DETERMINATION OF HEAT TRANSFER COEFFICIENT IN CHANNELS USING CFD

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Abstract. *The studies about fluids flow in channels are very important, once the knowledge about the process of heat transfer make us to improve equipments already existing, making them more efficient. The determination of correlations to the heat transfer coefficient is generally done empirically. The purpose of this work was to verify the relationship between the Nusselt's numbers and determine the coefficient described before using computational fluid dynamics (CFD). In the results is evident that the Nusselt's number is strongly influenced by Reynolds' number, and the correlation found for the simulated results were satisfactory..*

Keywords: *Fluids flow, CFD, channels, convective coefficient*

1. INTRODUCTION

The fluids flow in channels is present in various industrial processes, such as: pasteurization, refrigeration of foods, etc. According to Fernandes et al. (2002), the diversity of materials, possible arranges and different shapes of the plates make the plate heat exchanger very versatile. Meanwhile, its design is a quite complex because of the variety of possible configurations in the plates shape and numbers of arranges that can be obtained. Conventional literature about heat transfer correlates the dimensionless Nusselt (Nu), Reynolds (Re) and Prandtl (Pr) (FERNANDES, 2002). However, these correlations are determined empirically. Martins & Oliveira (2007), in order to get correlations for heat transfer coefficient in apple refrigeration, used the computational fluid dynamics (CFD). Shah & Fock (1988) in Gut & Pinto (2001), Cooper & Usher (1992) and Incropera (2002) present correlations to calculate the heat transfer coefficient in the channels flow.

Advances of numerical methods with the informatics development are solving some fluids flow and mass and heat transfer problems, integrating the CFD tools to projects procedures and flow analysis in the various areas of the engineering (MALISKA, 2004).

The objective of the present study was to verify the behavior of the heat transfer coefficient in relation to the Reynolds' number variation and also to determine a correlation to this coefficient applied to channels.

2. MATHEMATICAL FORMULATION

The channel shape considered in this study is described in the Figure 1.

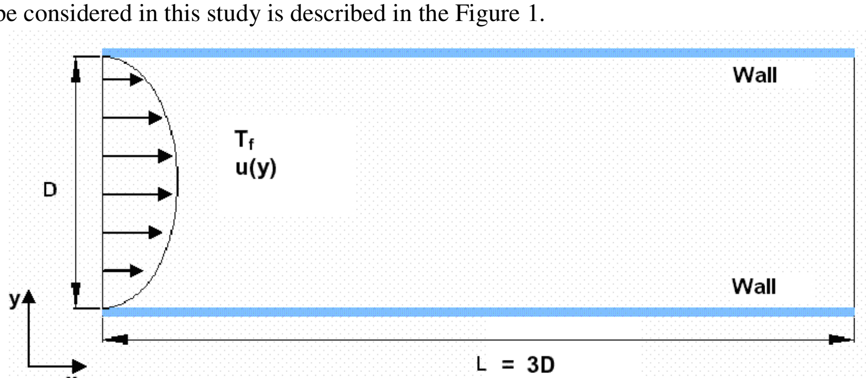


Figure 1. Schematic diagram of the channel.

It is considered to this problem that the fluid cannot be compressed, and floating effects are insignificant. The channel dimensions were defined in relation of the wall spacing D, being L=3D.

The generalized governing equations (continuity Eq. (1), momentum Eq. (2) and energy Eq. (3)) for this problem are show below in the dimensionless form:

$$\frac{\partial u^*}{\partial x^*}+\frac{\partial v^*}{\partial y^*}=0 \tag{1}$$

$$\frac{\partial u^*}{\partial \tau}+u^*\frac{\partial u^*}{\partial x^*}+v^*\frac{\partial u^*}{\partial y^*}= -\frac{\partial p^*}{\partial x^*}+\frac{1}{\mathrm{Re}}\left(\frac{\partial^2 u^*}{\partial x^{*2}}+\frac{\partial^2 u^*}{\partial y^{*2}}\right) \tag{2}$$

$$\frac{\partial v^*}{\partial \tau}+u^*\frac{\partial v^*}{\partial x^*}+v^*\frac{\partial v^*}{\partial y^*}= -\frac{\partial p^*}{\partial y^*}+\frac{1}{\mathrm{Re}}\left(\frac{\partial^2 v^*}{\partial \mathrm{x}^{*2}}+\frac{\partial^2 v^*}{\partial y^{*2}}\right) \tag{3}$$

$$\frac{\partial \theta_f}{\partial \tau}+u^*\frac{\partial \theta_f}{\partial x^*}+v^*\frac{\partial \theta_f}{\partial y^*}= \frac{1}{\mathrm{Re}\,\mathrm{Pr}}\left(\frac{\partial^2 \theta_f}{\partial x^{*2}}+\frac{\partial^2 \theta_f}{\partial y^{*2}}\right) \tag{4}$$

where x, y and τ are the dimensionless position and time, respectively, u and v are the dimensionless velocity components, p is the dimensionless pressure, θ_f is the dimensionless temperature for the fluid domain, Re is the Reynolds number based on the channel height (D) and Pr is the Prandt number.

The dimensionless variables used were:

$$x^*=\frac{x}{L};\quad y^*=\frac{y}{L};\quad u^*=\frac{u}{u_m};\quad v^*=\frac{v}{u_m};\quad \tau=\frac{u_m}{L}t$$
$$p^*=\frac{p}{\rho {u_m}^2};\quad \theta=\left(\frac{T-T_f}{T_i-T_f}\right);\quad \mathrm{Pr}=\frac{\nu}{\alpha};\quad \mathrm{Re}=\frac{\rho VL}{\mu} \tag{5}$$

were μ, α, ρ, ν, L, u_m are respectively the viscosity (kg m s^{-1}), thermal difusivity (m^2 s^{-1}), density (kg m^{-3}), kinematics viscosity (m^2 s^{-1}), characteristic length (m) and maximum velocity (m s^{-1})

2.1. Numerical solution

Numerical solution consists on two major steps: discretization on the solution domains and equation discretization. The first step will be carried out by means of a multi-purpose structured quadrilateral mesh generator in two-dimensions. This mesh generator was implemented using the Delaunay triangulation technique (Martins et al., 2004). An unstructured triangular mesh was generated for the calculation domain, as shown in Figure 2.

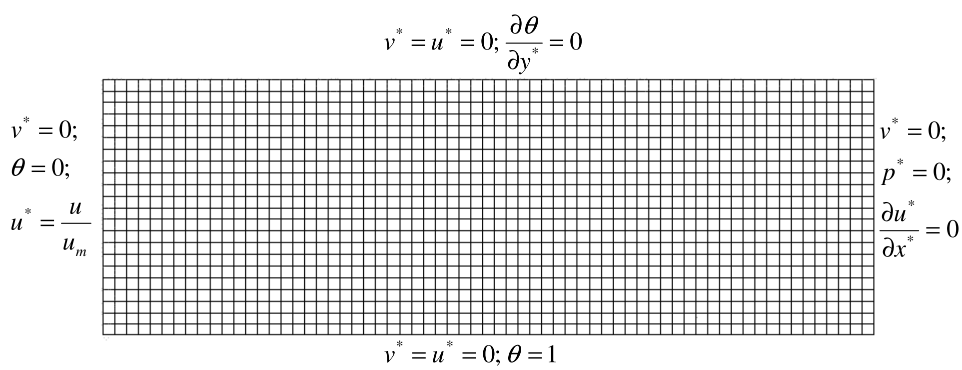


Figure 2. Quadrangular structured grid and contour conditions.

Since stepper temperature and velocity gradient are expected at the apple surfaces, a mesh refinement is employed in order to reduce errors associated to the numerical discretization of the conservation equations.

The conservation equations (Eqs. 1 and 2) can be rewritten in a general form as

$$\frac{\partial \phi}{\partial \tau}+\nabla\cdot(\mathbf{u}\phi)-\nabla\cdot(\Gamma\nabla\phi)=S_\phi(\phi) \tag{6}$$

where ϕ is a general scalar (u, v, θ_f, θ_s), u is the dimensionless velocity vector, Γ is a diffusivity and S_ϕ (ϕ) is a source term. The finite volume method uses an integral form over a control volume or cell (triangle) with the volume V and with a point P inside:

$$\int_V \frac{\partial \phi}{\partial t} dV + \oint_{\partial V} (\mathbf{u}\phi) \cdot d\mathbf{s} - \oint_{\partial V} (\Gamma \nabla \phi) \cdot d\mathbf{s} = \int_V S_\phi(\phi) dV \tag{7}$$

The second and the third terms in equation 5 represent the convective and the diffusive flux, respectively, across the cell boundary ∂V with normal surface vector s. The mass center defines the point P inside a triangular cell and each non-boundary cell has three neighbors N.

The time discretization scheme is implicit with first order accuracy. For the spatial discretization the central second-order differencing scheme is used for the diffusive flux, using a linear variation of ϕaround P with iterative non-orthogonal correction (Martins and Oliveira, 2005). The weighted upwind differencing scheme (WUDS) was used to discretize the convective flux. The resulting algebraic system of equations is solved by the bi-conjugated gradient method (Peric and Ferziger, 2002). The energy and momentum equations are coupled using the SIMPLE algorithm (Versteeg and Malalasekera, 1995).

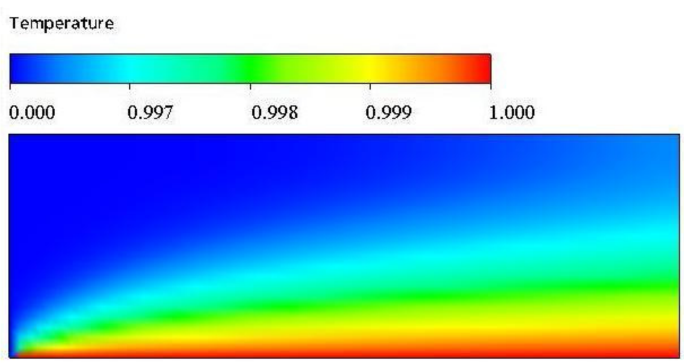
The local Nusselt number (Nu) and average Nusselt ($\overline{Nu}$), defined by equations Eq. (6) e Eq. (7) respectively were calculated based on the wall heat flux of the channel obtained by the simulation.

$$Nu = \left.\frac{d\theta}{dx^*}\right|_x \tag{6}$$

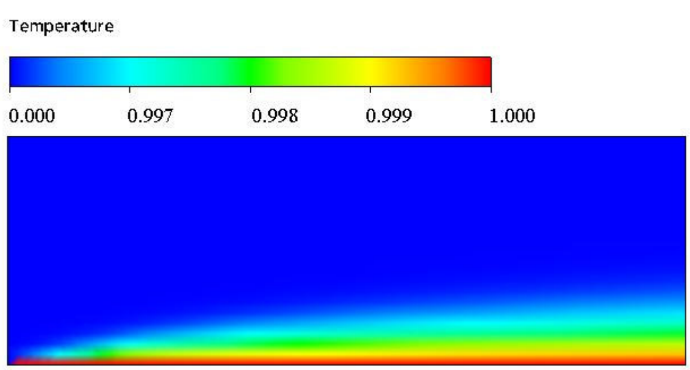
$$\overline{Nu} = \frac{1}{L}\int_0^L \frac{d\theta}{dx^*} dx^* \tag{7}$$

3. RESULTS AND DISCUSSION

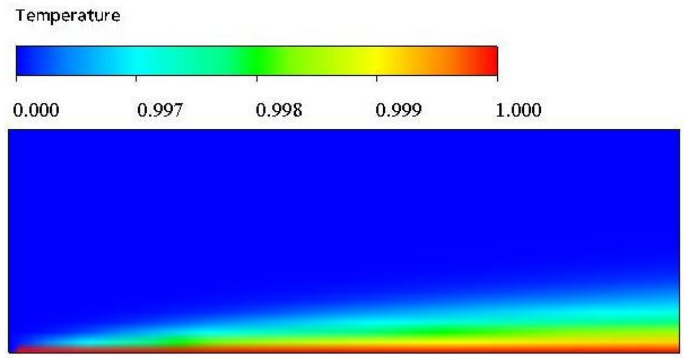
With the objective of proposing a new correlation for the average Nusselt number, a simulation was performed using a time step of 0,05s interrupted when steady stead was reached. The Reynolds number was varied from 2 to 2000. The temperature contour at the end of simulation time for Re = 200 and Re = 2000 are show in figure 3.



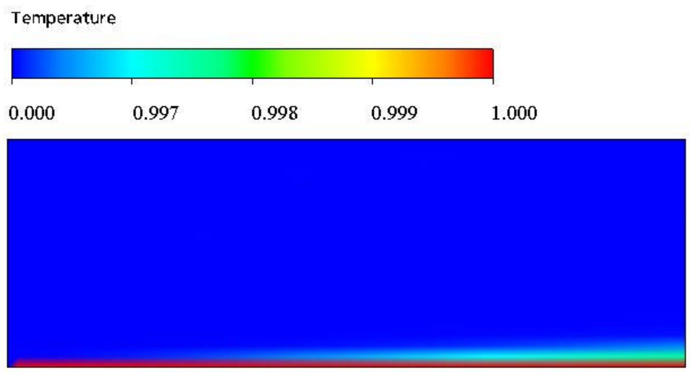
(a)



(b)



(c)



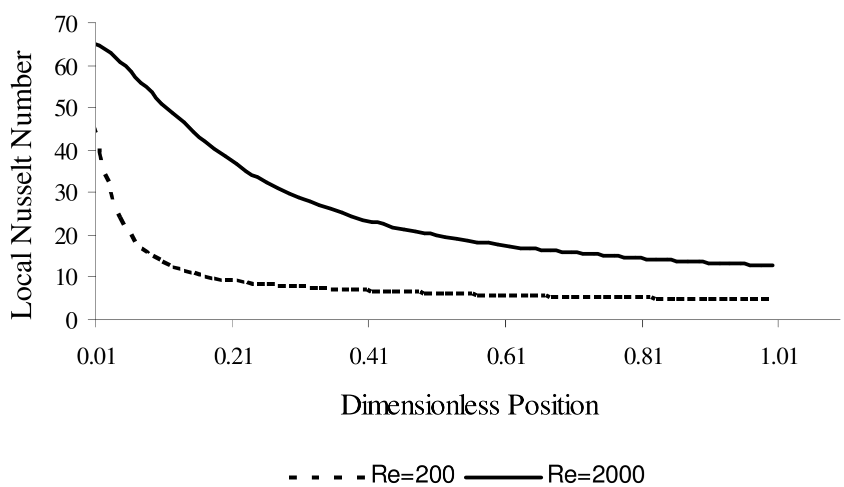
(d)

Figure 3 Temperature contours for (a) $Re_D = 200$, $Pr = 0.1$ e (b) $Re_D = 2000$, $Pr = 0.1$, (c) $Re_D = 200$, $Pr = 1$ e (d) $Re_D = 2000$, $Pr = 1$.

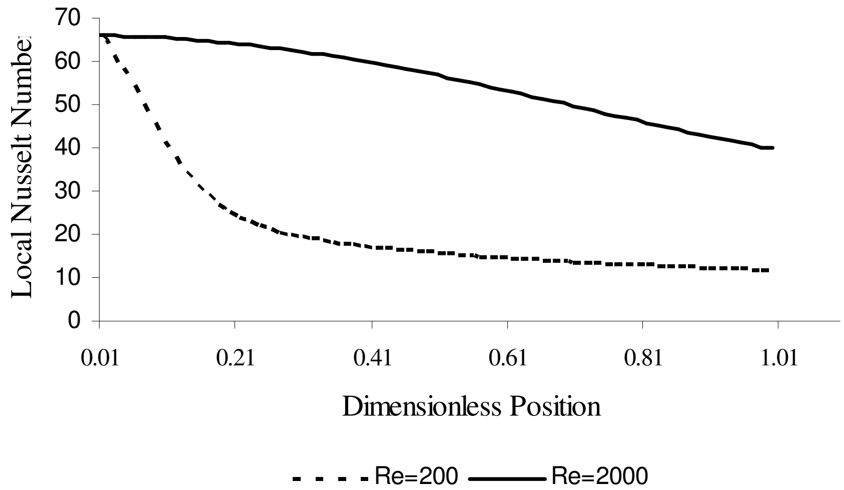
It was observed that increasing the Reynolds number reduces temperature variations along the channel. This is due to the fact that when the Reynolds number increases, the boundary layer thickness decreases.

The local Nusselt numbers are show in figure 4 for Re = 200 and Re = 2000 (Pr = 0.1) in function of position and there appears to be a decrease in the Nusselt number along the length of the channel.

Figure 5 shows the variation in the Nusselt number along the channel. It cam be observed that the Nusselt number stabilizes at a constant value with the increase of the Reynolds number, due to the narrowing of the thermal boundary layer limit.



(a)



(b)
Figure 5. Local Nusselt number variation for (a) Pr = 0.1 e (b) Pr = 1.

In the various branches of Engineering, average or global Nusselt number correlations are used for heat transfer calculation, as well as refrigeration (Martins e Oliveira, 2007). Experimental data are often used to determine empirical correlations, but numerical simulations can also be used to propose global Nusselt number correlations. These correlations are normally written in the following form:

$$Nu_D = C\,\mathrm{Re}_D^m \mathrm{Pr}^n \tag{8}$$

were C, m e n are Constant.

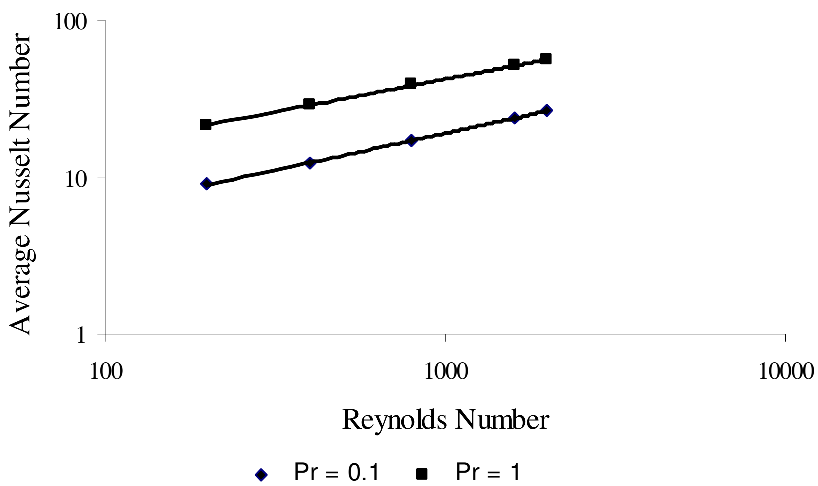


Figure 6. Average Nusselt number as function of Reynolds number.

The Nusselt number correlation founds was:

$$Nu_D = 2.4176\,\mathrm{Re}_D^{0.4136}\mathrm{Pr}^{0.5680} \qquad R^2 = 0{,}9996 \tag{9}$$

It was found a good fit with the numerical results, as verified by R^2 values in Equation 9. Both Figure 6 and Equation 9 show that the global Nusselt numbers increase as the Reynolds number increases.

4. CONCLUSION

The simulation showed that the variations in the Reynolds number are strongly related to the heat transfer from the channel wall to the fluid. The use of CFD to determine the convection coefficient is recommended and reliable, and it is also practical once there is no need to build physical prototypes to proceed the study of this and any others parameters involved in the heat transfer process.

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6. RESPONSIBILITY NOTICE