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A NUMERICAL STUDY OF PERFORATED PLATE LOCAL HEAT TRANSFER COEFFICIENT

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Abstract: 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 and perforated plate 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 local heat transfer characteristics of flow through a perforated plate with 2 mm in diameter, hole length to diameter ratio of 1 and porosity of 25.6%. For the determination of the local heat transfer, numerical simulations were performed. Reynolds numbers based on the perforated plate pitch were in the range from 80 to 300. The results of average Nusselt number prediction were compared with the related experimental correlations. The experimental results agreed on qualitatively with the results obtained using a CFD.
Keywords: local heat transfer, perforated plate, CFD

INTRODUCTION

One of the most important properties of heat exchangers, apart 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 surface to volume ratio [2,3].

The convective heat transfer characteristics of any heat exchanger surface can be determined using 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 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 represents a variation of the transient method in which the temperature of the fluid entering the heat exchanger is continuously varied.

In the present study, a local heat transfer over a single perforated through which air is flowing is investigated using a Computational Fluid Dynamics (CFD) method. The research was conducted in order to understand the thermal process at the surface of the perforated plate. For the numerical experiment, a block of 2x2 holes with diameter of 2 mm and 3.5 mm pitch between holes was modeled (Figure 1).

MATHEMATICAL MODEL

The mathematical model is based on following governing equations:

» continuity equation

$$\frac{\partial \rho}{\partial t}+\frac{\partial(\rho u_i)}{\partial x_i}=0 \tag{1}$$

» momentum (Navier-Stokes) equations

$$\frac{\partial(\rho u_i)}{\partial t}+\frac{\partial(\rho u_i u_j)}{\partial x_j}=\frac{\partial(\tau_{ij})}{\partial x_j}-\frac{\partial p}{\partial x_i}+f_i \qquad (2)$$

» energy equation

$$\frac{\partial(\rho h)}{\partial t}+\frac{\partial(\rho u_i h)}{\partial x_j}=\frac{\partial(j_{ih})}{\partial x_j}+\mu\phi+S_h \qquad (3)$$

where ρ is the density, u_i – the three main velocity components, p – the pressure, f_i are the body forces and any other additional momentum sources, h is the enthalpy, and S_h represents the generation/ destruction rate of enthalpy. The τ_{ij} is the momentum shear stress tensor and the j_{ih} – the diffusion flux of energy transport. In the energy equation, the diffusion flux of energy transport term j_{ih} includes the energy transfer due to conduction:

$$j_{ih}=\Gamma_T\frac{\partial T}{\partial x_i}. \qquad (4)$$

where the factors Γ_T are the diffusion coefficients for the enthalpy – thermal conductivity coefficient. The second term on the right hand side in eq. (3) represents the energy transport by diffusion of species and the Soret-effect species diffusion transport, respectively. Finally, the term Φ is the viscous dissipation defined as:

$$\phi=\frac{1}{2}\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)^2-\frac{2}{3}\frac{\partial u_k}{\partial x_k}\frac{\partial u_l}{\partial x_l}. \qquad (5)$$

Numerical research

Three-dimensional steady-state turbulent flow is studied using commercial software ANSYS Fluent 14. The Reynolds Averaged Navier–Stokes equations (RANS) together with an eddy viscosity turbulence model are solved. The Shear Stress Transport (SST) k-ω turbulence model is chosen for its advantage in resolving flow separation and generally better performance than the standard k-ε model [4,5]. In order to model the local flow structure, a representative unit with 4x4 holes is defined (Figure 1). 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 [6]. Symmetry planes were set on the side boundaries perpendicular to the flow direction (Figure 1). Domain is created as a sufficiently long (>20d), especially on the downstream side, to ensure the simulation results.

The computational mesh (Figure 1) uses continuously refined resolution near the solid wall boundaries, so that y^+ is less than 5 in order standard wall functions could be applied. The air velocity and the temperature on the inlet, as well as the constant temperature boundary conditions on the wall surface have been set according to the earlier experimental research conducted by Tomić et al. [7]. A typical convergence of the numerical research have been presented on the Figure 2.

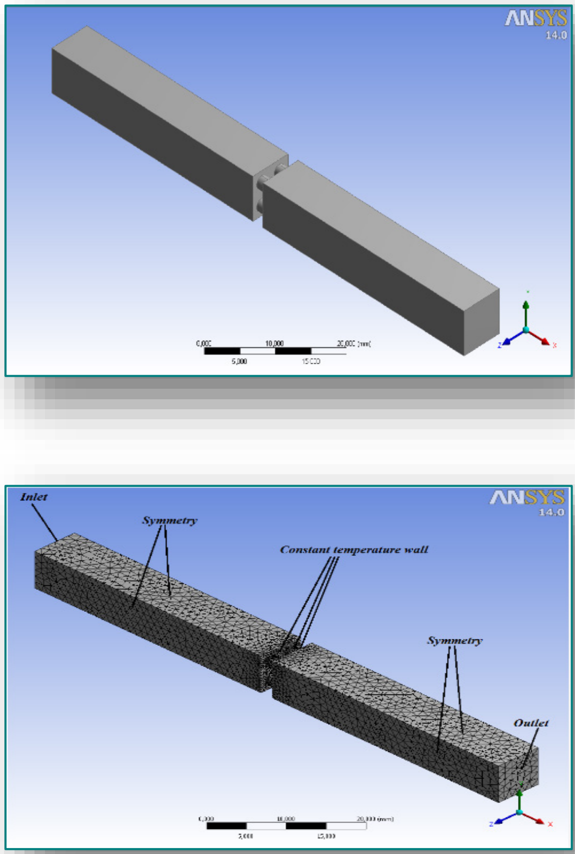

Figure 1. A 3D model and its numerical grid with boundary conditions

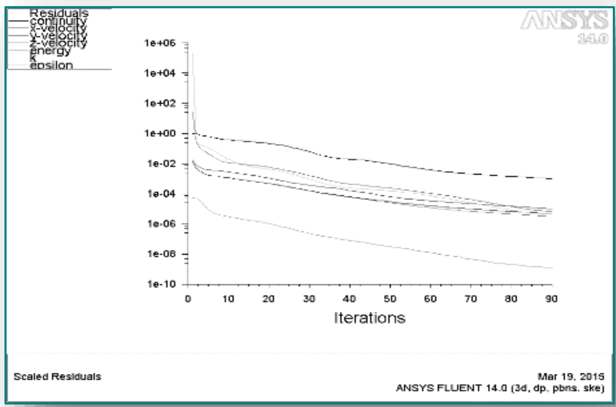

Figure 2. A typical convergence of the numerical model

2.2 Validation of the numerical model

The validation of the numerical experiment has been done by comparing with them with results of Tomić et al. [7]. On the Figs. 3-5 are presented comparison for the partial heat transfer coefficients, as well as overall heat transfer coefficient.

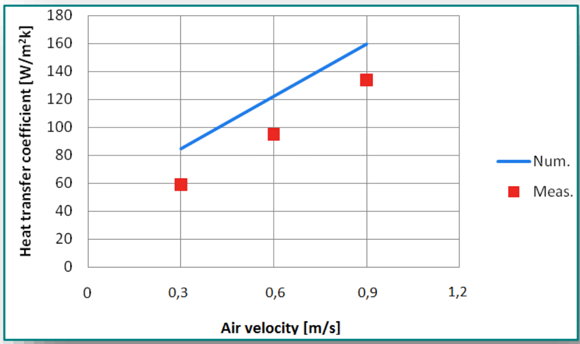

Figure 3. Comparison of the numerical model and experimental results for the heat transfer coefficient for the upwind side of the perforated plate

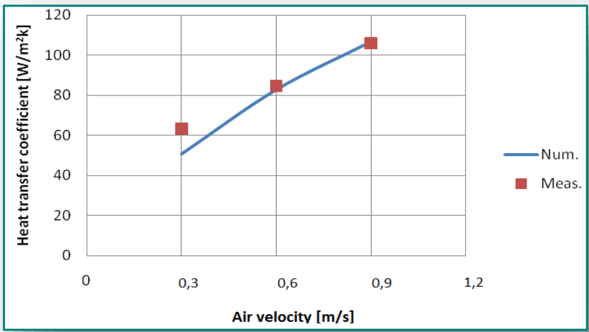

Figure 4. Comparison of the numerical model and experimental results for the heat transfer coefficient for the downwind side of the perforated plate

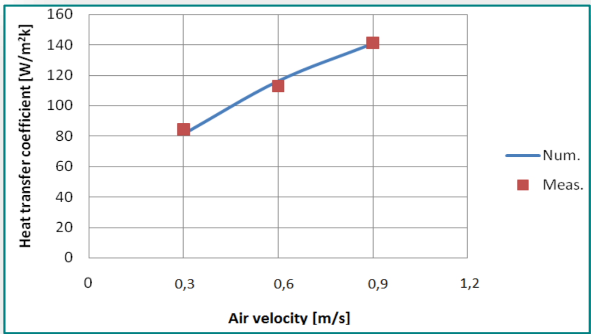

Figure 5. Comparison of the numerical model and experimental results for the overall heat transfer coefficient

RESULTS AND DISCUSION

Figure 6 and 7 shows the heat fluxes for the upwind and downwind side of the perforated plate along with velocity profile for the air velocity of 0,33 and 1,24 m/s, or in pitch based Reynolds numbers 80 to 300. As it cold be seen from the figures on the upwind side the difference between local heat transfer zones is the consequence of the flow separation and its acceleration through perforations, while on the downwind side, it is the consequence of the jet flow and surrounding recirculation zones. Each recirculation zone is located between diagonally neighbouring holes (Figure 6 and 7). Between the recirculation zones are „dead zones“ with low air velocity and thus low heat transfer.

Generally, the local differences in the heat transfer coefficients are upto 2 times and they are in the function of local air velocity. If the local heat transfer coefficient is assumed to be proportional to the local heat flux and according to the Nusselt criterial equation

$$\alpha_x \sim w_x^n . \quad (4)$$

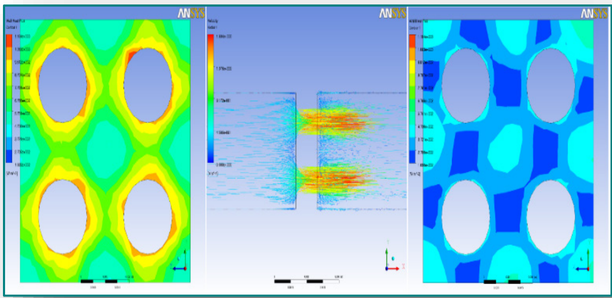

Figure 6 . Numerical results for the Reynolds number 80 - air velocity of 0,33 m/s

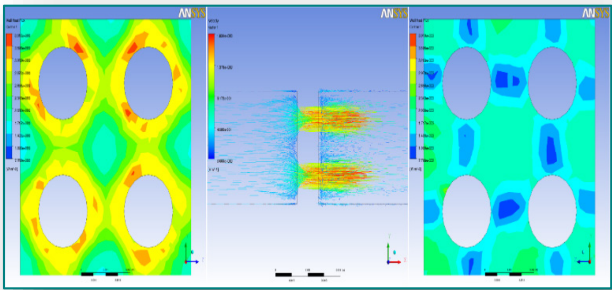

Figure 7. Numerical results for the Reynolds number 300 - air velocity of 1,24 m/s

At the upwind side, the heat transfer coefficient is proportional to

$$\alpha_x \sim w_x^{0,522} , \quad (5)$$

and on the downwind side

$$\alpha_x \sim w_x^{0,520} . \quad (6)$$

It could be assumed that power in the eqs. 5 and 6 has the mean value of 0.521.The results in the eqs. 5 and 6 are in good agreement with results of Tomić et al. for the overall heat transfer coefficient [7]

$$Nu = 1.055 Re^{0,524} . \quad (7)$$

CONCLUSION

The detailed gas flow and heat transfer through a single perforated plate is investigated using a Computational Fluid Dynamics method in order to determine local heat transfer coeficients. The obtained results show good general trend and mutual agreement. The current work lays a foundation for the future research of the influence of geometric parameters and number of plates in a perforated plate heat exchanger on the local heat transfer.

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Note

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