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A simple method for predicting bulk temperature from tube wall temperature with uniform outside wall heat flux☆

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ABSTRACT

An empirical approach is proposed to estimate the bulk temperature in practical laminar tube flow. To examine the correlation, heat transfer in different types of tubes with wall conduction and uniform constant heat flux at tube outer wall surface is numerically investigated. The predictions from the proposed correlation match well with the numerical results in all the cases studied for air flow in the Pe range from 105 to 1032 and for water flow in the Pe range from 70 to 700. The method is further testified via comparison with experimental data and numerical results of mini (micro)-channel water flow available in literature.

1. Introduction

Extensive studies on conjugate heat transfer in a tube with internal laminar flow have been performed using analytical and numerical methods in the past three decades. Mori et al. [1] investigated the wall conduction effects in a circular tube with either uniform constant heat flux or constant temperature at the outer surface of the tube. Faghri and Sparrow [2] numerically studied the coupled effect between solid wall and fluid axial conductions. It was found that the effect of the wall axial conduction can readily overwhelm the effect of the fluid axial conduction. The generalized conjugate Graetz problem with axial conduction was studied in Refs. [3–6]. All of the above-mentioned literature obtained temperature profiles assuming a known and temperature-independent velocity profile. As a matter of fact, the real velocity profile is not known and both the velocity and temperature fields are simultaneously developing at the entrance regime in most cases.

Experimentally the tube outer wall temperature can be easily measured whereas it is impossible to measure the local fluid bulk temperature. It would be very useful if the bulk temperatures can be acquired via the measured wall temperatures in order to calculate heat transfer coefficients. To this end, a simple method oriented from the experimental point of view is presented in this paper which avoids the complexity of the afore-mentioned analytical and numerical approaches. Air is considered as the working fluid, with Peclet number varying from 105 to 1032. In addition, the proposed method for predicting water bulk temperature is verified through comparison with published experimental data and numerical results. Meanwhile, two common tube materials (copper and stainless steel) are also considered to extend the flexibility of the method.

2. Physical model and proposed method

The physical model is a circular tube with finite wall thickness shown in Fig. 1, whose inner diameter is D_i and the ratio of length to inner diameter (L/D_i) is set to 100 in order to reach fully-developed state. The conjugate heat transfer problem is analyzed in laminar flow condition. When air is considered as the working fluid, natural convection is negligible. The laminar flow is characterized by steady-state, constant physical property and three-dimensional situations with no viscous dissipation. The general governing equations are omitted here due to space limitation.

A mass-flow-inlet is set as the inlet boundary condition and a pressure-outlet is set as the outlet boundary condition. The ambient fluid temperature (T_e) is 283.15 K for air (Pr = 0.7). A uniform constant heat flux boundary condition is imposed on the outer surface of the tube wall (q_{wo} = const) and the adiabatic boundary condition is imposed on both ends of the tube wall. The numerical models are solved by commercially available computer CFD software (FLUENT 6.3).

It is well known that, under the boundary condition of uniform constant heat flux at the circular tube without consideration of wall thickness, the fluid temperature distribution is linearly increased along the streamwise direction [7]. At the fully-developed region the temperature difference between the wall temperature and the fluid bulk temperature ($T_w - T_f$) is a constant.

In fact, any circular tube comes with a finite wall thickness. In experiments, the outer-wall temperature (T_{wo}) of a tube can be measured by thermocouples; and the inner-wall temperature (T_{wi}) is obtained via one-dimensional Fourier heat conduction analysis. The

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Nomenclature

c	coefficient in Eq. (2a)
d	coefficient in Eq. (2a)
D_h	hydraulic diameter (m)
D_i	inner diameter of outer-tube (m)
L	tube length (m)
q	wall heat flux (W/m^2)
Pe	Peclet number (=RePr)
Pr	Prandtl number
Re	Reynolds number
T	temperature (K)
x^*	dimensionless axial position ($4x/D_h$Pe)
x	axial position (m)
x, y, z	Cartesian coordinates

Greek symbols

λ	thermal conductivity (W/m K)
δ	thickness (m)
ε	error indicator
θ	dimensionless temperature
β	auxiliary function, Eq. (2a)

Subscripts

in	inlet
e	ambient fluid temperature
f	fluid
L	tube end
m	mean
max	maximum
out	outlet
w	wall
wi	inside wall
wo	outside wall
x	along x direction, local
0	tube entrance
1	results based on Eq.(5)
2	results based on Eq.(6)
3	results based on numerical results
*	dimensionless

inner-wall temperature (T_{wi}) may be assumed of the outer-wall temperature (T_{wo}) for tubes of high thermal conductivity or thin-wall tubes. The ambient fluid temperature (T_e) could be measured by Mercury thermometer or other measuring instruments. Usually, the fluid outlet temperature (T_{out}) of a test tube is measured. The inlet temperature of the fluid is approximately equal to the ambient fluid temperature ($T_{in}=T_f(0)=T_e$). Such consideration is correct in laminar tube flow without wall heat conduction or in turbulent flow [8,9]; but not appropriate for lower Reynolds number laminar flow because of the existence of axial wall heat conduction. If the inlet fluid temperature is to be measured, there exist five known parameters, namely, T_e, $T_f(0)$ ($=T_{in}$), $T_f(L)$ ($=T_{out}$), $T_{wi}(0)$ and $T_{wi}(L)$; then a linear correlation for the dimensionless temperature, θ , is obtainable in laminar tube flow. The dimensionless temperature,$\theta(x)$, is defined as

$$\theta(x) = (T_{wi}(x) - T_f(x) / (T_{wi}(x) - T_e) \tag{1}$$

where T_{wi} is the inner-wall temperature of the tube wall, T_e is the ambient fluid temperature, and T_f is the bulk temperature.

For an ideal circular tube with negligible wall thickness, the reciprocal of θ can be expressed as a linear function of the axial coordinate in the absence of axial wall heat conduction and $\theta(0)=1$ because of $T_f(0)=T_{in}=T_e$. Fig. 2a shows the numerical results of the variation of $(1/\theta)$ versus $x^*=4x/D_h/\text{Pe}$. As shown clearly in Fig. 2a, the distributions of $(1/\theta)$ vs. x^* are nearly linear for air (Pe = 105 and 1032).

For an actual circular tube with finite wall thickness, however, the fluid will be preheated ($T_{in}>T_e$) before entering into the heated tube because of the axial wall conduction. So the dimensionless temperature $\theta(0)$ at $x=0$ is less than 1 ($\theta(0)<1$), and the difference of $(1-\theta(0))$ could be congenially regarded to represent the axial heat transfer rate. The numerical results of the variation of $1/(\theta(x)+1-\theta(0))$ are illustrated in Fig. 2b and c, from which a quasi-linear equation can be obtained. It is seen that the linearity of $1/(\theta(x)+1-\theta(0))$ for a circular tube with finite wall thickness is quite good at low thermal conductivity and high Peclet number but poor at high thermal conductivity and low Peclet number. However, the error for predicting fluid bulk temperature is satisfactory in terms of engineering application as described in the next section because $\theta(x)$ is a decimal between 0 and 1.

Now the quasi-linear equation for a circular tube with axial wall heat conduction is given as

$$\beta(x) = \frac{1}{\theta(x) + (1-\theta(0))} = c + dx^* \tag{2a}$$

at

$$x = 0, x^* = 0 : \theta(0) = (T_{wi}(0) - T_{in}) / (T_{wi}(0) - T_e), \beta(0) = 1 \tag{2b}$$

at

$$x = L, x^* = \frac{4L/D_h}{Pe} : \quad \theta(L) = (T_{wi}(L) - T_{out}) / (T_{wi}(L) - T_e), \beta(L) = \frac{1}{\theta(L) + (1-\theta(0))} \tag{2c}$$

For a circular tube, the hydraulic diameter is $D_h=D_i$.

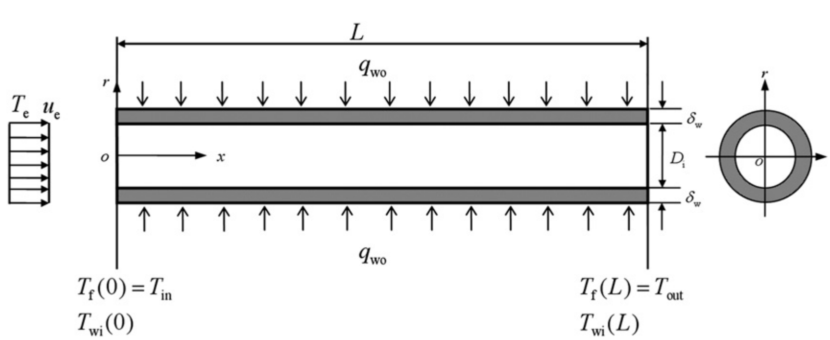

Fig. 1. Sketch of a circular tube with specified thermal boundary conditions.