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Experimental analysis of the evaporation of a thin liquid film deposited on a capillary heated tube: estimation of the local heat transfer coefficient

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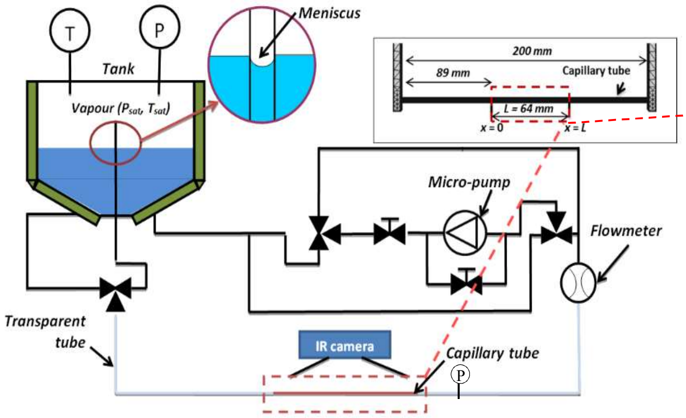
Abstract. The aim of this work is to estimate the local heat flux and heat transfer coefficient for the case of evaporation of thin liquid film deposited on capillary heated channel: it plays a fundamental role in the two-phase heat transfer processes inside mini-channels. In the present analysis it is investigated a semi-infinite slug flow (one liquid slug followed by one single vapour bubble) in a heated capillary copper tube. The estimation procedure here adopted is based on the solution of the inverse heat conduction problem within the wall domain adopting, as input data, the temperature field on the external tube wall acquired by means of infrared thermography.

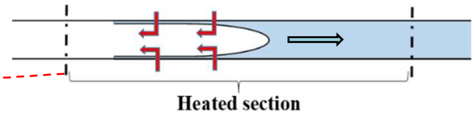
1. Introduction

One of the main current issues of electronic components thermal management, considering the rapid and constant miniaturization of chips and the continuous growth of their dissipated power, requires innovative technologies that guarantee the coupling between high thermal performances and reduced dimensions [1]. A high potential response is represented by two-phase heat transfer inside micro and mini channels: these latter fulfil requirements for heat power increase and reduce operational and manufacturing costs while having a significantly positive environmental impact by minimizing the amount of material and/or refrigerant fluid in systems [2]. A crucial role in the two-phase heat transfer processes inside mini/micro-channels is played by evaporation of thin liquid films in bubbles: between liquid and vapour phases the meniscus deposits a thin liquid film at the wall; this film brings about intense heat transfers by evaporation, leading to a highly significant decrease of external wall temperature [3]. In the present work the local heat flux and the local heat transfer coefficient (HTC) connected to the film evaporation in a heated capillary tube are investigated by studying a semi-infinite slug flow (one liquid slug followed by one single vapour bubble) in a heated capillary copper tube. One of the most promising way to estimate the local heat fluxes on the inner surface of a tube is found by solving the Inverse Heat Conduction Problem (IHCP) in the solid domain starting from the temperature distribution acquired on the exterior wall surface [4]. However, because IHCPs are generally ill-posed, the solution may not be unique and would have great sensitivity to small variations in the input data. To cope with this difficulty, many techniques have been proposed and in the present work, the Tikhonov regularization method [4] is adopted: it is applied to experimental infrared temperature data to estimate the heat flux of evaporation and HTC inside the studied capillary channel.

2. Experimental Setup

The experimental device (figure 1) consists of a capillary copper tube (200 mm long and 2 mm-inner- and 2.4 mm -outer- diameters), heated by Joule effect thanks to an electric power supply (XANTREX XDC-20-300). An infrared camera (FLIR SC7200) is disposed above the tube to measure the outside wall temperature field of the channel, covered with a black paint with an emissivity value of 0.95 ±0.01. The camera wavelength band ranges from 1.5 to 5.1 μm (accuracy: ± 1 K in absolute temperature, thermal sensitivity < 25 mK and frame rate up to 7700 Hz). The sample window range is 64 mm long, made up of 320 pixels length and 12 pixels width. The adopted liquid is water, and its flow is ensured by a volumetric pump (MICROPUMP GA X21) from the two-phase tank, large enough to avoid any modification of the thermodynamic state of vapour leaving it. The tank is instrumented by a pressure transducer (UNIK 5000 GE, ±80 Pa) and a 4-wires Platinum sensor (± 0.1 K) located on its upper part, in the vapour phase. The saturation state of the fluid can be checked using fluids liquid/vapor equilibrium diagram. To ensure the accuracy of this estimate an additional pressure transducer has been set at the upstream location of the heated section. It allows direct measurement of vapor pressure downstream of the meniscus. The overall loop, including lines and reservoir, is hermetically sealed to remain at saturation state during tests. Thus, the fluid properties have been evaluated at the saturation temperature. Starting from a pure single-phase liquid flow from the reservoir (at imposed temperature), the three-way valve located above it permits the transition to a liquid slug flow, ended by a meniscus, and followed by an infinite vapour bubble. The flow in a capillary tube at imposed velocity of a meniscus separating an upstream liquid plug from a vapour bubble, both of semi-infinite lengths, is thus established.





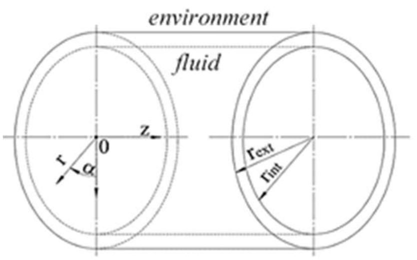


Figure 1. Schematic view of the experimental set-up [3]

Figure 2. Geometrical domain with coordinate system

3. Estimation Procedure

The physical problem is represented by the heat conduction in the wall of a cylindrical pipe. The test section shown in figure 2 was modelled as a 2-D axisymmetric solid domain since the temperature gradient is almost negligible along the circumferential coordinate [3]. Under these conditions the energy balance equation in the solid domain is expressed as follows:

$$k\nabla^2 T + q_g = \rho c_p \frac{\partial T}{\partial t} \tag{1}$$

with k, ρ and c_p the tube thermal conductivity, density and specific heat, and q_g the volume heat. The following two boundary conditions, applied on the internal surface ($k\,\partial T/\partial r = q$) and on the external surface (($k\,\partial T/\partial r = -(T - T_{env})/R_{env}$) of the pipe, complete the energy balance equation. The heat conduction problem, according to [4], can be written in the discrete domain as follows:

$$\mathbf{T} = \mathbf{X}\mathbf{q} + \mathbf{T}_{\mathbf{q=0}} \tag{2}$$

with **T** the vector of discrete temperature data at the external surface, **q** the heat flux vector at the fluid-internal wall interface and $\mathbf{T}_{\mathbf{q=0}}$ a constant term determined by imposing a null heat flux. The sensitivity matrix **X** and the term $\mathbf{T}_{\mathbf{q=0}}$, for the problem under test, can be calculated numerically [4]. The **T** and **q** vectors can be written as:

$$\mathbf{T}=\begin{bmatrix}\mathbf{T(1)}\\ \mathbf{T(2)}\\ \vdots\\ \mathbf{T(N)}\end{bmatrix},\qquad \mathbf{T}(\mathrm{i})=\begin{bmatrix}\mathrm{T}_1(\mathrm{i})\\ \mathrm{T}_2(\mathrm{i})\\ \vdots\\ \mathrm{T}_J(\mathrm{i})\end{bmatrix};\qquad \mathbf{q}=\begin{bmatrix}\mathbf{q}(1)\\ \mathbf{q}(2)\\ \vdots\\ \mathbf{q}(\mathrm{N})\end{bmatrix},\qquad \mathbf{q}(\mathrm{i})=\begin{bmatrix}\mathrm{q}_1(\mathrm{i})\\ \mathrm{q}_2(\mathrm{i})\\ \vdots\\ \mathrm{q}_K(\mathrm{i})\end{bmatrix} \tag{3}$$

where *i* refers to the time step and *N* is the total number of time steps. The elements of vector **T**(i) are the external wall temperatures at different *j* sensor location (1, 2,…*J*) along the axial coordinate at the i^{th} time step. Analogously, the components of vector **q**(i) are defined at different *k* locations (1, 2,…,*K*) along the axial coordinate at the i^{th} time step. The sensitivity matrix is expressed as:

$$\mathbf{X}=\begin{bmatrix}\mathbf{x}(1) & 0 & 0 & \cdots & 0\\ \mathbf{x}(2) & \mathbf{x}(1) & 0 & \cdots & 0\\ \mathbf{x}(3) & \mathbf{x}(2) & \mathbf{x}(1) & \cdots & \vdots\\ \vdots & \vdots & \vdots & & 0\\ \mathbf{x}(\mathrm{N}) & \mathbf{x}(\mathrm{N-1}) & & \cdots & \mathbf{x}(1)\end{bmatrix},\qquad \mathbf{x}(\mathrm{i})=\begin{bmatrix}\mathrm{x}_{11}(\mathrm{i}) & \mathrm{x}_{12}(\mathrm{i}) & \cdots & & \mathrm{x}_{1K}(\mathrm{i})\\ \mathrm{x}_{21}(\mathrm{i}) & \mathrm{x}_{22}(\mathrm{i}) & \cdots & & \mathrm{x}_{2K}(\mathrm{i})\\ \mathrm{x}_{31}(\mathrm{i}) & \mathrm{x}_{32}(\mathrm{i}) & \ddots & \cdots & \mathrm{x}_{3K}(\mathrm{i})\\ \vdots & \vdots & \mathrm{x}_{jk}(\mathrm{i}) & & \vdots\\ & & & \ddots & \\ \mathrm{x}_{J1}(\mathrm{i}) & \mathrm{x}_{J2}(\mathrm{i}) & \cdots & \cdots & \mathrm{x}_{JK}(\mathrm{i})\end{bmatrix},\qquad \mathrm{x}_{jk}(\mathrm{i})=\frac{\partial \mathrm{T}\left(\mathrm{x}_j,\mathrm{t}_i\right)}{\partial \mathrm{q}_k(\mathrm{t}_1)} \tag{4}$$

In the inverse formulation, the computed temperature distribution **T** (Eq. 2) is forced to match the experimental temperature distribution by tuning the convective heat-flux distribution on the internal wall side **q**. The matching of the two temperature distributions (the computed and the experimentally acquired) could be easily performed under a least square approach. However, due to the ill-posed nature of the problem, the least square solution is ineffective because it is dominated by noise, and some type of regularisation is required. As mentioned before, the Tikhonov regularisation method was adopted [4].

4. Experimental results

The estimation procedure described above was then applied to the IR acquisitions. In figure 3 (left) it is reported the experimental temperature acquired by the infrared camera on the external wall of the copper capillary tube that was used as input data in the solution of the IHCP in the tube wall. It was considered the temperature distribution of the whole pipe length shot by the infrared camera for a time interval of 1 s corresponding to the passage of the meniscus. The local heat flux distribution restored by solving the inverse heat conduction problem is reported in figure 3 (right). The passage of the meniscus is clearly noticeable, corresponding to the sharp decrease of temperature between the left part (liquid flow) and the right part (thin liquid two-phase flow). In correspondence of the passage of the meniscus there is a sudden increase of the heat flux, due to the evaporation of the film entrapped between the wall and the vapor bubble.

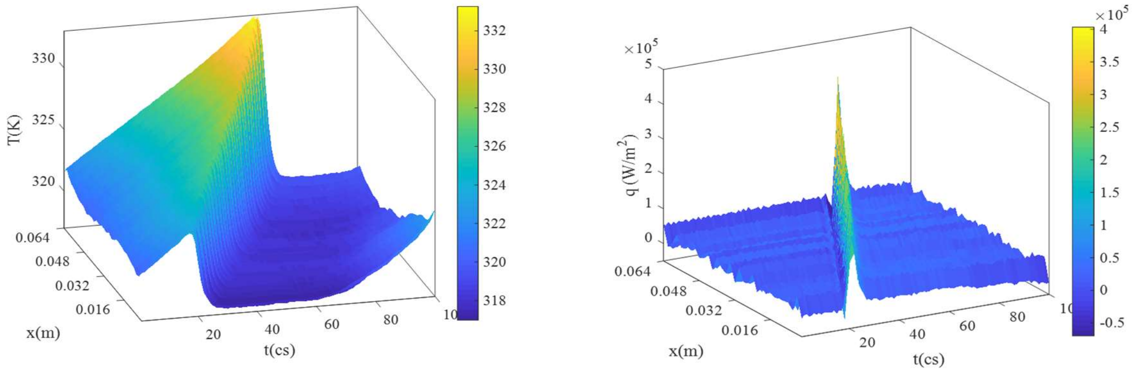


Figure 3. Experimental temperature (left) and the corresponding restored local heat flux distribution (right).

Once the heat flux it is restored it is possible to compute the local HTC distribution dividing the heat flux by the difference between the temperature at the internal wall of the pipe, obtained by solving the direct problem imposing the heat flux shown in figure 3b at the internal wall to fluid interface, and the fluid bulk temperature that was assumed equal to the saturation temperature at the reservoir (i.e. T_b = 44.25°C). In figure 4 it is reported the computed HTC distribution where in correspondence of the passage of the meniscus a sharp increase of the HTC distribution is noticeable: in this area HTC is at least one order of magnitude higher than that corresponding to pure liquid flow.

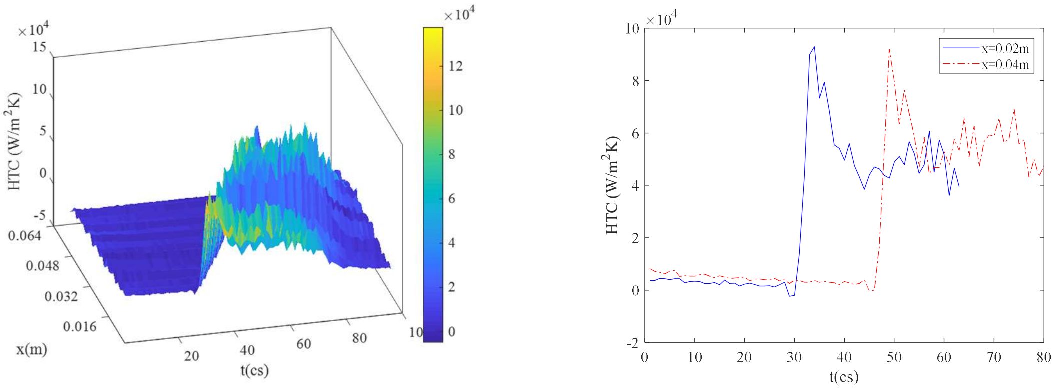


Figure 4. HTC distribution (left) and function of time for different axial coordinates (right).

This zone is followed by a heat transfer time governed by evaporation in the deposited thin liquid film on the wall where a very high HTC and a quasi-constant temperature distribution can be observed (figure 3 left). Finally, it is found that the local HTC can reach values up to 0.1 MW/m^{2}K during the evaporation of the thin liquid film deposited at the channel wall. The results are in good agreement with the outcomes of Thome et al. [5] that investigated evaporation in microchannels.

5. Conclusions

In the present study an inverse analysis technique is applied to experimental infrared temperature data to estimate the local heat flux and HTC for the case of evaporation of liquid film deposited on capillary heated channel. A semi-infinite slug flow (one liquid slug followed by one single vapor bubble) in a heated capillary copper tube has been considered. In correspondence of the passage of meniscus it can be noticed a sharp decrease of temperature corresponding to a sudden increase of the heat transfer, due to thin film evaporation. In this area the HTC is at least one order of magnitude higher than in the area corresponding to pure liquid flow. This zone is followed by a heat transfer time governed by evaporation in the deposited thin liquid film on the wall where very high HTC and a nearly constant temperature distribution can be observed. Moreover, another important outcome of this paper is the proposed approach that will be successfully applied to many types of two-phase heat transfer devices to achieve optimal thermal processes.

Acknowledgments

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