of air side is equal to:

$$\dot{Q}_L = \rho_L \dot{V}_L c_p \Delta T_L \tag{2}$$

The heat transfer rate for the perforated plate was calculated as the average value of water and air side:

$$\dot{Q}_\mathrm{av} = \frac{\dot{Q}_L + \dot{Q}_\mathrm{w}}{2} \tag{3}$$

and the measurement error is estimated:

$$\varepsilon = \sqrt{\frac{(\dot{Q}_{\text{av}} - \dot{Q}_L)^2 + (\dot{Q}_{\text{av}} - \dot{Q}_{\text{w}})^2}{\dot{Q}_{\text{av}}^2}} \tag{4}$$

For the analysis, only measurements with the error less than 10% were used [8-10]. The overall heat transfer coefficient, α, is defined:

$$\alpha = \frac{\dot{Q}_{\text{av}}}{F\Delta\theta} \tag{5}$$

where $\Delta\theta$ is the difference between the average air temperature and the average temperature of perforated plates, and F – the overall heat exchanger surface on the air side. The partial heat transfer coefficients were determined on the basis of the air temperature difference between the upwind and downwind side of the observed plate and the average plate temperature:

$$\alpha_i = \frac{\dot{m}c_p(T_{\text{dow}} - T_{\text{upw}})}{F_i\Delta\theta_i} \tag{6}$$

where α_i represents the heat transfer coefficient for the i^{th} plate, and $\Delta\theta_i$ – the difference between the average plate temperature and the average air temperature. The obtained results for the individual heat transfer coefficients have been presented in the fig. 5.

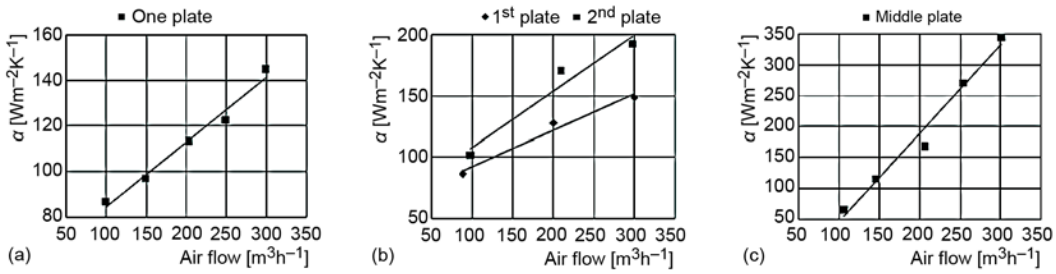

Figure 5. Heat transfer coefficients on the air side for the (a) package of one plate, (b) the first and the second plate in the package of two plates, and (c) middle plate in the package of three plates

Analytical research

For the additional analysis of the heat transfer process an analytical model was established under the following assumptions.

- The temperature of the perforated plate is a constant (equal to the average temperature).
- Heat transfer coefficients for individual plates are calculated according to experimental data presented in fig. 5.
- The air temperature field in front of the package, between plates in the package and behind the package is constant.

According to some researchers, the overall heat transfer coefficient for the package of n perforated plates could be calculated [3, 10, 11]:

$$\alpha = \frac{\sum_{i=1}^{n} \alpha_i F_i}{\sum_{i=1}^{n} F_i} \tag{7}$$

This way, it would be possible to calculate the heat transfer coefficients for the package of perforated plates in the function of air flow, a number of plates and their position in the package. For the analytical analysis the following iterative calculation was done:

(1) the air temperature at the inlet (upwind of the first plate, and $i = 1$) was set,
(2) in the first iteration, the air temperature downwind the first plate is assumed to be equal to the temperature at the inlet, and
(3) the plate heat flux value was calculated:

$$\dot{Q}_i^n = \alpha_i F_i \left(T_{\mathrm{pl},i} - \frac{T_{\mathrm{in},i} + T_i^n}{2} \right) \tag{8}$$

where heat transfer coefficient was set according to the results of measurement (fig. 5).

On the basis of the obtained value for the heat flux, the new corrected value of the air temperature downwind the observed plate is calculated:

$$\Delta T_i^n = \frac{\dot{Q}_i^n}{\dot{m} c_p} \tag{9}$$

$$T_i^{n+1} = T_i^n + \Delta T_i^n \tag{10}$$

The iteration procedure was carried out until the convergence criteria of 0.1% was achieved. In the further calculation, the obtained value of the air temperature downwind the first plate becomes the inlet value of the second plate and the new series of iterations are done.

Results and discussion

The comparison of measurements and analytical results for the air temperature in the package of three plates is presented in fig. 6. The maximal deviation of experimental and analytical results was under 6.5%. Further, in the fig. 7 are presented obtained heat transfer coefficients according to the algebraic relation presented with the eq. (6), the analytical model and experimental research. As it could be seen, there is a good agreement.

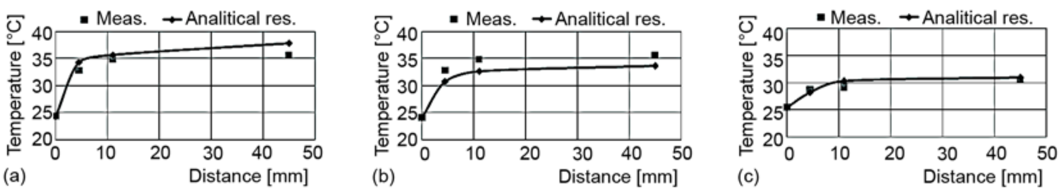

Figure 6. Air temperature profile for the airflow of (a) 100 m^3/h, (b) 200 m^3/h, and (c) 300 m^3/h

In fig. 8 are presented results for the package of three to eight plates and air flow of 200 m^3/h. The analysis has shown that with the increasing number of plates the overall heat transfer coefficient becomes a constant, tending to the value of heat transfer coefficient of the inner plate, figs. 5 and 8.

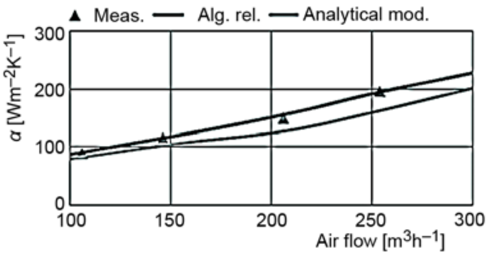

Figure 7. Heat transfer coefficient for the package of three perforated plates

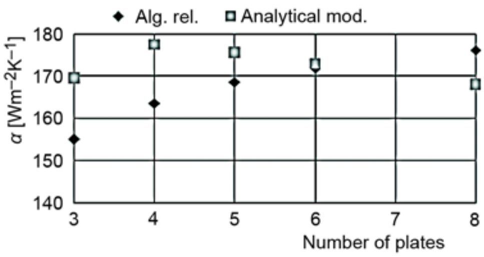

Figure 8. Heat transfer coefficient variation with increase of number of perforated plates

Conclusion

In this paper, an experimental and analytical research on the heat transfer process in the package of perforated plates was done. On the basis of the experimental research, the heat transfer coefficient was determined for individual perforated plate regarding the air flow rate and its position in the package. In the second part of the research, the overall heat transfer coefficient was investigated. In order to conduct the research, a simple analytical model was derived. The results have shown that after the five plates in the package, heat transfer coefficient stabilize itself around the value of heat transfer coefficient for the inner plate. This leads to the conclusion that with increased number of plates only the exchanger surface increases, but not the heat transfer coefficient.

Acknowledgment

The research presented in this paper was supported by the project III 42008 of the Ministry of Education, Science and Technological Development of the Republic of Serbia and College of Applied Technical Sciences in Nis. Also, the authors would like to express their gratitude to Mr. Milivoj Pejin and Mr. Branko Babić for their contribution to the research.

Nomenclature

c_p – heat capacity, [$\mathrm{Jkg^{-1}K^{-1}}$]
F – heat transfer area, [$\mathrm{m^2}$]
$\dot{m}$ – mass flow rate, [$\mathrm{kgs^{-1}}$]
$\dot{Q}$ – heat flux, [W]
T – temperature, [K]
$\dot{V}$ – volume flow rate, [$\mathrm{m^3s^{-2}}$]

Greek symbols

α – heat transfer coefficient, [$\mathrm{Wm^{-2}K^{-1}}$]
θ – temperature difference, [K]
ρ – a density of the fluid, [$\mathrm{kgm^{-3}}$]

Subscripts

av – average
dow – downwind
i – co-ordinate
in – inlet
L – air
pl – plate
upw – upwind
w – water

Superscripts

n – number of iterations

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Paper submitted: April 1, 2016
Paper revised: October 4, 2016
Paper accepted: October 16, 2016