International Journal of Heat and Mass Transfer 89 (2015) 454–459

Contents lists available at ScienceDirect

International Journal of Heat and Mass Transfer

journal homepage: www.elsevier.com/locate/ijhmt

Numerical and analytic study on the time-of-flight thermal flow sensor

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ARTICLE INFO

Article history:
Received 31 December 2014
Received in revised form 4 May 2015
Accepted 4 May 2015

Keywords:
Time-of-flight (TOF)
Flow sensor
Finite Volume Method (FVM)

ABSTRACT

In this study, a numerical study on the time shift of time-of-flight (TOF) thermal flow sensor is performed based on Finite Volume Method (FVM). The TOF is modeled as parallel cylinders, and the governing equations for transient fluid flow and heat transfer, as well as boundary conditions, are established. In order to provide a physical understanding, a simple analytic model is also developed, and an excellent agreement is shown between numerical and analytic results. Based on the validated simulation results, the effects of wire diameter, flow velocity, and fluid Prandtl number on the time shift is investigated. The results show that the convection time shift becomes negligible as either the wire diameter or the flow velocity decreases. The effect of the wire diameter on the time shift becomes insignificant as the wire diameter decreases beyond several tens of micrometers. The time shift is shown to increase as the Prandtl number of fluid increases. The suggested computational scheme, together with the simple analytic model, can be effectively utilized for calibrating and designing the TOF sensors.

1. Introduction

The measurement and control of gas flow rates is critical in many engineering applications, including semiconductor manufacturing processes and chemical processes [1]. Thermal flow sensors are most widely used for measuring mass flow rates in the semiconductor industry [2]. This type of sensors utilizes the convective heat loss for the flow rate measurement. Hot-wire and hot-film anemometers are typical representatives. They are usually operated under constant temperature or constant current conditions and are also employed for flow-rate measurement in a wide range of applications [3]. There is another subclass of thermal sensors, utilizing 'time-of-flight measurements', which yield velocity information from which mass flow rate can be deduced. Sensors of this kind consist of a flow channel, a sending wire and a receiving wire. The schematic diagram of that kind of sensor is illustrated in Fig. 1. The two wires are placed in the flow channel, and mounted perpendicular to the flow. The sending wire is heated electrically to provide the flow with a time-varying thermal signal. This resultant thermal signal is convected downstream to the receiving wire which determines the delayed arrival of the thermal signal. The flow velocity can be computed from the measured time shift, or 'time of flight.' This type of flow measuring has been reported to be an appropriate solution to the problem of measuring small fluid flow. It can give a high accuracy and its linearity is excellent [4–6].

It is important to find how the time-of-flight (TOF) is correlated with the parameters of the actual flow cell in order to perform a calibration. The time-of-flight correlation for the flow velocity is often regarded as a transportation delay of l/U, where l is distance between wires and U is flow velocity. However, the actual time shift may deviate much from the above idealized value as shown by many investigators [7,8] due to the thermal time response of the finite-diameter sending and receiving wires. Additionally, Eaton et al. [9] found that the thermal diffusion is influenced by local flow conditions and fluid properties, and therefore a new calibration is required if a fluid with high thermal diffusivity is used. Bradbury [7] suggested an empirical fit to the calibration taking the thermal diffusivity into account and explained why the non-linearity in the calibration appears. Handford and Bradshaw [8] showed that the flow velocity will be equal to the distance between the sending and receiving wires divided by the measured time of flight plus a second term representing the various non-ideal effects. More recently, Yang et al. [4,5] solved an energy equation for heat transfer analytically under the assumption of sinusoidal change in temperature of the receiving wire. They concluded that the dynamic response of the receiving wire is decreases with increasing the frequency of temperature variation at the sending wire. They explained that the difference between the time of flight and the reciprocal flow velocity is mainly due to the time constant of the receiving wire and the sending wire and also due to the thermal diffusion. The same problem is solved more rigorously using numerical method by Durst et al. [3,10,11]. They carried out an extensive research including parametric studies for the purpose

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http://dx.doi.org/10.1016/j.ijheatmasstransfer.2015.05.011
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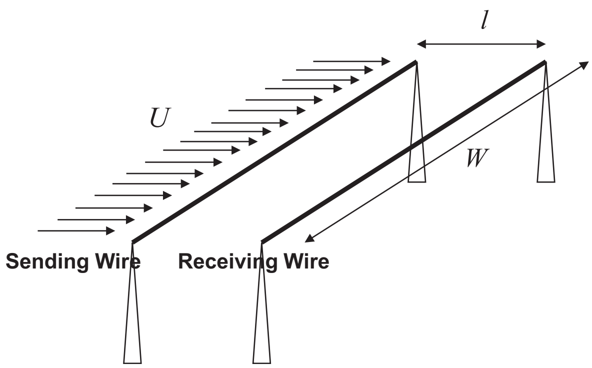

Fig. 1. Schematic diagram of time-of-flight sensor.

of large sensor operating range. There have been research studies on the TOF. Models proposed by recent studies account for the effect of thermal time constant of wires and thermal diffusion and predict the TOF accurately. However, there is no study on how the TOF is correlated with the parameters of the actual flow cell. Research of Durst [3,10,11] is concentrated on confirming a wide operating range for the sensors, includes no parametric study. In order to perform a calibration for flow sensor correctly under various environments, the effect of wire diameter, the flow velocity and the fluid property on the time shift of TOF sensor should be accurately determined.

In this study, a parametric study on the time shift of TOF is performed based on Finite Volume Method (FVM). The TOF is modeled as parallel cylinders, and the governing equations for transient fluid flow and heat transfer, as well as boundary conditions, are established. In order to provide a physical understanding, a simple analytic model is also developed. Based on the validated simulation results, the effects of wire diameter, flow velocity, and fluid Prandtl number on the time shift is investigated.

2. Numerical simulation

In the present study, an incompressible and be two dimensional flow is assumed, because the ratio of wire diameter to wire length is so small that the rheological properties do not vary along the direction of wire length. Additionally, the Reynolds number of concern is so small that the three dimensional effects such as vortex shedding can be neglected [3,7].

A schematic diagram of the physical problem under consideration is shown in Fig. 1. A sending wire and a receiving wire with length W are aligned parallel to each other with a distance l. A uniform velocity U is applied normal to the wire as shown in this figure. The temperature of the sending wire is heated and controlled so that it has a sinusoidal variation with respect to time. The temperature of the fluid increases as it passes through the sending wire. The heated fluid flows rightward and transports the thermal energy to the receiving wire. The average temperature of the receiving wire is tracked as a function of time.

The governing equations for this numerical problem consist of: (i) continuity equation; (ii) Navier–Stokes equation; (iii) energy equation. The continuity equation for fluid used for the present numerical study is as follows [12,13]:

$$\nabla \cdot \vec{u} = 0 \tag{1}$$

The Navier–Stokes equation is as follows:

$$\rho \frac{\partial \vec{u}}{\partial t} + \rho(\vec{u} \cdot \nabla)\vec{u} = -\nabla p + \nabla \cdot \tau \tag{2}$$

where p is the static pressure, τ is the stress tensor, respectively. The stress tensor τ is given as follows:

$$\tau = \mu\left[(\nabla\vec{u} + \nabla\vec{u}^T) - \frac{2}{3}\nabla \cdot \vec{u}I\right] \tag{3}$$

where μ is the fluid viscosity, I is the unit tensor, and the second term on the right hand is the effect of volume dilation.

The energy equation is used for both fluid and solid (wire) regions. In fluid regions, the energy equations are as follows:

$$\frac{\partial T_f}{\partial t} + (\vec{u} \cdot \nabla)T_f = \frac{\nu}{\Pr}\nabla \cdot (\nabla T_f) + \frac{\alpha_f \mu}{k_f}\Phi \tag{4}$$

In Eq. (4), the viscous dissipation function Φ is given as follows:

$$\Phi = 2\left[\left(\frac{\partial u}{\partial x}\right)^2 + \left(\frac{\partial v}{\partial y}\right)^2\right] + \left(\frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}\right)^2 - \frac{2}{3}\left(\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y}\right)^2 \tag{5}$$

The energy equation for the solid regions is as the following:

$$\frac{\partial T_s}{\partial t} = \alpha_s \nabla \cdot (\nabla T_f) \tag{6}$$

The computational domain and the boundary conditions for the problem are shown in Fig. 2. At the inlet, the inflow velocity is defined for flow, and the temperature is set to be the ambient temperature (Dirichlet boundary condition for temperature). At the outlet, the outflow static pressure is defined and the backflow total temperature is set to be the ambient temperature. Two side walls are set as adiabatic (Neumann boundary condition for temperature), impermeable, and no-slip boundaries. The sending and receiving wires are modeled as cylinders. On the wire surface, no-slip boundary condition for the velocity, and no-jump condition for the temperature are imposed. For the sending wire, the surface temperature is set to be uniform. The temperature of the sending wire is varied as a sinusoidal function of time as the following equation:

$$T(t) = T_0 + \Delta T \sin(2\pi f t) \tag{7}$$

The ambient temperature was set to be 300 K and the amplitude of oscillation is set to be 50 K. The frequency is set to be 10 Hz in this study. The size of the computational domain is selected to be large enough so that the existence of adiabatic sidewalls does not affect the temperature distribution around the wire. The discretization of the rectangular domain was performed by coupling a pave type quadrangle grid inside the cylinder with a map type quadrangle grid outside the cylinder.

The grid distribution is plotted in Fig. 3(a). The grid points are concentrated around the cylinder and in the wake region. A better view of the grid distribution near the cylinder is shown in the detail in Fig. 3(b). For Reynolds numbers larger than about five, a pair of standing vortices appears at the rear of the cylinder [10], which demands a finer discretization of this region. The configuration of grid near the cylinder was chosen based on extensive numerical tests and was then employed as the core region of all numerical grids for the computations. The surface of the cylinder was discretized with more than 200 segments as shown in Fig. 3(b). To catch the effect of thermal time-lag at the receiving wire, each wire was discretized with maximum 1892 CVs, as shown in Fig. 3(c). The computational domain does not include any non-conformal grid, which may result in errors and more computational time [14]. The grid sensitivity of the described model is tested by first running with 14,498 elements and then comparing the results with the results of doubled elements number. The resulting receiving wire temperature changed by 0.5 K. Based on the selection of first-order shape functions, it was estimated that further grid refinement beyond 14,498 cells would only change wall temperature by 1 K, a value which would only influence the second significant figure of the results. The time step was selected after extensive numerical test in order to confirm the stability of solution. Each time step, the criteria for convergence were set to