HEAT TRANSFER ENHANCEMENT IN LAMINAR FORCED CONVECTION

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INTRODUCTION

Within the scope of thermal engineering, energy conservation and sustainable development demands have been driving research efforts towards more energy efficient equipments and processes. In this context, the scale reduction in mechanical fabrication has been permitting the miniaturization of thermal devices, such as in the case of micro-heat exchangers. Novel experimental, modeling and simulation approaches have been required to explain deviations of the heat transfer behavior of micro-systems as compared to classical macro-scale phenomena. Also, a new class of thermal carrier fluids are under analysis within various research groups worldwide, based on the suspension of metallic or metal oxide nanoparticles within the most usual liquids employed in heat exchange equipment. Thus, nanotechnology and micro-fabrication techniques have been allowing for the enhancement and development of a number of engineering applications, while providing new challenging scientific perspectives in fundamental research.

The petroleum and process industries have been quite active in progressively incorporating heat transfer enhancement solutions to the efficiency increase requirements along the years. More recently, heat exchangers employing micro-channels with characteristic dimensions below 500 microns have been calling the attention of researchers and practitioners, towards applications that require high heat removal demands and/or space and weight limitations. Motivated by the search for optimal solutions in heat exchange rates, Steinke & Kandlikar [2004] critically analyzed various heat transfer enhancement techniques as applied to the micro-channels scale. Among the passive enhancement techniques then discussed, the authors emphasize the utilization of treated surfaces, rough or corrugated walls and additives for working fluids. Several other approaches were disregarded in light of the difficulties in manufacturing or mechanically modifying the thermal system at the micro-scale.

Our research effort in this context is first related to the fundamental analysis of forced convection within micro-channels, as required for the design of micro-heat exchangers, including the effects of axial heat conduction and wall corrugation or roughness on heat transfer enhancement, Castellões & Cotta [2008]. Second, the research is also directed towards the fabrication, characterization, modeling, and experimental evaluation of nanofluids and their associated thermal convective behavior in laminar flow applications, Cotta et al. [2007]. The present lecture specifically addresses the convective heat transfer within micro-channels possibly enhanced by the presence of axial heat conduction in the fluid and wall corrugations. First, the typical low Reynolds numbers in such micro-systems may lead to low values of the Peclet number that bring up some relevance to the axial heat diffusion along the fluid stream, especially for regions close to the inlet. Then, both the upstream and downstream sections of the micro-channel that are not actually part of the heat transfer section, may participate in the overall heat transfer process, and finally yield different predictions than those reached by making use of conventional macro-scale relations for ordinary liquids or gases. Second, either due to the inherent difficulties in achieving smooth surfaces during micro-fabrication processes or to the actual purpose of improving mixing and/or heat transfer, micro-channels with irregularly shaped walls started gaining some focus in the heat and mass transfer literature. Thus, the analysis of laminar forced convection within micro-channels with corrugated walls, and the possible heat transfer enhancement effect achieved, is another objective of the present study. Then, the marked variation with temperature of the themophysical properties associated with nanofluids, especially thermal conductivity and viscosity, has here deserved a critical inspection of such effects on heat transfer coefficients in developing laminar tube flow.

PROBLEM FOMULATION

Smooth and Corrugated Micro-channels

We consider transient laminar forced convection within micro-channels formed by smooth or corrugated plates. Three regions along the channel are considered in the problem formulation, as described in Figure 1 below. First, an adiabatic region with smooth walls, followed by the heat transfer section with prescribed temperatures at the corrugated walls, and the third one, following the corrugated region, is again made of smooth adiabatic walls. The problem formulation adopts the same geometry and boundary conditions as presented by Wang & Chen [2002]. The walls boundaries are described by sinusoidal functions along the longitudinal coordinate in the specific example here considered. The two-dimensional flow is assumed to be laminar and incompressible, with temperature independent thermophysical properties, while viscous dissipation and natural convection effects are neglected. Due to the possible low values of Peclet number, in light of the lower range of Reynolds numbers, axial diffusion along the fluid is not disregarded.

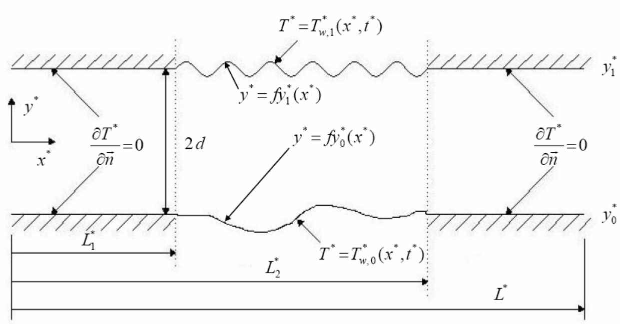

Figure 1 – Geometry and coordinates system for heat transfer in corrugated channel.

In obtaining the velocity field along the flow, the full Navier-Stokes equations are employed, yielding the variable streamfunction and velocity components along the transversal and longitudinal directions, where the Generalized Integral Transform Technique (GITT), Cotta[1993], has been employed in the hybrid numerical-analytical solution of this laminar flow. Also, for sufficiently low Reynolds numbers and smooth variations on the wall corrugations, an approximate solution has also been examined, essentially by accounting for the variable cross section within the local mass balance equation, but neglecting the momentum equations influence on the velocity components modification. These explicit solutions for the velocity components are particularly handy, especially in design and optimization tasks. Once the velocity field is available, the energy balance is given as:

$$\frac{\partial T^*(x^*,y^*,t^*)}{\partial t^*}+u^*(x^*,y^*)\frac{\partial T^*}{\partial x^*}+v^*(x^*,y^*)\frac{\partial T^*}{\partial y^*}=\alpha\left(\frac{\partial^2 T^*}{\partial x^{*2}}+\frac{\partial^2 T^*}{\partial y^{*2}}\right) \quad (1a)$$

$$T^*(x^*,y^*,0)=T_0^*(x^*,y^*)\,,\quad T^*(0,y^*,t^*)=T_e^*(y^*,t^*)\,,\quad \left.\frac{\partial T^*}{\partial x^*}\right|_{x^*=L^*}=0 \quad (1b\text{-}d)$$

$$\text{for } 0<x^*<L_1^*,\ \begin{cases}\left.\dfrac{\partial T^*}{\partial y^*}\right|_{y^*=fy_0^*(x^*)}=0\\ \left.\dfrac{\partial T^*}{\partial y^*}\right|_{y^*=fy_1^*(x^*)}=0\end{cases},\quad \text{for } L_1^*\le x^*\le L_2^*,\ \begin{cases}T^*(x^*,y^*=fy_0^*(x^*),t^*)=T_{w,0}^*\\ T^*(x^*,y^*=fy_1^*(x^*),t^*)=T_{w,1}^*\end{cases} \quad (1e\text{-}h)$$

$$\text{for } L_2^*<x^*<L^*,\ \begin{cases}\left.\dfrac{\partial T^*}{\partial y^*}\right|_{y^*=fy_0^*(x^*)}=0\\ \left.\dfrac{\partial T^*}{\partial y^*}\right|_{y^*=fy_1^*(x^*)}=0\end{cases} \quad (1i,j)$$

Forced Convection in Nanofluids with Temperature Dependent Properties

We consider forced convection heat transfer inside a circular tube for incompressible laminar flow of a Newtonian liquid with temperature dependent thermophysical properties, including viscosity, thermal capacitance, and thermal conductivity. The tube is subjected to a prescribed uniform wall heat flux, with uniform inlet temperature and negligible viscous dissipation effects. This problem has a strong practical motivation with a renewed interest due to more recent applications in forced convection such as microchannels and nanofluids. The related energy equation and inlet and boundary conditions are written as, Cotta et al. [2007]:

$$\rho(T)c_p(T)u(r,T)\frac{\partial T(r,z)}{\partial z}=\frac{1}{r}\frac{\partial}{\partial r}\left[rk(T)\frac{\partial T(r,z)}{\partial r}\right],\ 0<r<r_w,\ z>0 \quad (2a)$$

$$T(r,0)=T_0,\quad 0\le r\le r_w \quad (2b)$$

$$\frac{\partial T(r,z)}{\partial r}=0,\ \ r=0;\quad -k(T)\frac{\partial T(r,z)}{\partial r}=-q_w,\ \ r=r_w, z>0 \quad (2c,d)$$

where the temperature dependent fully developed velocity profile is obtained by the GITT of the related momentum equation in boundary layer formulation, coupled to the above thermal problem.

RESULTS AND DISCUSSION

Figure 2 illustrates the effect of the Peclet number on the bulk temperatures for the steady-state situation for the corrugated micro-channel, by taking the two values Pe=10 and 30. One may clearly observe the more significant pre-heating effect in region 1 due to the lower value of Pe, but also the more pronounced effects on the bulk temperature fluctuations due to the wall corrugations in the case of a smaller axial diffusion of heat, when the transversal effects start playing a major role. For the heat transfer enhancement analysis it is of interest to evaluate the behavior of the Nusselt number under different corrugation conditions. Figure 3, for instance, illustrates the local Nusselt number results for Pe=10 and α = 0.1 and 0.2. The smooth parallel plates case is also plotted for reference purposes, as the solid black line. One may see that even with the lower corrugation amplitude value some noticeable heat transfer enhancement is already evident, and marked increases in the local heat transfer coefficient are achieved for the higher corrugation amplitude value for this value of Pe.

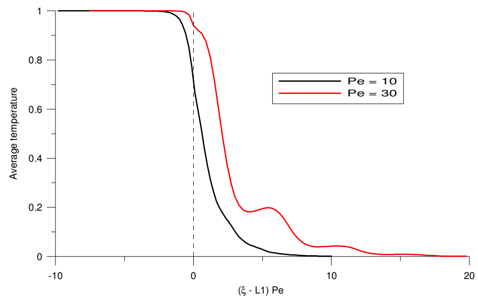

Figure 2 – Influence of Peclet number on the bulk temperature behavior along regions 1 and 2 in steady-state, for Pe=10 and 30, and α= 0.1.

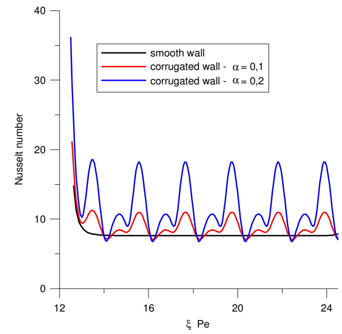

Figure 3 – Local Nusselt numbers at steady-state for smooth and corrugated channels with Pe=10 and α =0 (parallel-plates), 0.1 and 0.2.

Figure 4 below illustrates the dimensionless temperature radial distributions along the channel length for the nanofluid laminar flow in a circular tube, for different axial locations, which are here represented by colors ranging from pure blue to pure red (Z=0.0013, 0.0179, 0.0353, 0.0699, 0.1080, 0.1480, 0.1807, 0.2180). The solid lines correspond to the full nonlinear formulation here considered while the dashed lines are obtained from the classical linear formulation of Graetz problem. As expected the deviations are more significant within the regions of larger temperature gradients, corresponding to regions closer to the wall and as the fluid heating progresses. Also, the heat transfer enhancement effect may be observed in the reduction of the duct wall temperatures as the nonlinear properties are accounted for, especially due to the reduction of the viscosities close to the hotter duct wall, with the subsequent fluid acceleration in this region.

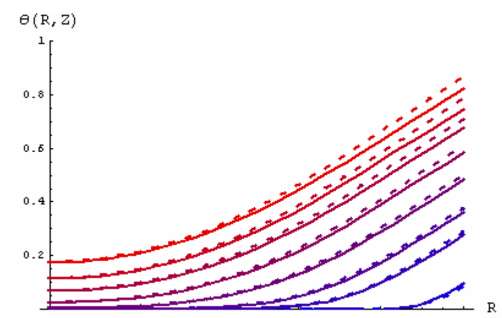

Figure 4- Dimensionless radial temperature distributions for linear (dashed lines) and nonlinear (solid lines) formulations and axial positions increasing from blue to red.

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