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Experimental assessment of the fluid bulk temperature profile in a mini channel through inversion of external surface temperature measurements

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ABSTRACT

In previous works, a semi-analytical model allowing the simulation of heat transfer in a mini-channel has been presented. This model does not rely on the local internal heat transfer coefficient distribution. The focus of this study concerns the experimental characterization of the conjugate heat transfer in a vertical flat mini-channel of 1 mm thickness. The objective here is to validate experimentally the previous approach on an experimental bench by using infrared thermography measurements on the external faces of the channel. The temperature observations are used then within an inverse approach in order to recover the external heat transfer coefficient (air) over the external faces (parameter estimation problem) as well as the internal boundary conditions (function estimation inverse problem). The interest of such an inverse approach, coupling a model and measurements, is to avoid introducing an intrusive instrumentation at this scale. It allows to validate a semi-analytical heat transfer model that takes into account conduction and advection in the fluid as well as conduction in the solid (conjugate heat transfer) without the use of any internal heat transfer coefficient.

1. Introduction

Single-phase forced convection in micro- and mini- channels, whose hydraulic diameter ranges from a few micrometers to several millimeters, has been the subject of an increased interest in the last decade. This stems from several advantages of such systems. Their compactness, that is the high ratio of fluid/wall interface with respect to the total volume of the system, allows large rates of heat flow. The low volume of circulating fluid makes them interesting when the fluid is toxic, harmful to the environment or simply expensive. However some drawbacks have also to be considered: high pressure drops may be associated to their small hydraulic diameters, since in laminar regime the pressure drop is inversely proportional to the fourth power of the hydraulic diameter, for a given volume flow rate [1], especially in the presence of sharp bends for high enough flow rates [2], and pure fluids have to be used, in order to avoid particle deposition and fouling.

These channels are used in several heat transfer applications such as the cooling of microprocessors by compact heat exchangers or the control of the temperature in injection molding of plastic or composite parts, just to quote a few. The design of such experimental devices requires a modeling of heat transfer at length scales which differ from those of the traditional macro-systems [3]. So, the high volume fraction of the solid walls modifies the transfer of heat at the solid wall/fluid flow interface: the distribution of the heat flux at this interface is not always normal to it and the effect of axial conduction has to be taken into account [4]. This effect depends not only on the characteristics of the flow (Reynolds and Prandtl numbers, hydraulic diameter and length of the channel) but also on the nature of the material used for the solid wall (thermal conductivity) as well as its thickness.

In a previous work [5], a semi-analytical steady state 2D model allowing the simulation of heat transfer in a flat mini-channel has been presented. This model does not rely on any assumption regarding the local internal heat transfer coefficient distribution. The only hypothesis is that the flow is laminar and developed and that heating is implemented on a part of the two external surfaces. The inverse problem of estimating the internal temperature and flux

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Nomenclature

c	heat capacity, J/kg K
e	thickness, m
$\mathbf{F}$	fluid transfer matrix
$\mathbf{H}$	external transfer matrix
h	heat transfer coefficient, W m^{-2} K^{-1}
$\mathcal{J}$	cost function
$2L$	virtual channel length, m
$2l$	channel length, m
M	axial conduction number
Pe	Péclet number
Re	Reynolds number
$\mathbf{S}$	solid transfer matrix
T	temperature, K
T_∞	ambient temperature, K
U	mean velocity, m/s
$u(y)$	inlet velocity profile, m/s
$x,\ y$	spatial coordinates, m
Greek symbols	
α	hyperparameter for TSVD regularization
α_n	discrete eigenvalue of order n
λ	thermal conductivity, W/m K
ν	kinematic viscosity, m^2/s
φ	heat flux density in y direction, W/m^2
$\tilde{\varphi}$	heat flux Fourier transform
ρ	density, kg/m^3
θ	temperature variation with respect to ambient, K
Superscripts	
T	transposed of a matrix
$\sim$	Fourier transform
Subscripts	
b	bulk
c	cold
f	fluid
h	hot
∞	outside air
s	solid
wc	internal cold wall
wh	internal hot wall

distributions over the boundaries and inside the system (two walls and fluid flow) as well as the fluid bulk temperature profile, starting from the measurement of the corresponding temperature distribution, by infrared thermography, over the external faces has been considered and simulated, using synthetic noisy measurements.

The objective of the present paper is to validate the previous approach using a specific experimental bench, that is to say, using infrared thermography measurements over part of the external faces of the channel. This has to be made in two stages. First, for an independently measured liquid flow rate, the external heat transfer coefficient (air) over the external faces has to be recovered (parameter estimation problem). In a second stage, the fluid (water here) bulk temperature distribution (function estimation inverse problem) has to be retrieved by an inverse technique using some form of regularization, Rectangular Least Squares or Truncated Singular Value Decomposition here.

This kind of non-intrusive inverse technique (there is no temperature sensor inside the flowing liquid) is used here to show the interest of an internal h-less model in a walls/channel conjugate heat transfer configuration [4]. In such a configuration, the traditional approach based on a local internal heat transfer coefficient has to be revisited to prevent modeling biases and overestimation of the wall/fluid heat flow rate, and should be replaced ideally by a more legitimate transfer function-based model [6].

2. The studied system and its modelling

Let us consider the following system (Fig. 1): a liquid flows in a flat channel of length 2ℓ and of thickness e_f, which is defined by two parallel plates of thicknesses e_1 and e_2. A developed velocity profile $u(y)$ and a temperature T_∞ are imposed at the entrance of the channel. Two uniform surface heat sources (q_{hot} and q_{cold}) are imposed on a portion $\ell_h = \ell_c = x_2 - x_1 = x_4 - x_3$ (see Fig. 4) of the external faces. These external faces are subject to convective losses to the ambient environment at temperature T_∞ (uniform h coefficient), see Fig. 1.

The two solid plates are characterized by a thermal conductivity λ_s. The fluid is characterized by a conductivity λ_f, a volumetric heat capacity ρc_f and a kinematic viscosity ν_f, and $a_f = \frac{\lambda_f}{\rho c_f}$ is its thermal diffusivity.

2.1. Analytical model

The equations describing heat transfer in the mini-channel and in the adjacent parallel walls with the corresponding boundary conditions are given below:

- Heat equation in the walls:

$$\frac{\partial^2 T_s}{\partial x^2} + \frac{\partial^2 T_s}{\partial y^2} = 0 \tag{1}$$

- Heat equation in the fluid:

$$\left(\frac{\partial^2 T_f}{\partial x^2} + \frac{\partial^2 T_f}{\partial y^2}\right) - \frac{u(y)}{a_f}\frac{\partial T_f}{\partial x} = 0 \tag{2}$$

- Transverse boundary conditions on the external faces, where φ denotes the heat flux in the y direction and $\mathcal{H}$ the Heaviside step function:
  - at $y = -e_f/2 - e_1$:

$$\varphi_h = -\lambda_s \frac{\partial T}{\partial y} = q_h(x) - h(T - T_\infty) \tag{3}$$

  where $q_h(x) = q_{hot}[\mathcal{H}(x - x_1) - \mathcal{H}(x - x_2)]$
  - at $y = +e_f/2 + e_2$

$$\varphi_c = -\lambda_s \frac{\partial T}{\partial y} = q_c(x) + h(T - T_\infty) \tag{4}$$

  where $q_c(x) = q_{cold}[\mathcal{H}(x - x_3) - \mathcal{H}(x - x_4)]$
- and the solid/fluid interface conditions at $y = \pm e_f/2$ are:

$$-\lambda_s \frac{\partial T_s}{\partial y} = -\lambda_f \frac{\partial T_f}{\partial y} \quad \text{and} \quad T_s = T_f \tag{5}$$

- In real applications, the lateral boundary conditions in both the fluid and the walls at the ends of the channel, that is for $x = -\ell$ (entrance) and $x = +\ell$ (exit) are not known. In particular assuming adiabatic conditions is never realistic for a steady state case. So the system will be modelled for $x \in]-\infty, +\infty[$, with the same uniform heat transfer coefficient on this infinite domain. Since external surface excitation (q_h and q_c) only occurs on the