# POROSITY EFFECT ON FLOW AND HEAT TRANSFER IN A PARALLEL PLATE CHANNEL WITH POROUS AND SOLID BAFFLES

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***Abstract.*** *Simulations are presented for turbulent flow in a channel containing baffles made with solid (impermeable) and porous materials. The equations of mass continuity, momentum and energy are written for an elementary representative volume yielding a set of equations valid for the entire computational domain. These equations are discretized using the control volume method and the resulting system of algebraic equations is relaxed with the SIMPLE method. The presented numerical results for the friction factor f and for the Nusselt number Nu were compared with available data. Further simulations comparing the effectiveness of using porous material showed that no advantages are obtained when low porosity baffles are used in turbulent flow regime.*

***Keywords***: *Porous Media, Heat Transfer, Porous Baffles, Channel flow*

## 1. Introduction

Flow and heat transfer in channels with solid or impermeable baffles have attracted attention of researchers due to the technological advantages in increasing energy transfer in equipment. If the baffles are made of a permeable or perforated material, flow head losses might be substantially reduced. Accordingly, due to the growing applicability of porous media in many fields of engineering and science, such as petroleum and gas engineering, heat exchangers, combustion in porous matrices, cooling of electronic devices, to mention only a few, a good understanding of transport processes in such media is of advantage to the design and analysis of engineering equipment.

Berner, et. al. (1984) presented the main features of the flow in a channel with solid baffles (Figure 1a). This was obtained using flow visualization techniques, manometry, and laser-Doppler anemometry. The experiment was applied to a two-dimensional water flow around baffles, with *L/H*=1, *h/H* from 0.5 to 0.9, and Reynolds number *Re* ranging from 600 to 10,500. They suggested that laminar flow occurred for Re ≤ 600. Following the works of Berner, et. al. (1984), Kelkar & Patankar (1987) presented numerical simulations for laminar flow in a channel with solid baffles for *Re* raging from 100 to 500. They concluded that for high Prandtl number fluids heat transfer within the channel is significantly enhanced, which is a result in agreement with similar predictions by Webb & Ramadhyani (1985). Lopez, et. al (1996) investigated three-dimensional effects for this same geometry and obtained similar results as those taken at the center of the channel.

The first experimental work investigating the use of permeable baffles, instead of using solid plates, was presented by Hwang (1997). In that work, the author found out that heat transfer between the channel walls and the fins was enhanced, showing then that for turbulent flow the use of porous baffles benefited heat transfer, i.e. the use of porous baffles in substitution to solid material represented a net gain. Motivated by this article, Yang & Hwang (2003) carried out a numerical investigation of the problem, reaching similar conclusions. For corroborating Hwang (1997) conclusion, Ko & Anand (2003) carried out an experimental program for turbulent flow with porous baffles made of different materials. However, results in Ko & Anand (2003) were not very promising. In their analysis, the porous baffles presented in most cases a flow behavior as good as the one with solid baffles. Miranda & Anand (2004) presented a numerical analysis of laminar air flow in a channel with porous baffles having height *h/H* = 0.33, thickness *t/H* from 0.09 to 0.25, permeability *K* from $7.6\times10^{-8}\,\mathrm{m}^2$ to $1\times10^{-7}\,\mathrm{m}^2$, porosity $\phi = 0.92$ and $\lambda = k_s t/(k_f L)$ ranging from 0.09 to 330. They noticed that in none of the cases analyzed (with the substitution of the solid baffles for porous baffles) an increase in the effective heat transfer along the channel was obtained.

Following up with the work on baffled channels, Santos & de Lemos (2004) and de Lemos & Santos (2004) carried out a numerical analysis of flow and heat transfer in a parallel plate channel containing 16 porous baffles made with material having permeability $K$=1.0×10$^{-9}$ m$^2$ and porosity $\phi$ = 0.40. Their results indicated that, in spite of using a permeable medium, total drag and heat transfer along the channel were close to those obtained with solid plates, possibly due to the extremely low permeability used. Santos & de Lemos (2004) further investigated the influence of permeability and porosity on that geometry with *h/H* = 0.5, *Pr* = 0.7 and $\lambda$= 0.4. Recently, Santos & de Lemos (2005) fully documented such numerical study on laminar flow and heat transfer along a channel with porous and solid baffles. The influence of the permeability and porosity of the material was analyzed, in addition to the effect of the fluid properties ($0.7<Pr<7.0$), baffle material ($0.04<\lambda<40$) and geometry ($0.25 < h/H < 0.75$). They concluded that there was not an effective increase on heat transfer when porous baffles were used.

The purpose of the present contribution is to extend the work in Santos & de Lemos (2005) to turbulent flow regime. The influence of the porosity of the material is analyzed. Here, only one unique set of transport equations is used for both the clear domain and the porous medium.

## 2. Mathematical Model

The mathematical model here employed has its origin in the work of Pedras & de Lemos (2001a). The implementation of the jump condition at the interface was considered in Silva & de Lemos (2003) based on the theory proposed in Ochoa-Tapia & Whitaker (1995). Nevertheless, for the sake of simplicity, in this work no jump condition is considered. Therefore, these equations will be reproduced here and details about their derivations can be obtained in the mentioned papers. These equations are:

### 2.1. Macroscopic continuity equation:

$$\nabla . \mathbf{u}_D = 0 \tag{1}$$

where, $\mathbf{u}_D$ is the average surface velocity (also known as seepage, superficial, filter or Darcy velocity). Equation (1) represents the macroscopic continuity equation for an incompressible fluid.

### 2.2. Macroscopic momentum equation:

$$\rho \nabla \cdot \left( \frac{\overline{\mathbf{u}}_D \overline{\mathbf{u}}_D}{\phi} \right) = -\nabla \left( \phi \langle \overline{p} \rangle^i \right) + \mu \nabla^2 \overline{\mathbf{u}}_D + \nabla \cdot \left( -\rho \phi \langle \overline{\mathbf{u}'\mathbf{u}'} \rangle^i \right) + \phi \rho \mathbf{g} - \left[ \frac{\mu \phi}{K} \overline{\mathbf{u}}_D + \frac{c_F \phi \rho |\overline{\mathbf{u}}_D| \, \overline{\mathbf{u}}_D}{\sqrt{K}} \right] \tag{2}$$

where the last two terms in equation (2), represent the Darcy-Forchheimer contribution. The symbol $K$ is the porous medium permeability, $c_F = 0.55$ is the form drag coefficient (Forchheimer coefficient), $\langle p \rangle^i$ is the intrinsic (volume-averaged on fluid phase) pressure of the fluid, $\rho$ is the fluid density, $\mu$ represents the fluid viscosity and $\phi$ is the porosity of the porous medium.

Here, a High Reynolds macroscopic $k-\varepsilon$ closure, which models the term $\nabla \cdot \left( -\rho \phi \langle \overline{\mathbf{u}'\mathbf{u}'} \rangle^i \right)$ and uses wall functions to handle wall proximity, is applied. As mentioned, details on the mathematical model employed are fully documented in the open literature and for that the equations will be presented here without any derivation. The interested reader is referred to the works of Pedras & de Lemos (2000-2003) for details. In that model, a macroscopic eddy-diffusivity $\mu_{t_\phi}$ is given by

$$\mu_{t_\phi} = \rho C_\mu f_\mu \frac{\langle k \rangle^{i^2}}{\langle \varepsilon \rangle^i} \tag{3}$$

where $k$ and $\varepsilon$ are computed via transport equations modeled in the form,

$$\rho \nabla \cdot \left( \overline{\mathbf{u}}_D \langle k \rangle^i \right) = \nabla \cdot \left[ \left( \mu + \frac{\mu_{t_\phi}}{\sigma_k} \right) \nabla \left( \phi \langle k \rangle^i \right) \right] - \rho \langle \overline{\mathbf{u}'\mathbf{u}'} \rangle^i : \nabla \overline{\mathbf{u}}_D + C_k \rho \frac{\phi k_\phi |\overline{\mathbf{u}}_D|}{\sqrt{K}} - \rho \phi \langle \varepsilon \rangle^i \tag{4}$$

$$\rho \nabla \cdot \left( \overline{\mathbf{u}}_D \langle \varepsilon \rangle^i \right) = \nabla \cdot \left[ \left( \mu + \frac{\mu_{t_\phi}}{\sigma_\varepsilon} \right) \nabla \left( \phi \langle \varepsilon \rangle^i \right) \right] + C_1 \left( -\rho \langle \overline{\mathbf{u}'\mathbf{u}'} \rangle^i : \nabla \overline{\mathbf{u}}_D \right) \frac{\langle \varepsilon \rangle^i}{\langle k \rangle^i} + C_2 f_2 C_k \rho \frac{\phi \varepsilon_\phi |\overline{\mathbf{u}}_D|}{\sqrt{K}} - C_2 f_2 \rho \phi \frac{\langle \varepsilon \rangle^{i^2}}{\langle k \rangle^i} \tag{5}$$

### 2.3. Macroscopic Energy Equation:

$$\begin{aligned} \left( \rho c_p \right)_f \nabla \cdot \left( \overline{\mathbf{u}}_D \langle \overline{T} \rangle^i \right) = {} & \nabla \cdot \left\{ \left[ k_f \phi + k_s (1-\phi) \right] \nabla \langle \overline{T} \rangle^i \right\} + \nabla \cdot \left[ \frac{1}{\Delta V} \int_{A_i} \mathbf{n} \left( k_f \overline{T_f} - k_s \overline{T_s} \right) dS \right] + \\ & - \left( \rho c_p \right)_f \nabla \cdot \left[ \phi \left( \overline{\langle \mathbf{u}' \rangle^i \langle T_f' \rangle^i} + \langle {}^i \overline{\mathbf{u}}^i \, \overline{T_f} \rangle^i + \langle \overline{{}^i \mathbf{u}' \, {}^i T_f'} \rangle^i \right) \right] \end{aligned} \tag{6}$$

where $T_f$ and $T_s$ are the local fluid and solid temperatures, respectively, and $k_f$ and $k_s$ are the thermal conductivities for the fluid and for the solid. Further, the homogenous model assumes that intrinsic (phase-average) temperatures for the solid and for the fluid are equal, or $\langle T_s \rangle^i = \langle T_f \rangle^i = \langle T \rangle^i$.

At the interface, the conditions of continuity of velocity, pressure and temperature reads,

$$\mathbf{u}_D \big|_{0<\phi<1} = \mathbf{u}_D \big|_{\phi=1} \tag{7}$$

$$\langle p \rangle^i \big|_{0<\phi<1} = \langle p \rangle^i \big|_{\phi=1} \tag{8}$$

$$\langle T \rangle^i \big|_{0<\phi<1} = \langle T \rangle^i \big|_{\phi=1} \tag{9}$$

The non-slip condition for velocity is applied on both walls.

## 3. Numerical Method and Computational Details

The numerical method utilized to solve the flow equations is the finite volume method applied to a boundary-fitted coordinate system. Equations (1)-(6) subjected to boundary and interface conditions Eqs. (7)-(9), were discretized in a two-dimensional control volume involving both clear and porous media. The numerical method used in the resolution of the equations above was the SIMPLE algorithm, also described in Patankar (1980). The interface is positioned to coincide with the border between two control volumes, generating, in such a way, only volumes of the types 'totally porous' or 'totally clear'. The flow equations are then resolved in the porous and clear domains, considering the interface conditions mentioned earlier. Details of the numerical implementation can be seen in Pedras & de Lemos (2001b-2003), and Silva & de Lemos (2003).

### 3.1. Friction Factor

The friction factor in a section of the channel, shown in Figure 1b, can be calculated as,

$$f = \frac{\Delta P}{\rho} \frac{D_h}{L} \frac{2}{\overline{U}^2} \tag{10}$$

where $\Delta P$ is the pressure loss for the cell, $\overline{U}$ is the mean velocity and $D_h$ =2H is the hydraulic diameter. Also, for flow in between infinite parallel plates without baffles, the friction factor is given by, $(f\,\mathrm{Re})_0 = 0.33\,\mathrm{Re}^{-0.25}$, as Blasius and Dittus-Boelter have suggested, where the subscript "0" identifies values for unobstructed channels and the Reynolds number is defined as $\mathrm{Re} = \rho \overline{U} D_h / \mu$.

### 3.2. Nusselt Number

Heat transfer between the channel walls and the fluid, in the cell shown in Figure 1b, is numerically calculated using the Nusselt number defined by,

$$\overline{Nu} = \frac{\overline{h} D_h}{k_f} \tag{11}$$

where

$$\overline{h} = \frac{Q_{cell}}{(2L \times 1)\,(\overline{T_w} - \overline{T_{bCell}})} \tag{12}$$

$$\overline{T_{bCell}} = \int_0^L \frac{\int_0^H uT dy'}{\int_0^H u dy'} dx' \tag{13}$$

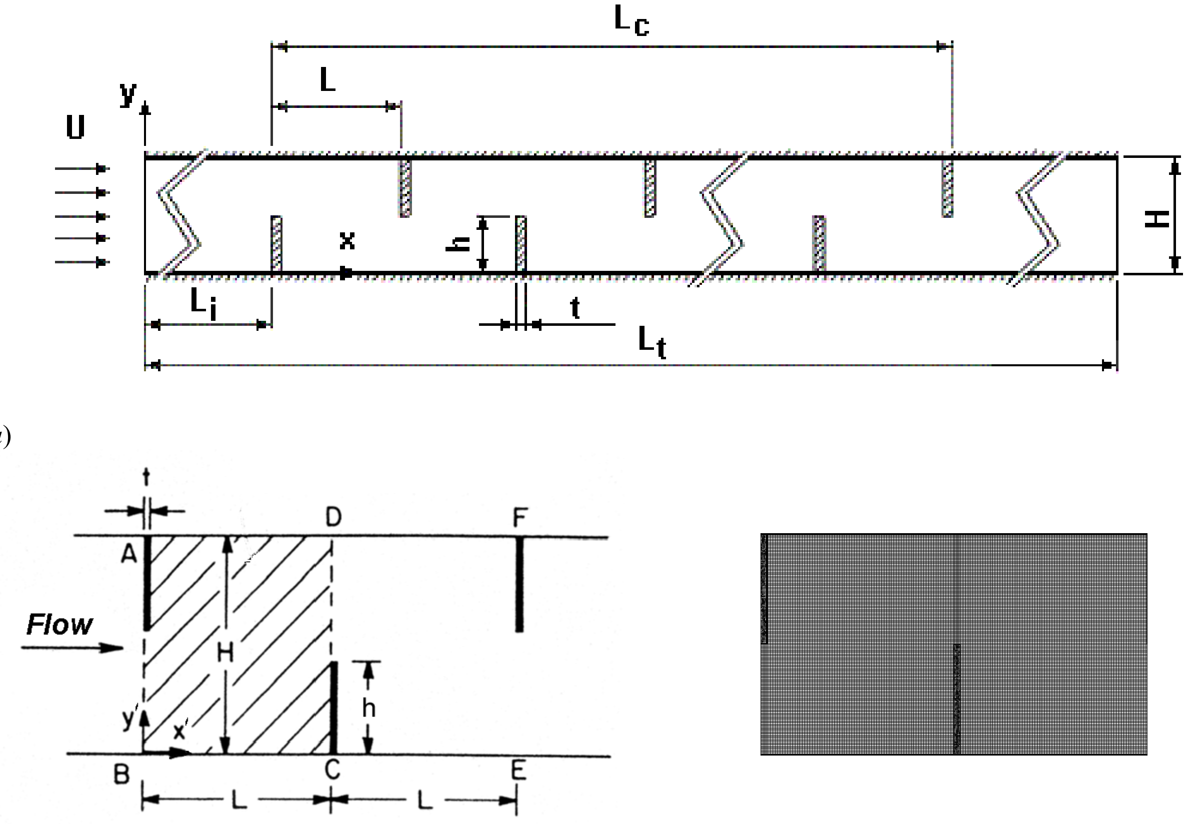
This formulation takes into consideration the constant heat transferred to the fluid, $Q_{cell} = 4000\ \mathrm{W/m^2}$, through the surfaces, the cell bulk temperature, $\overline{T_{bCell}}$ and the medium wall temperature $\overline{T_w}$, that is the media between the north and south wall, for each cell.

## 4. Results and Discussion

Before proceeding, a word about the nomenclature, simulating conditions and grid size here employed seems timely.

In the context herein, the expression "solid baffle" means a structure through which no fluid permeates. On the other hand, "porous baffles" are characterized by a solid matrix, not fully compacted, so that fluid can freely flow within the existing void space. In addition, all results in the figures are divided by corresponding values obtained in channels without baffles, which are here identified with subscript "0".

Different grid sizes were employed in order to check their influence on the numerical values of $f$ and $Nu$, calculated in the fully developed channel cell schematically shown in Figure 1b. Grid independence studies indicated that for a grids of size greater than 102x1057 no substantial differences were detected in the solution. All computations below were then obtained with such grid, which is illustrated in Figure 1c.



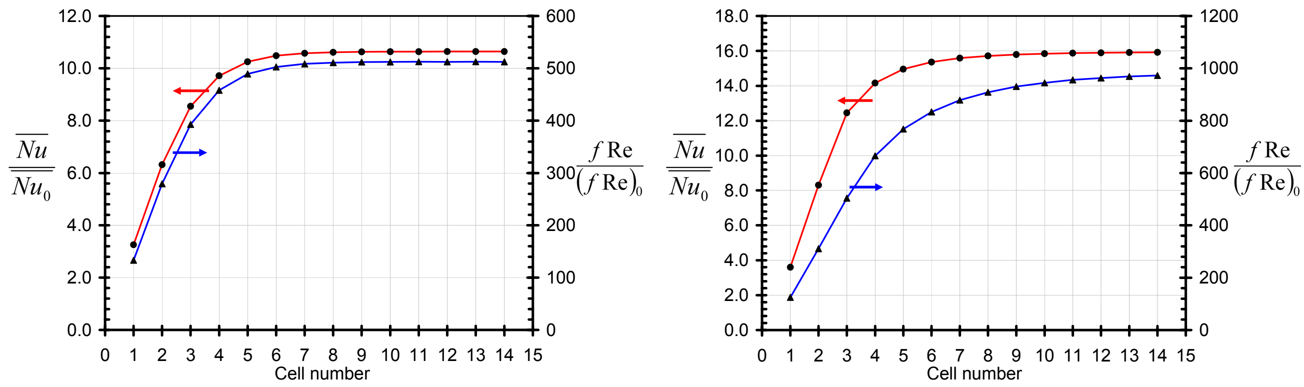

*a*)

*b*) *c*)

Figure 1 – Problem under consideration: *a*) Geometry, *b*) Channel cell, *c*) Section of computational grid of length $2L$.

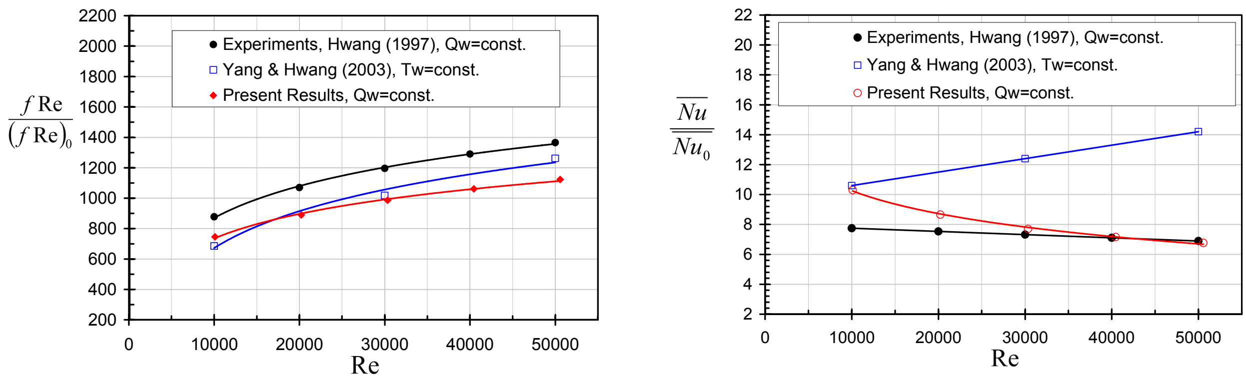
### 4.1. Developing Flow

Figure 2 shows numerical results for the normalized friction factor and Nusselt number along the channel in Figure 1a, already integrated over each cell shown in Figure 1b. The figure indicates that the developing length is a function of the Reynolds number and that for higher values of $Re$ a greater number of cells is necessary for thermal-hydraulic development. Also, when simulations herein are compared with those by Berner, et. al. (1984) and Hwang (1997), who indicated that after 5 cells flow properties reached constant values, we notice that even at $Re$=10000 it takes longer for the flow to reach the fully developed condition. For that, in this work all simulations shown below, which consider fully developed flow and heat transfer parameters, are taken past the $12^{th}$ cell. At that position, the friction factor differs by less than 1% from the one at next station.




a) b)

Figure 2 – Developing *Nu* and *f* along the channel, *h/H*=0.5: a) *Re*=10000, b) *Re*=50000



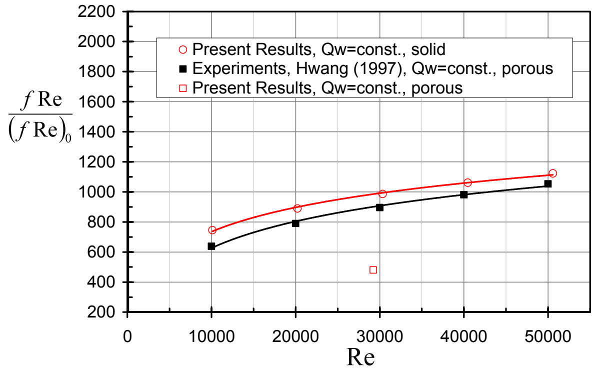

a) b)

Figure 3 – Comparison of friction factor and Nusselt number for solid baffles with *h/H*=0.50, *Pr* = 0.7 and $\lambda$ = 40.

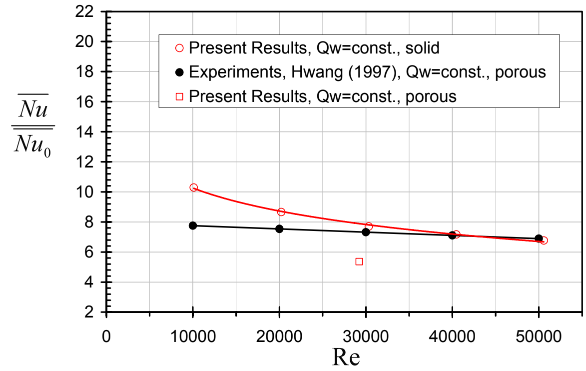
**4.2. Fully Developed Flow**

Figure 3 shows computations for the solid baffle case compared with experiments by Hwang (1997) and simulations of Yang & Hwang (2003). Results are for analyzing the influence of *Re* on the integrated parameters. On the overall, results herein for *f* (Figure 3a) are lower that those by Hwang (1997) and close to those presented in Yang & Hwang (2003), which were conducted using a different boundary condition for temperature. Further, Figure 3b indicates that *Nu* agrees well with available experimental values and that there seems to be a trend towards reduction of Nusselt as *Re* increases. This observation, however, contradicts similar computation shown in Yang & Hwang (2003).

Computations using porous baffles are further compared with experimental values in Figure 4. So far, only for *Re*=30000 and for porous baffles with porosity $\phi$ = 0.42 and permeability $K = 8.25\times10^{-10}$ m² have computations been performed. This unique result is presented in Figure 4. When permeable plates are employed, a lower friction factor is obtained. Computations herein, however, fall lower than data by Hwang (1997). It is believed that the source of such discrepancy might be associated with the use of an inadequate combination of computational grid size and layout and of a High *Re* turbulence model. It is well known that High *Re* models perform better when the imposed wall condition for velocity (lag-law) is coherent with the position of the first grid point close to the wall. For flows within the cell channel of Figure 1b, an optimal wall distance for the first grid point might be dependent on the axial position of the node closest to the wall. However, the grid sketched in Figure 1c, which is based on a Cartesian coordinate system, considers the same distance for all nodes in the axial direction. As such, modifying the location of the first grid point close to the walls might have influence on the final results and efforts are being made in order to fully access such dependency. Nevertheless, results herein can be seen as a first step towards obtaining reliable computations.




a)




b)

Figure 4 – Comparison of friction factor and Nusselt number for porous baffles with *h/H*=0.50, porosity $\phi = 0.42$, permeability $K = 8.25 \times 10^{-10}$ m².

Figure 5 shows computations for *Nu* when porosity is varied. The single value obtained by Hwang (1997) is also plotted for comparison. As seen, calculated Nusselt values are lower than similar results available in the literature, as stated earlier (see Figure 4). Further, the friction factor increases slightly with the baffle porosity and the Nusselt number remains almost insensitive to variations in the value of $\phi$. In Santos & de Lemos (2005), a stronger sensitivity on the numerical values of *K*, rather than on variations in porosity, has also been observed for laminar flows.

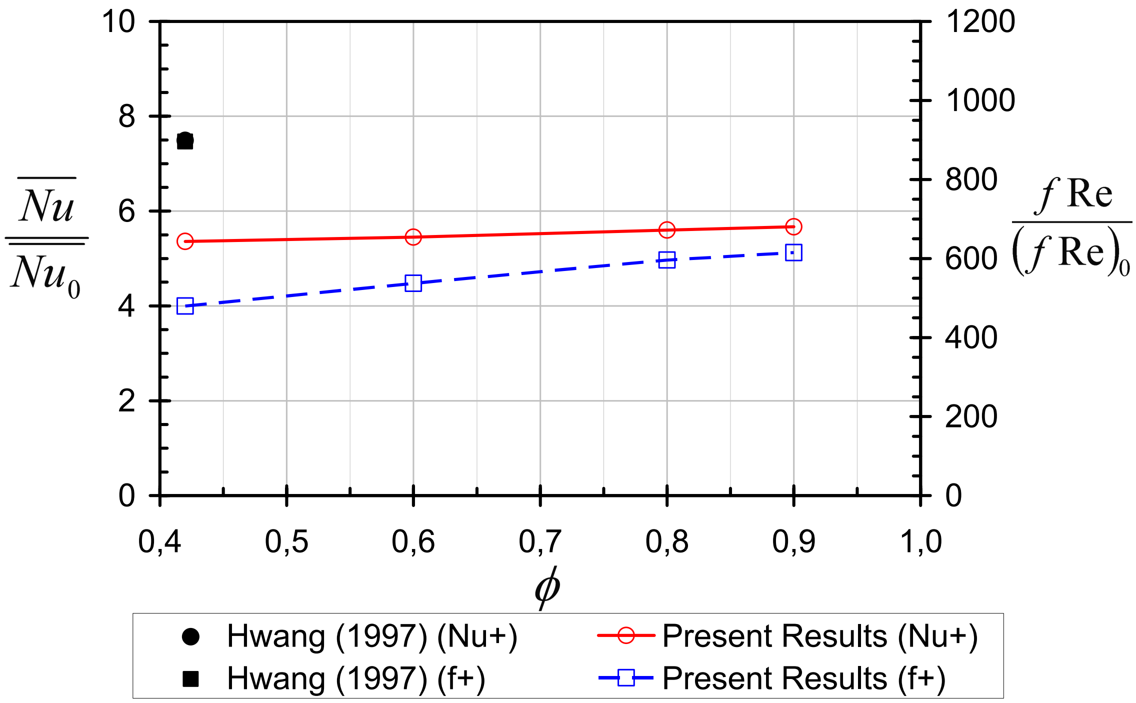



Figure 5 – Porosity effect on normalized *f* and *Nu* for *h/H*=0.50, $K = 8.25 \times 10^{-10}$ m$^2$ and *Re*=30000.

### 4.3. Effective Heat Transfer

For analyzing the effective influence of the type of obstruction (solid or porous), a modified Nusselt number $\overline{Nu}^* = \left(\overline{Nu}/\overline{Nu_0}\right)/\left(f/f_0\right)^{1/3}$ has been proposed in the literature, Ko & Anand (2003). This modified Nusselt number is a measure of the head loss, when a porous medium is used, with the corresponding reduction in the heat transfer characteristics. Such analysis is useful when comparing performances of baffles of different types. Ultimately, if using porous baffles increases the value of $\overline{Nu}^*$, in relation to using solid material, then the system is considered to be more effective. In this case, although *Nu* is reduced, the corresponding decrease in the necessary pumping power (reduction of *f* ) makes it advantageous the use of permeable material for manufacturing the baffles (see Ko & Anand (2003) for details). Table 1 finally presents numerical results for $\overline{Nu}^*$. For the entire range of *Re* here analyzed, computations by Hwang (1997) clearly indicate a net gain (increase in $\overline{Nu}^*$ value) when porous baffles are employed. This result is not endorsed by the computations herein. Further, the table shows that the numerical value of $\overline{Nu}^*$ remains nearly the same as the porosity of the baffles increases. As stated earlier, further investigation is necessary in order to access the role of grid layout and related wall boundary models on the results. Also, the use of mode elaborated turbulence models, such as Low Re closures, might bed necessary in order to improve the accuracy of computations here presented.

Table 1 – Effective heat transfer coefficient $\overline{Nu}^*$ for solid and porous baffles, with *h/H*=0.50, $K = 8.25\times10^{-10}$ m$^2$, *Pr*=0.7 and $\lambda = 40$.

| $\phi$ | solid | solid | 0.42 | solid | 0.42 | 0.60 | 0.80 | 0.90 |
|---|---|---|---|---|---|---|---|---|
| *Re* | Yang & Hwang (2003) | Hwang (1997) | | Present Results | | | | |
| 10000 | 1.20 | 0.81 | 0.99 | 1.14 | --- | ---- | ---- | ---- |
| 30000 | 1.23 | 0.69 | 0.78 | 0.77 | 0.68 | 0.67 | 0.67 | 0.67 |
| 50000 | 1.31 | 0.62 | 0.69 | 0.65 | --- | ---- | ---- | ---- |

## 5. Conclusion

This work presented results for the numerical solution of turbulent flow in a channel containing porous and solid obstructions. Discretization of the governing equations used the finite volume method of and the set of algebraic equations was solved by the SIMPLE method. The presented numerical results for the friction factor *f* and for the Nusselt number *Nu* were compared with data of Hwang (1997) and Yang & Hwang (2003). Further simulations comparing the effectiveness of the porous material used showed that no advantages are obtained when using permeable baffles in the turbulent flow regime. Finally, results herein are encouraging and motivate further analyses for simulating turbulent flow in porous-baffled channels.

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