

## EXPERIMENTAL AND NUMERICAL INVESTIGATION OF THERMAL AND FLUID-FLOW PROCESSES IN A MATRIX HEAT EXCHANGER

by

**Mladen A. TOMIĆ<sup>a\*</sup>, Predrag M. ŽIVKOVIĆ<sup>b</sup>, Biljana B. MILUTINOVIĆ<sup>c</sup>,  
Mića V. VUKIĆ<sup>b</sup>, and Aleksandar S. ANDJELKOVIĆ<sup>a</sup>**

<sup>a</sup> Faculty of Technical Sciences, University of Novi Sad, Novi Sad, Serbia

<sup>b</sup> Faculty of Mechanical Engineering, University of Nis, Nis, Serbia

<sup>c</sup> College of Applied Technical Sciences, Nis, Serbia

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*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 matrix heat exchangers. These heat exchangers consist of a series of perforated plates mutually separated and sealed by spacers. The goal of this research was to investigate the heat transfer process of matrix heat exchangers on the air side, at the close to ambient conditions. The research was conducted in two directions – experimental research and CFD research. The experimental investigation was carried out over a perforated plate package with the porosity of 25.6%. The air/water matrix heat exchanger was heated by hot water and was installed in an experimental chamber at which entrance was a fan with the variable flow rate and heated by hot water. The thermocouples were attached to the surface of the perforated plate at the upwind and downwind sides, as well as at the inlet and the outlet of the chamber. During each experiment, the thermocouple readings and the air and water-flow and temperatures were recorded. In the numerical part of the research, the matrix heat exchangers with different plate porosity from 10 to 50% were investigated. The results of the numerical simulations were validated against the experimental results. On the basis of the experimental and numerical results, equations for heat transfer as the function of Reynolds number and geometrical parameters was established.*

Key words: CFD, heat transfer, matrix heat exchangers, perforated plates

### Introduction

One of the most important properties of heat exchangers, a part of having a high efficiency 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 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<sup>2</sup>/m<sup>3</sup> surfaces to volume ratio [2, 3]. The whole pack together with a cast aluminum header at each

\* Corresponding author, e-mail: mladen.tomic@uns.ac.rs

end was held together by means of steel tie rods. The neoprene spacers almost eliminated axial conduction and provided gas tight seals even at liquid air temperatures. The construction was extremely simple and repairs were easy. Full-scale commercial units were intended to be fabricated for use in liquid oxygen production.

In 1966, an extensive experimental study of convective heat transfer and flow friction based on transient technique was published for eight different perforated surfaces [4]. 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.

Venkatarathnam *et al.* [2, 3] and Moheisen [5] gave a good literature review of MHE, their constructions, and Nusselt criteria. The MHE were used mainly as helium liquefies and low power cryorefrigerators based on Claude and reverse Brayton cycles [3], but also perforated plates could be used as fins for improving cooling of the electronic equipment [6].

The goal of this paper is to investigate thermal- and fluid-flow process at close to ambient conditions on the air side of an air/water, perforated plate heat exchanger, based on the steady-state method. The research was conducted in two directions: experimental research of the perforated plate package with one, two and three plates with constant porosity and an analytical research of the single plate with varied porosity.

### Literature review

The heat transfer improvement may be achieved by increasing the heat transfer coefficient, heat transfer surface areas, or both. In papers [7-10] the authors conclude that for certain values of perforation dimension, a perforated plate enhances heat transfer in comparison to the solid one. Perforated plate convective heat transfer takes place on three surfaces, fig. 1:

- the front area surface,
- the tubular surface of perforations, and
- the back surface of the plate.

The flow through the tubular section can be considered as a developing flow with a very high heat transfer coefficient. The heat transfer coefficient in the developing flow is well studied and it can be calculated in the function of the Pecklet number [11]:

$$\alpha = \zeta 0.0465 \text{Pe}^{0.75} \frac{\lambda}{d} \quad (1)$$

where  $\zeta$  is the function of pipe length,  $L$ , and the length of the pipe needed for flow to develop  $L'$ . The length  $L'$  is equal:

$$L' = 0.015 \text{Pe} d \quad (2)$$

Value  $\text{Pe}$  represents the Pecklet number, the product of the Reynolds and Prandtl number and  $d$  is the pipe diameter. The value of  $\zeta$  is presented in tab. 1.

Linghui *et al.* [12] studied the flow through hexagonally arranged perforations on the plate. The purpose of the research was to determine how the length to diameter ratio,  $\delta/d$ ,

**Table 1. The value of  $\zeta$  in the function of developing length**

$L/L'$	0	0.01	0.05	0.1	0.2	0.4	0.6	0.8	1.0	$\infty$
$\zeta$	$\infty$	1.26	1.16	1.12	1.08	1.05	1.03	1.01	1.00	1.00

of the plate's holes affected the heat transfer coefficient. They studied the ratios varying from 0.333 to 1.1666, holding the diameter constant while varying the thickness. Their experiments used the naphthalene sublimation technique to determine the heat transfer of the plate. The research led to the conclusion that there was little change in the heat transfer coefficients between the  $\delta/d$  ratios of 0.5 and 1.1. The final equation for the Nusselt number inside the tube was:

$$Nu = 2.058Re^{0.487} \quad (3)$$

The heat transfer from the front face of the plate was studied by Sparrow and Ortiz [13]. In their experiments the Reynolds number and the hole's pitch to diameter ratio was varied per hole. The suggested Nusselt criteria was in the function of the Reynolds and Prandtl number, but the characteristic length in the Nusselt criteria was the ratio of the module surface area of the pitch for  $2000 < Re < 20000$ :

$$Nu = 0.881Re^{0.476}Pr^{1/3} \quad (4)$$

and

$$Nu = \frac{\alpha A}{\lambda p} \quad (5)$$

The result was established for the limited case, where the relative spacing was  $2 < p/d < 2.5$ .

Dorignac *et al.* [14] conducted a series of experiments of air-flow leading to the result for the Reynolds number of 1000 to 1200:

$$Nu = 1.202 \left( \frac{p}{A^{0.5}} \right)^{1.879} \left( \frac{p}{d} \right)^{0.163} Re^{0.409} \quad (6)$$

where  $p$  is the pitch length and  $A$  is the area.

Heat transfer rate in the back face of the last plate was high due to flow separation and resulting turbulence [15]. Brunger *et al.* [16] studied the effectiveness for each of the three zones of heat transfer on a perforated plate: the front of the plate, the inside of the tube, and the back of the plate. In their study, they considered large pitch to diameter ratios ( $> 6.67$ ). For each of the heat transfer regions, an equation for effectiveness was given. The authors also stated that under typical operating conditions, about 62% of the ultimate temperature rise of the air was predicted to occur on the front surface, 28% in the hole, and 10% on the back of the plate.

An average heat transfer for the perforated plate may be defined:

$$\alpha = \frac{\sum_{i=1}^n \alpha_i A_i}{\sum_{i=1}^n A_i} \quad (7)$$

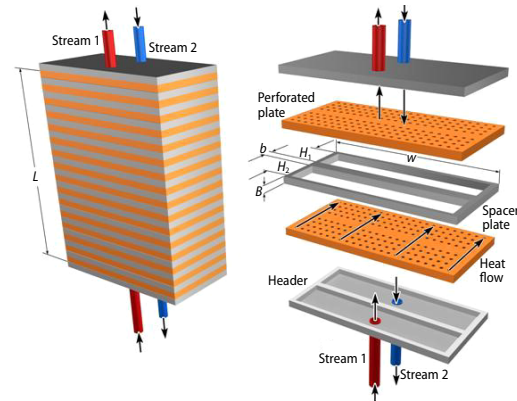


Figure 1. Matrix heat exchanger schema [5]

The Nusselt number correlation as a function of the Reynolds number, Prandtl number, and geometry factors apply to higher Reynolds numbers and lower plate porosities. The majority of authors derive empirical correlations for the Nusselt number and the friction factor vs. the Reynolds number. The general approach is to find a relation in the form:

$$Nu = CRe^n \quad (8)$$

where  $C$  and  $n$  are the functions of geometric parameters. An excellent review of these functions can be found in [2-5].

Hayes *et al.* [17] developed a model to determine the convective heat transfer coefficient of the upstream face, tube walls, and the leeward face of a perforated plate using CFD. The plate's holes were modelled as hexagonally shaped flow patterns, similar to Sparrow's and O'Brien's [15] model. The data obtained from the CFD model were found to agree within a few percent with Sparrow's data. This led to the conclusion that the CFD model and solution were valid. The final equation for the Nusselt number, for  $2000 \leq Re \leq 20000$  for the front side of a perforated plate was presented:

$$Nu = 1.057Re^{0.457}Pr^{0.333} \quad (9)$$

The model was applied to the tube surface of the perforations and the leeward side of the perforated plate to study their effect on the overall convective heat transfer coefficient of the matrix heat exchanger. This showed that the plate thickness had a substantial influence on the amount of heat transfer occurring within the tube part of a matrix heat exchanger, and that the leeward side of the perforated plate required more investigation. In the conclusion, an equation for the Nusselt number as a function of the Reynolds number was presented, taking into account the convection of the front, the back, and the inside of the perforation hole, considering air as a working fluid ( $Pr = 0.7$ ):

$$Nu = 0.397Re^{0.652} \quad (10)$$

Kutscher [18] did experiments to assess the overall convective heat transfer process for unglazed transpired solar collectors (UTC) subjected to uniform approaching flow. The UTC consist of dark porous cladding installed as the exterior layer of the building envelope (commonly roof or façade) with a narrow gap beneath it. The cladding absorbs solar radiation, thus heating up the air flowing through the perforations driven by a suction fan. The following empirical correlation is obtained for low-porosity plate for the average Nusselt number of the entire plate surface:

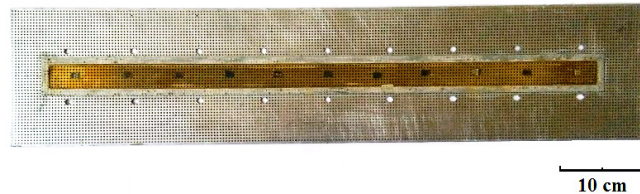
$$Nu = 2.75 \left( \frac{p}{d} \right)^{-1.2} Re^{0.43} \quad (11)$$

where the Reynolds number is based on the free stream velocity and the hole diameter. The Nusselt criterion in the equation is for low porosity plates. The plate thickness in the experiment is 0.794 mm and according to Kutscher [18], it does not have a significant influence.

### 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. 2. The plates were placed

in the channel of the experimental chamber, at which entrance was a thrust fan with the air-flow from 100 to 300 m<sup>3</sup>/h. The distance between the plates in the package was set to 5 mm in order to provide access for the measuring equipment.

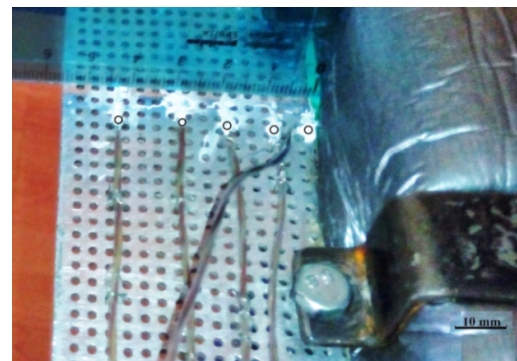


**Figure 2. A perforated plate with a gasket**

As a heating fluid, water was used, while the heat source was a boiler with adjustable power. Hot water enters the collector and flows through the central part of the plate. 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, a measuring plate was set-up. The measuring plate consists of thermocouples, which were set on the perforated plate. In total 11 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. Thermocouples were calibrated before the experiment. Also, the temperatures of air at the inlet and the outlet of the chamber were measured by thermocouples.

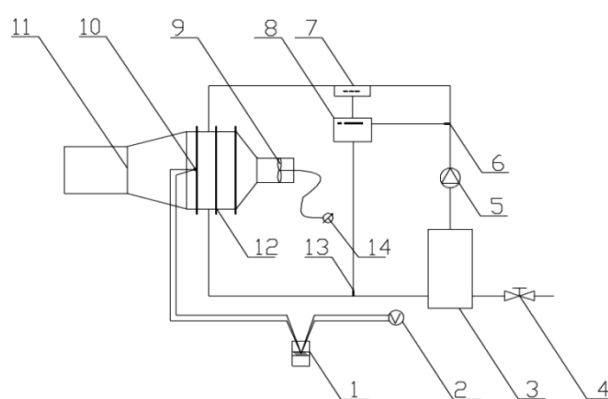


**Figure 3. Thermocouples positioning at a perforated plate, positions are indicated by circles**

The cold end of thermocouples was obtained as a mixture of water and ice. During each experiment, the air-flow rate, water-flow rate, temperatures at the inlet and outlet of the chamber, temperatures at the plate surface, and temperatures of air between the plates were measured. The scheme of the experimental set-up is presented in the fig. 4.

During the experimental research the following assumptions were made [19]:

- 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, and



**Figure 4. Experimental set-up:** 1 – cold end of thermocouples, 2 – millivoltmeter, 3 – boiler, 4 – valve for water supply, 5 – pump, 6, 13 – PT probe, 7 – ultra sonic water-flow meter, 8 – acquisition unit, 9 – fan unit, 10 – thermocouples, 11 – Alnore balometer, 12 – experimental chamber, 14 – fan speed control

- the second plate in the package of three plates acts as any inner plate in the package of three or more plates.

According to these assumptions the packages of one, two and three plates were formed. The 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 [20, 21].

After achieving the thermal equilibrium the following readings were done:

- water-flow rate,
- water temperature at the inlet,
- water temperature at the outlet,
- air-flow rate,
- air temperature at the chamber inlet,
- air temperature at the chamber outlet, and
- temperatures of the perforated plate.

According to the readings, the convective heat transfer rate,  $\dot{Q}_w$ , from the water side is equal:

$$\dot{Q}_w = \rho_w \dot{V}_w c_w \Delta T_w \quad (12)$$

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

$$\dot{Q}_L = \rho_L \dot{V}_L c_L \Delta T_L \quad (13)$$

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

$$\dot{Q}_{av} = \frac{\dot{Q}_L + \dot{Q}_w}{2} \quad (14)$$

and the measurement error is estimated:

$$\varepsilon = \frac{\sqrt{(\dot{Q}_L - \dot{Q}_{av})^2 + (\dot{Q}_w - \dot{Q}_{av})^2}}{\dot{Q}_{av}} \quad (15)$$

For the analysis, only measurements with the error less than 10% were used [21, 22]. The overall heat transfer coefficient  $\alpha$  is defined:

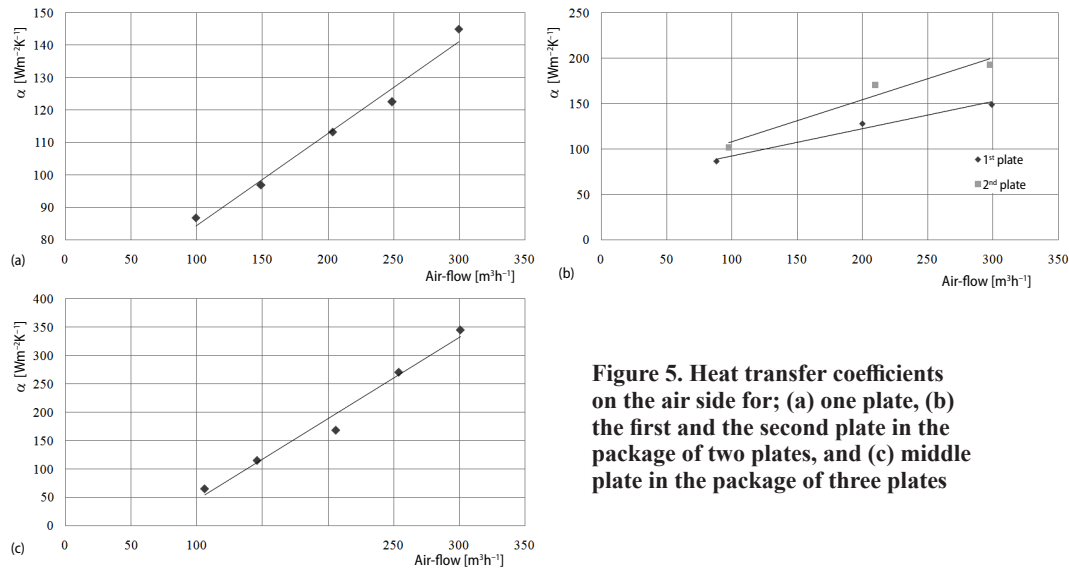
$$\alpha = \frac{\dot{Q}_{av}}{F \Delta \theta} \quad (16)$$

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

$$\alpha_i = \frac{\dot{m} c_L (T_{dow} - T_{upw})}{F \Delta \theta_i} \quad (17)$$

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





**Figure 5.** Heat transfer coefficients on the air side for; (a) 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

### Mathematical model

The 3-D steady-state turbulent flow is studied using the commercial software PHOENICS. The RANS equations together with an eddy viscosity turbulence model are solved. In order to choose the best suitable model, several of  $k$ - $\varepsilon$  variants were tested: standard  $k$ - $\varepsilon$  model, low-RE  $k$ - $\varepsilon$ , RNG  $k$ - $\varepsilon$ , and  $k$ - $\omega$  model. Although all of them gave similar results, the RNG  $k$ - $\varepsilon$  model presented itself as the fastest converging, and therefore was adopted for the simulation. The advantage of RNG  $k$ - $\varepsilon$  turbulence model stands in resolving flow separation and generally better performance than the standard  $k$ - $\varepsilon$  model and better treatment of flow for lower Reynolds numbers [23-25]. In general, the RNG  $k$ - $\varepsilon$  model could be written in the form of a transport equation:

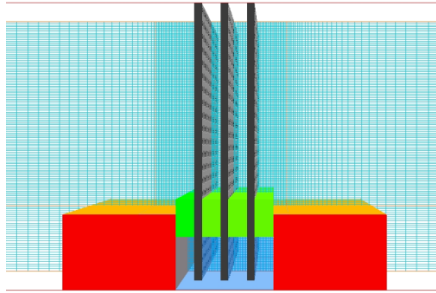
$$\frac{\partial(\rho\Phi)}{\partial t} + \frac{\partial(\rho\Phi u_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \Gamma_\Phi \left( \frac{\partial \Phi}{\partial x_j} \right) + S_\Phi \quad (18)$$

where the details have been presented in tab. 2.

**Table 2.** The RNG  $k$ - $\varepsilon$  turbulent model transport equations and constants

Transport equation	$\Phi$	$\Gamma_\Phi$	$S_\Phi$
Turbulent kinetic energy	$k$	$\nu_t/\sigma_k$	$\rho(G-\varepsilon)$
Turbulent kinetic energy dissipation	$\varepsilon$	$\nu_t/\sigma_\varepsilon$	$\rho(\varepsilon/k)(C_{\varepsilon 1}G-C_{\varepsilon 2}\varepsilon)$
$G=\nu_t(\partial_k U_i+\partial_i U_k)\partial_k U_i$		$\nu_t=C_\mu k^2/\varepsilon$	
$\sigma_k=0.7194, \sigma_\varepsilon=0.7194, C_{\varepsilon 1}=1.42, C_{\varepsilon 2}=1.68, C_\mu=0.0845$			

In order to model the local flow structure, a representative unit with  $16 \times 16$  holes is defined, which corresponds to the geometry of the experimental model, figs. 2 and 6. A uniform velocity is set at the inlet and a constant pressure boundary at the outlet. Turbulence quantities at the inlet are determined from the empirical correlations for turbulence intensity for internal pipe flows [24]. The domain is created as a sufficiently long ( $> 20d$ ) on the downstream side to ensure the simulation results.



16 × 16 × 3  $p = 0.256$  0.278 m/s 107 m<sup>3</sup>/h RNG

**Figure 6. Numerical model of a matrix heat exchanger – a side view**

parameter [23, 26]. The results showed that under the cell size of 0.8 mm, the temperature at the outlet was varying not more than 0.1°C, tab. 3 [23].

### Numerical modelling

The purpose of the numerical model was to determine the influence of the porosity, number of perforated plates and the distance between plates on the heat transfer process in a matrix heat exchanger. In order to generate the optimal grid the grid independence test was done. The grid size was varied in two directions along the plate length and width, regarding the plane Y-Z plane perpendicular to the flow direction, fig. 6. The grid was set to square in the Y-Z plane, and the length of the cell in these two directions was varied from 0.88 to 0.2 mm, while the fluid temperature at the outlet was chosen as a quality

**Table 3. Temperature variation at the outlet**

Cell size [mm]	0.88	0.8	0.72	0.6	0.53	0.35	0.26
Outlet temperature [°C]	21.65	22.3	22.28	22.33	22.35	22.22	22.4

According to this, the cell length was chosen to be 0.5 mm, *i. e.* minimum four cells per hole diameter. The grid in the flow direction was refined near solid walls so that  $y^+$  was less than 30 and the standard wall functions could be applied, fig. 6. The air velocity and the temperature at the inlet, as well as the constant heat flux boundary conditions on the plate surface, have been set according to the experimental research results presented in the fig. 5 and earlier research results [19, 23, 27]. In comparison to earlier research, single plate was substituted with a package of two, or three plates. A typical package is presented in the fig. 6.

The energy balance for the fluid side is represented:

$$\dot{Q} = \rho w A c_L (T_{\text{out}} - T_{\text{in}}) \quad (19)$$

where  $c_L$  represents mass specific heat capacity of air,  $A$  – the cross-section area, and  $w$  – the free stream fluid velocity. On the other hand the heat transmitted from the plate is equal:

$$\dot{Q} = \alpha F (T_{\text{pl}} - T_L) \quad (20)$$

where  $F$  represents the active heat transfer surface. Combining eqs. (19) and (20), heat transfer coefficient is then equal:

$$\alpha = \frac{\rho w A c_L (T_{\text{out}} - T_{\text{in}})}{F (T_{\text{pl}} - T_L)} \quad (21)$$

### Results and discussion

For the Nusselt criteria for a single plate an arbitrary function was chosen in the form:

$$\text{Nu} = C \left( \frac{p}{d} \right)^m \text{Re}^n \quad (22)$$



according to earlier research of the Kutcher [18]. The term  $p/d$  of eq. (22) on the right-hand side is inverse proportional to the square root of the porosity and therefore represents the influence of the porosity on the heat transfer. The Reynolds number is defined in the function of free-stream fluid velocity,  $w$ :

$$Re = \frac{w p}{\nu} \quad (23)$$

and the Nusselt number is equal:

$$Nu = \frac{\alpha p}{\lambda} \quad (24)$$

The characteristic length in Reynolds and Nusselt number,  $p$ , represents the size of the pitch between holes. The properties of air were set to correspond to the average temperature, in comparison to earlier research [23]. The best fitting results for the single plate were found at:

$$Nu = 0.803 \left( \frac{p}{d} \right)^{0.492} Re^{0.524} \quad (25)$$

for  $50 \leq Re \leq 500$  with regression of 0.97. Since the Prandtl number of air for testing conditions is equal to 0.71, the Nusselt criteria for gases could be written:

$$Nu = 0.900 \left( \frac{p}{d} \right)^{0.492} Re^{0.524} Pr^{0.333} \quad (26)$$

The comparison of results obtained by numerical research and measurements with the eq. (26) is presented on the fig. 7. Further, the influence of the number of plates on the Nusselt number is assumed to be in the form:

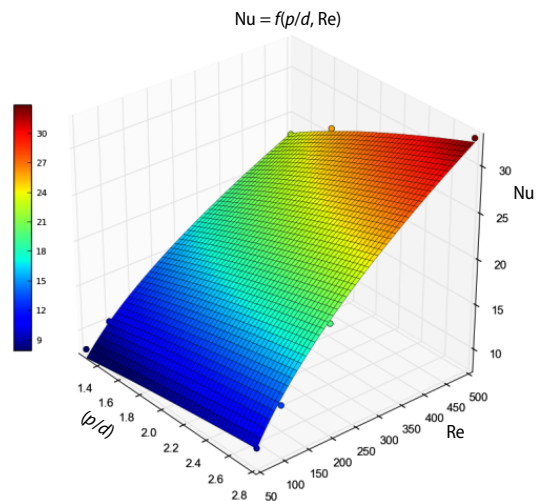
$$Nu_{\text{pack}} = N Nu \quad (27)$$

where  $N$  represents the influence of plate numbers and  $Nu_{\text{pack}}$  represents the Nusselt number of the package of perforated plates. The best fitting has been obtained from the function in the form:

$$N = 1.02n^{\left(0.56 + \frac{51.36}{Re}\right)} \quad (28)$$

and combined with eq. (26), the Nusselt criteria for the matrix heat exchanger gets the final form:

$$Nu = 0.918n^{\left(0.56 + \frac{51.36}{Re}\right)} \left( \frac{p}{d} \right)^{0.492} Re^{0.524} Pr^{0.333} \quad (29)$$



**Figure 7. Nusselt criteria for the single plate as the function of geometry parameters and Reynolds number compared with measuring and numerical results (for color image see journal web site)**

The research has shown that for the packages of five and more plates the heat transfer coefficient becomes a constant, and as it could be expected tends to the value of the heat transfer coefficient of the middle plate. Further, the research on the destination between plates influence has shown that there is no substantial difference in heat transfer process for the distance between plates between 1 mm and 5 mm, *i. e.* for the distance and diameter ratio of  $0.5 \leq s/d \leq 2.5$ , where  $s$  stands for the distance between the plates.

## Conclusion

The goal of this research was to investigate the heat transfer process of MHE on the air side, at the close to ambient conditions. The research was conducted in two directions – experimental research and CFD research. The results have shown that the convective heat transfer for a matrix heat exchanger depends strongly on geometry parameters – the plate porosity and partly on the number of plates, but also on the number of perforated plates in the matrix heat exchanger. Further, a Nusselt criteria equation was derived as the function of geometry parameters, Reynolds number, Prandtl number, and a number of plates. The obtained equations for a single plate and a package of plates represents the equations for the design of a MHE, which are applicable for a wide range of geometry parameters. In the end, the comparison of experimental and numerical results against obtained criteria equations was done and it has shown a good agreement.

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## Nomenclature

$A$	– area, [m <sup>2</sup> ]
$c$	– heat capacity, [Jkg <sup>-1</sup> K <sup>-1</sup> ]
$d$	– perforation diameter, [m]
$F$	– heat transfer area, [m <sup>2</sup> ]
$k$	– kinetic energy of turbulence, [m <sup>2</sup> s <sup>-2</sup> ]
$\dot{m}$	– mass-flow rate, [kg s <sup>-1</sup> ]
$N$	– function of number of plates, [–]
$n$	– number of plates, [–]
$p$	– pitch, [m]
$\dot{Q}$	– heat flux, [W]
$s$	– destination between plates, [mm]
$T$	– temperature, [K]
$\dot{V}$	– volume flow rate, [m <sup>3</sup> s <sup>-2</sup> ]
$w$	– fluid velocity, [ms <sup>-1</sup> ]

## Greek symbols

$\alpha$	– heat transfer coefficient, [Wm <sup>-2</sup> K <sup>-1</sup> ]
$\varepsilon$	– dissipation of kinetic energy of turbulence, [m <sup>2</sup> s <sup>-3</sup> ]
$\lambda$	– thermal conductivity, [Wm <sup>-1</sup> K <sup>-1</sup> ]
$\rho$	– fluid density, [kgm <sup>-3</sup> ]

## Subscripts

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

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