Heat Transfer Investigation of Air Flow in Microtubes— Part II: Scale and Axial Conduction Effects

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In this paper, the scale effects are specifically addressed by conducting experiments with air flow in different microtubes. Three stainless steel tubes of 962, 308, and 83 μm inner diameter (ID) are investigated for friction factor, and the first two are investigated for heat transfer. Viscous heating effects are studied in the laminar as well as turbulent flow regimes by varying the air flow rate. The axial conduction effects in microtubes are experimentally explored for the first time by comparing the heat transfer in SS304 tube with a 910 μm ID/2005 μm outer diameter nickel tube specifically fabricated using an electrodeposition technique. After carefully accounting for the variable heat losses along the tube length, it is seen that the viscous heating and the axial conduction effects become more important at microscale and the present models are able to predict these effects accurately. It is concluded that neglecting these effects is the main source of discrepancies in the data reported in the earlier literature. [DOI: 10.1115/1.4007877]

Keywords: scale effect, axial conduction, heat transfer coefficient, microtubes, nickel tubes, heat loss, viscous heating, axial conduction

1 Introduction

Microchannels are being used increasingly in microelectromechanical systems (MEMS) where heat transfer constitutes an important process step in many devices. The high heat transfer coefficient and low thermal resistance of these channels also make them attractive in compact heat exchanger applications. In order to design these devices for reliable operation, it is essential to understand the flow and heat transfer characteristics in microchannels. The heat transfer data for gas flow in microchannels reported by earlier researchers were found to deviate considerably from the established heat transfer correlations. Although deviations in the friction factor data for the microchannels are now understood as being attributable to experimental uncertainties, incorrect estimation of boundary conditions, and entrance region effects, the heat transfer data has not yet been clearly reconciled. In general, an effect of Reynolds number (Re) on Nusselt number (Nu) in the laminar flow region has been reported in previous literature. This could stem from a lack of understanding of how heat transfer is affected by parameters that are usually negligible at macroscale, such as roughness, entrance region effect (especially in turbulent flows in tubes with large length to diameter ratio), heat losses, axial heat conduction, viscous heating, and rarefaction effects.

The flow characteristics and heat transfer performance in microchannels has been investigated by a few researchers who reported that their experimental data departed from conventional macroscale correlations [1–7]. In the work reported by Yu et al. [1], the friction factor data of water and gas flow in 52–102 μm channels were found to be lower than the predictions from conventional correlations. However, later publications, e.g., Judy et al. [8], have indicated that the data for water flow in microchannels was in good agreement with conventional correlations. Error in channel diameter measurement was noted as one of the main

reasons for such deviations in the earlier water data. As to be expected, it has been experimentally verified that there are no microscale effects on friction factors for liquid flow in microchannels [9–12].

Gas flow with nitrogen in microtubes ranging in diameter from 30 to 500 μm and having surface roughness values $<1\%$ has been experimentally investigated by Celata et al. [13]. Their results showed that the conventional correlations are able to predict the friction factor in laminar flow for these tubes. In the fully developed turbulent regime, good agreement was found between their experimental data for smooth tubes and the Blasius equation, while the data for rough tubes agreed well with the Colebrook equation. Some of the recent studies also indicated that friction factors for liquid flows in smooth channels with diameters higher than 15 μm and heat transfer coefficients in channels with diameters higher than 123 μm can be predicted well by conventional correlations [8–11,14]. The difficulties associated with heat transfer measurements at microscale are discussed by Guo and Li [15].

In gas flow, the friction factor was found to be in good agreement with the predictions from conventional correlations with diameters larger than 4.7 μm in laminar flow and 100 μm in turbulent flow [16,17]. However, there is no reliable experimental data available for heat transfer with gas flow at microscale which account for viscous heating effect, axial conduction effect and heat losses. Further discussion on available literature is presented in Part I of the present study [18].

The objectives of this study are to generate accurate experimental data for friction factor and heat transfer at microscale and systematically study the scale effects by varying the tube diameter, flow velocity, tube material, and tube wall thickness. The effects of viscous heating and axial conduction will be investigated to identify their influence at microscale.

2 Experimental Methods

The experimental gas flow loop consisted of an air bottle, a regulator, and three flow meters with different working ranges. The

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Fig. 1 Schematic of the test section with LCT temperature measurement

air flow rate was adjusted with a regulator and measured by the flow meters. The temperature and pressure drop of the air flow through the microtubes was measured with a resistance temperature detector and pressure transducer installed at the inlet of the test section. The test section was an open system with the outlet air discharged to the atmosphere. The outer wall temperatures of the tube were measured by fine wire 36 gauge type E thermocouples placed uniformly at three locations along the tube length. The tube was directly heated by a high current dc power supply. The whole test section was enclosed in a vacuum environment to minimize the heat losses. The whole test system is very similar to that in Part I of the present study [18]. However, in addition to the fine wire thermocouples, the wall temperature of the microtubes in the present study was also measured by liquid crystal thermography (LCT). The microtubes were coated with thermochromic liquid crystal (TLC) and imaged with a CCD camera to determine the wall temperature. A transparent window was installed on the vacuum chamber for optical access for the CCD camera as shown in Fig. 1. Details of the temperature measurement technique using LCT in

microtubes can be found in Lin and Yang [9]. Flow meters, thermocouples, pressure transducer, and power supply were interfaced with a LabVIEW program for data acquisition.

To investigate the effect of axial heat conduction in the tube wall, a tube with thick walls (548 μm thickness) and high thermal conductivity (nickel) was manufactured. Figure 2 shows the cross section of the microtubes used in this study. The internal and external diameters of the stainless steel tube shown in Fig. 2(a) are 962 μm and 1260 μm , and the corresponding values for the nickel tube shown in Fig. 2(b) are 910 μm and 2005 μm , respectively. Another tube used to investigate axial conduction effect is a smaller diameter tube shown in Fig. 2(c). The internal and external diameters are 308 μm and 579 μm , respectively. There was also an 83 μm tube investigated, although it was only utilized in the study of friction factor, and not heat transfer. The internal tube diameter was defined as $d_i = (4 A_f / \pi)^{0.5}$ and each tube diameter was calculated from individual cross-sectional areas A_f . To reduce the measurement uncertainties, seven tubes were bundled together, cut, and ground to have smooth cross-sectional surfaces.

Fig. 2 Cross-sectional views of the microtubes used in the present study, (a) stainless steel 304, $d_i = 962 \mu\text{m}$, $d_o = 1260 \mu\text{m}$, (b) nickel, $d_i = 910 \mu\text{m}$, $d_o = 2005 \mu\text{m}$, (c) stainless steel 304, $d_i = 308 \mu\text{m}$, $d_o = 579 \mu\text{m}$, and (d) stainless steel 304, $d_i = 83 \mu\text{m}$, $d_o = 270 \mu\text{m}$

Table 1 Geometrical details of the test section tubes used in the experiments

Tube notation	d_i (μm)	d_o (μm)	Tube materials	L (mm)	$L_{h,x}$ (mm)
Tube A	962	1260	SS 304	330	25, 125, 229
Tube B	910	2005	Nickel	171	19, 35, 57
Tube C	308	579	SS 304	172	16.5, 17.5, 44, 65
Tube D	83	270	SS 304	23	NA

Each tube diameter was thus calculated and all values were averaged to obtain the average tube diameter. The standard deviation of the derived tube diameters was used as the uncertainty. Surface roughness was measured by a laser confocal scanning microscope and found to be between 0.1 and 1 μm . Details of the test tubes are given in Table 1.

Errors in the measurement of the cross-sectional area have a significant impact on the friction factor estimation as described by Celata et al. [13]. Using a 40 $\times$ objective, they measured the diameter to be 84.7 μm and the resulting friction factor was significantly higher than the prediction. When the same measurement was made using a 400 $\times$ magnification microscope, the diameter of the tube was found to be 80.0 μm , which made their data fit the conventional correlation predictions very well.

The cross-sectional area of the tubes affect the axial heat conduction as described by Lin and Kandlikar [19]. The ratio of the Nu without considering the axial conduction and the correct theoretical value is given by the following equation:

$$\frac{\text{Nu}_{k0}}{\text{Nu}_{th}} = \frac{1}{1 + 4 \frac{k_w A_{h,s} \text{Nu}_{th}}{k_f A_f (\text{Re Pr})^2}} \quad (1)$$

where Nu_{k0} is the Nusselt number neglecting the effect of axial heat conduction, Nu_{th} is the theoretical laminar fully developed flow Nusselt number; while $A_{h,s}$ is the cross-sectional area of the channel wall and A_f is the cross-sectional area for fluid flow in the channel. The $A_{h,s}$ values for the 304 stainless steel tube and the nickel tube were 0.52 and 2.51 mm^2 and the A_f values were 0.73 and 0.65 mm^2 , respectively. Furthermore, the thermal conductivity of stainless steel 304 and nickel are 14.9 W/mK and 90.7 W/mK, respectively. The corresponding ratios $k_w/k_f A_{h,s}/A_f$ for the two tubes were 410 and 13,446, respectively; the ratio for the nickel tube being greater by a factor of 32.8. The axial conduction of the nickel tube is expected to be significantly higher than that of stainless steel 304 tube in the range of Re-Pr investigated in the present study.

In addition to the tube diameter measurements, the wall temperature measurements in microtubes can also pose considerable difficulties [15,20]. Due to the small diameter of the microtubes,

thermocouples cannot be properly attached to the tube. Also, there could be a thermal shunt problem [9,10,21–23], since the sizes of the thermocouple and test tube are similar in magnitude. To avoid the thermal shunt problem, a noncontact temperature measurement method using LCT developed by Lin and Yang [9] was utilized in this study. TLCs with active ranges of 42–47 $^{\circ}\text{C}$ and 35–45 $^{\circ}\text{C}$ were used to measure the wall temperatures of 962 μm and 308 μm tubes, respectively.

Figure 3 shows the captured images of a tube with TLC under different temperature conditions. A clear change in hue was observed under different temperatures. The hue of TLC changed from red to yellow, to green, and to blue as temperature increased. The hue values were obtained by averaging the local hue values in the central tube region (30 $\times$ 30 pixels for 962 μm tube and 20 $\times$ 20 pixels for 308 μm tube). The data in the central region had more uniform hue and intensity values, hence the derived temperature values were more accurate as illustrated by Lin and Yang [9].

Each TLC was calibrated in situ individually using E-type thermocouples before conducting experiments. The calibration of LCT hue values to temperature is shown in Fig. 4. Calibration data set for 42–47 $^{\circ}\text{C}$ TLC shows a very linear change of temperature as a function of hue values while a polynomial tendency was observed for 35–45 $^{\circ}\text{C}$ TLC. The slope increases with increasing hue values indicating a poor temperature resolution in the high hue region. Figure 5 shows the temperature resolution dT/dH as a function of hue. The dT/dH represents the temperature accuracy represented by the hue values; the lower the dT/dH , the more accurate the temperature measurement will be. For the 42–47 $^{\circ}\text{C}$ TLC, the temperature accuracy per hue dT/dH slightly increases with hue from about 0.05 $^{\circ}\text{C}/H$ to 0.1 $^{\circ}\text{C}/H$. Comparatively, the 35–45 $^{\circ}\text{C}$ TLC has higher dT/dH values, ranging from about 0.05 $^{\circ}\text{C}/H$ to 0.25 $^{\circ}\text{C}/H$.

From Figs. 4 and 5, it can be concluded that the temperature resolution was low in the high hue region. By maintaining the tube wall temperature at 36–38 $^{\circ}\text{C}$ (or TLC at hue around 60–100) the determined wall temperature has a higher accuracy. The same hue value trend was observed for the 42–47 $^{\circ}\text{C}$ TLC. It should be noted that the temperature resolution accuracy for 42–47 $^{\circ}\text{C}$ TLC toward the lower or higher hue region is better than that of 35–45 $^{\circ}\text{C}$ TLC.

Fig. 3 TLC Thermography images of (a) 962 μm tube and (b) 308 μm tube at different temperatures

Fig. 4 Calibration curve of TLC hue to temperature

Fig. 5 Temperature resolution of thermochromic liquid

The data reduction procedure is the same as that employed in Part I [18] of this study.

3 Results

The primary result of this work was the determination of friction factor and heat transfer coefficient for air flow of microtubes of different diameters, materials, and wall thicknesses. The data were obtained using two different temperature measurement methods.

3.1 Friction Factor. Figure 6 shows the friction factor in three commercial smooth stainless steel 304 tubes with diameters of 962 μm , 308 μm , and 83 μm . The friction factor can be predicted very well by conventional correlations. For laminar flow, the friction factor followed the prediction of Hagen–Poiseuille flow theory; while for the turbulent flow, the data agreed well with the Blasius equation. The transition from laminar to turbulent flow was observed in the range of $\text{Re}=2000\text{--}3000$, which agrees with the conventional transition Re . There was no scale effect in friction factor observed for the smooth tubes in the experimental diameter range. The experimental uncertainties for f and Nu are shown in Table 2 and are not included in the plot to avoid overcrowding.

3.2 Heat Transfer. The wall temperatures in this study were measured by both thermocouples and LCT in order to compare the two measurement methods. Experiments were performed at different heat fluxes to validate the heat loss estimation. The Nu data for 962 μm stainless steel 304 tube under different heat fluxes and tube wall measurement methods is shown in Fig. 7. The Nu for 962 μm tube was seen to be in good agreement with conventional correlations shown in Part I of the present study [18]. All of the Nu data sets reveal a good agreement with conventional correlations which verified that both temperature measurement methods and the heat loss estimation are accurate.

Fig. 6 Friction factors for stainless steel 30 tubes of three different diameters

Table 2 Uncertainty in measurement of experimental parameters

Uncertainty	$d_i = 962 \mu\text{m}$	$d_i = 308 \mu\text{m}$	$d_i = 83 \mu\text{m}$
Nu (%)	Turbulent: 5–8 Laminar: 10–23	Turbulent: 7–9 Laminar: 10–39	N/A
$T_{f,x}$ (°C)	Turbulent: 0.1–0.2 Laminar: 0.2–0.8	Turbulent: 0.2–0.3 Laminar: 0.3–1.0	N/A
$T_{w,x}$ (°C)	0.2	0.2	N/A
$\dot{m}$ (%)	1–6	1–11	2–15
ΔP (%)	1.0–9.2	0.9–7.5	0.1–2.0
Re (%)	1–7	2–10	3–16
d_i (μm)	1.3	3.5	1.3
f (%)	3–16	5–14	4–19

Fig. 7 Comparison of Nu as a function of Re for 962 μm stainless steel 304 tube with different heat flux and temperature measurement methods

Figure 8 shows the Nu values for air flow in 962 μm and 308 μm diameter tubes as a function of Re . For laminar flow. The experimental Nu data is seen to be in good agreement with the theoretical value of 4.364 for laminar fully developed flow. The Nu slightly increases with Re due to the increase in developing length. For turbulent flow, the heat transfer coefficient increases significantly with Re and is predicted very well by the Gnielinski correlation. The Re transition from laminar to turbulent flow is in the region of 2000–4000 which matches conventional flow. From the Nu data, it is seen that heat transfer coefficient in this diameter range can be predicted well by the conventional correlations and there are no microscale effects.

Figure 9 shows the viscous heating effect on the heat transfer of 308 μm and 962 μm tubes. The experimental Nu values are normalized as $\text{Nu}^*_{\text{viscous}}$ by dividing them with Nu obtained without considering viscous heating. From Eq. (10) in Part I of this study [18], it is observed that viscous heating increases the net heat input to the fluid and as a result, the local fluid temperature

Fig. 8 Heat transfer comparison for 308 μm and 962 μm inner diameter stainless steel 304 tubes

Fig. 9 Viscous heating effect in 308 μm and 962 μm inner diameter stainless steel 304 tubes during laminar flow

Fig. 10 Comparison of heat transfer performance by different wall temperature measurement methods for 308 μm inner diameter tube

increases. Viscous heating effects also result in an increase in the estimated Nu value due to the increased net heat input to the fluid. It is seen from Fig. 9 that the viscous heating effect on 308 μm tube is significantly higher than that of 962 μm tube. The value of Nu^*_{viscous} is close to 1 at low Re for the 962 μm tube and increases with Re. The increase is about 2%–10% from low to high Re. However, for the 308 μm tube the increase is 14%–28% due to the increase in flow velocity and viscous heating effect.

Figure 10 compares the experimental Nu between the two measurement methods. There is a minimal difference observed between the two temperature measurement methods; using thermocouples or TLC.

The comparison of axial heat conduction effects in 308 μm tube to conventional correlations and the correlation proposed by Lin and Kandlikar [19] is shown in Fig. 11. The theoretical Nu value

Fig. 11 Comparison of experimental Nu with and without considering axial heat conduction for stainless steel 304 tubes, $d_i = 308 \mu\text{m}$, $d_o = 2005 \mu\text{m}$

Fig. 12 Axial conduction effects on heat transfer coefficient for thick walled nickel tube $d_i = 910 \mu\text{m}$, $d_o = 2005 \mu\text{m}$

is 4.364 for laminar fully developed flow in the smooth tube with constant heat flux heating. From the correlation proposed by Lin and Kandlikar [19] it is observed that the effect of axial conduction is negligible for Re higher than 1000. For Re less than 1000 the Nu values are lower than 4.364 and decrease with Re. At Re = 200, the predicted Nu is approximately 2.0. As shown in Fig. 11, the axial heat conduction effects need to be considered for this case as given by Eq. (1).

The axial conduction effect on Nu as a function of Re for the thick wall nickel tube is shown in Fig. 12. It was found that without considering axial conduction, the calculated Nu is significantly lower than the theoretical laminar fully developed flow values for Re as high as 1000. The axial conduction effect was negligible for Re higher than 1000 for the 308 μm 304SS tube, as shown in Fig. 11. However, the axial conduction equation proposed by Lin and Kandlikar [19] revealed that the axial conduction was not negligible for the thick wall nickel tube with Re is as high as 1000. This is due to the increased thickness of the tube wall and the higher thermal conductivity of nickel. The thermal conductivities for stainless steel 304 and nickel are 14.9 and 90.7 W/mK, respectively. The axial heat conduction effect due to the thermal conductivity of nickel tube is more than six times greater than that of the stainless steel tube. Furthermore, the ratio of cross-sectional area to the fluid channel area for nickel tube is greater than the 304SS tube. The k_w/k_f , $A_{h,s}/A_f$ in Eq. (1) for 304SS and nickel tubes are 1398 and 12948, respectively, differing by a factor of 9.3. At Re = 250, the predicted Nu is approximately 0.5, as shown in Fig. 12, which is much lower than the theoretical value without considering axial heat conduction.

4 Conclusions

Heat transfer of air flow in microtubes was experimentally investigated in this study. The effect of scale and tube wall

thickness on Nu was discussed in detail. The tube wall temperature was measured by means of both thermocouples and liquid crystal thermography for tube diameters higher than 308 μm . The following conclusions are drawn from the study:

1. Viscous heating effects need to be considered for air flow in small diameter tubes. These effects increase with Re.
2. Accurate estimation of heat losses, which depend on the local wall temperature, is extremely important in obtaining reliable heat transfer data with gas flow in microtubes.
3. Axial heat conduction causes an underestimation of Nusselt number and the magnitude of deviation increases with decreasing Re, increasing tube wall thickness, and increasing tube wall material thermal conductivity. The model proposed by Lin and Kandlikar [19] was able to predict the experimental data considering the axial conduction effects. After considering the effects of entrance region, variable heat loss, viscous heating, and axial conduction, the heat transfer coefficient estimation for air flow in microtubes agree well with the conventional correlations for tube diameter higher than 308 μm tested in this study. Neglecting some or all of these effects is believed to be primary reasons for discrepancies in the experimental heat transfer data published in literature.

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Nomenclature

- $A_{h,s}$ = cross-sectional area of channel, m^2
 A_f = cross-sectional area of fluid flow in channel, m^2
 D_h = hydraulic diameter, m
 d_i = internal tube diameter, m
 d_o = external tube diameter, m
 f = friction factor, dimensionless
 H = hue, dimensionless
 k_w = thermal conductivity of the wall, $\text{W/m } ^\circ\text{C}$
 k_f = thermal conductivity of the fluid, $\text{W/m } ^\circ\text{C}$
 L = tube length, m
 $L_{h,x}$ = local heating location distance from the entrance, m
 $\dot{m}$ = mass flow rate, kg/s
 $\text{Nu}_{k,0}$ = Nusselt number without axial conduction, dimensionless
 Nu_{th} = theoretical Nusselt number, dimensionless
 Nu_x = local Nusselt number, dimensionless
 Pr = Prandtl number, dimensionless
 Re = Reynolds number, dimensionless
 $T_{f,x}$ = local fluid temperature, $^\circ\text{C}$
 $T_{w,x}$ = local wall temperature, $^\circ\text{C}$

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