The Nu Number Behavior on Micro-tube Heat Transfer and Fluid Flow of Dielectric Fluid

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Abstract: The numerical modeling of the microchannel heat transfer and fluid flow was analyzed to estimate the thermal behavior of the dielectric fluids with emphasize on Nu number behavior. The geometry of a model is the same as in (Lelea *et al*., 2004). The diameters of the tube were D_i /D_o=125.4/300 μm, the total length L=70 mm, Q=0.75 W and only a portion of the tube was heated. The laminar and stationary regime was considered with variable fluid properties. Besides, two different heat flux directions are considered: heating and cooling.

INTRODUCTION

Micro Thermal Systems (MTS), defined as the systems in which the key size has a length scale of a micrometer, could attain the high heat transfer coefficients. For instance, they are used as the cooling devices for LSI chips. On the other hand μ - TAS (Micro Total Chemical Analyzing System), MEMS (Micro Electric – Mechanical Systems) or bio – chips are some of the examples of MTS. The research reports in this field, concerning the thermal and hydrodynamic results, are mostly oriented to the water as a working fluid. However, in defense electronics applications, like radars, lasers or avionics the dielectric fluids are used due to a sensibility of the operating conditions.

The first microchannel fluid flow experiment was made by Poiseuille [1] in 1870 on a glass tube with internal diameter ranging from 29 to 140 μm with water as the working fluid and non-heating working conditions. Based on these results, the well-known relation for the volume flow rate was established and extended lately to the macrochannels.

Tuckerman and Pease [2] have increased the interest on microchannel heat transfer phenomena with the microchannel heat sink used for the cooling of the VLSI devices. Wu and Little [3, 4] have made the microchannel heat transfer and fluid flow experiments used for designing the Joule-Thomson micro-refrigerator. The working fluid in their research was nitrogen and inner diameters of the tubes were from 100 to 300 μm. Their heat transfer and hydrodynamic results shown differences against the conventional results for macrotubes. In the following years, large amount of work has been done in order to explain these discrepancies.

Morini [5] has also presented the review on a single phase microchannel heat transfer, indicating some of the reasons for a large dispersion of the experimental results. Both gas and liquid flows have been considered.

In the recent years, Lelea *et al*. [6], have made the experimental research on microtube heat transfer and fluid flow with inner diameters between 100 and 500 μm for laminar regime of the water flow. These results have shown the good agreement with the conventional theories even for the entrance region of the tube.

Tiselj *et al*. [7] have presented the experimental research on microchannel heat transfer and fluid flow of water through the multichannel configuration with triangular cross-section of the channels. The hydraulic diameter was 160 μm and low Re number range (3.2–64) considering the axial conduction in the tube wall. The results are also in good agreement with conventional theories, confirmed also with the numerical modeling using the conventional set of the Navier – Stokes equations.

The outcome of the research reports mentioned above is that special attention has to be paid to macroscale phenomena that are amplified at the microscale. For example, due to a high heat transfer rate, the temperature variable fluid properties have to be considered. Lelea [8] has investigated the influence of the temperature dependent fluid viscosity on Po number. On the other hand, the small diameter and large length of the tube can result in viscous heating even in the case of liquid flow, as presented in Koo & Kleinstreuer [9].

Maranzana *et al*. [10] have analyzed the influence of the axial conduction in the tube wall on microchannel heat transfer. The fluid flow between the parallel plates as well as the fluid flow in the counter flow heat exchanger was considered. It was found that the axial conduction has the influence on heat transfer parameters as far as the axial conduction number M, defined in this study, is higher than 10^{-2}. In this case the total length of the tube was heated. However, in industrial or laboratory applications, the heating length is not always equal to the total length of the tube.

Moreover, most of the research results have the water as the working fluids. Because of the sensibility of some specific electronic devices, water might not be a suitable fluid, so the dielectric fluids must be used. In the present

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research the dielectric fluid Novec-7600 [11] is used for calculations. In addition, the variable fluid properties have a great influence on some dimensionless criteria like Nu number.

Following these ideas, the microtube heat transfer fluid flow of dielectric fluids with variable fluid properties is considered in this paper. Special attention is paid to Nu number behavior for three different heat transfer applications (cooling, heating and k=const.).

PROBLEM DESCRIPTION AND NUMERICAL DETAILS

In order to discuss the axial conduction influence, the velocity and temperature distributions were numerically solved considering the temperature variation of the fluid properties, procedure described in Lelea *et al.* [6].

The computational domain is presented in Fig. (**1**), as follows:

The fluid flow domain defined at $r = 0, R_i$ and $z = 0$, L

The temperature field domain defined at $r = 0, R_o$ and $z=0, L$

The outer portion of the tube has two parts, the heated and insulated part. So the respective insulated part was included in the numerical domain. The following set of partial differential equations is used to describe the phenomena, considering the variable thermophysical properties of the dielectric fluid.

Continuity equation

$$\frac{\partial(\rho(T)\cdot u)}{\partial z}+\frac{1}{r}\frac{\partial(r\cdot\rho(T)\cdot v)}{\partial r}=0 \tag{1}$$

Momentum equation

$$\frac{\partial(\rho(T)vu)}{\partial r}+\frac{\partial(\rho(T)uu)}{\partial z}=-\frac{dp}{dz}+\frac{1}{r}\frac{\partial}{\partial r}\left(\mu(T)r\frac{\partial u}{\partial r}\right) \tag{2}$$

Energy equation

$$\frac{\partial(\rho(T)c_p(T)vT)}{\partial r}+\frac{\partial(\rho(T)\cdot c_p(T)uT)}{\partial z}= \cdot$$

$$\left[\frac{1}{r}\frac{\partial}{\partial r}\left(k(T)\cdot r\frac{\partial T}{\partial r}\right)+\frac{\partial}{\partial z}\left(k(T)\frac{\partial T}{\partial z}\right)\right] \tag{3}$$

At the inlet of the tube, the uniform velocity and temperature field is considered, while at the exit the temperature gradient is equal to zero.

The boundary conditions are

$z = 0$, $0 < r < R_o$: $u = u_0$, $T = T_w = T_0$

$0 < z < L_{tot}$:r=0, $\frac{\partial u}{\partial r}=0$, $\frac{\partial T}{\partial r}=0$, $v = 0$

$r = R_i$, $u = v = 0$

The Joule heating of the tube wall can be expressed either by the uniform heat generation through the tube wall or by the uniform heat flux imposed on the outer surface of the wall. For the latter case, the boundary condition is defined as,

$r = R_o$: $q_O = k_S\frac{\partial T}{\partial r}$ (for the heated portion of the tube)

$k_S\frac{\partial T}{\partial r}=0$ (for the insulated portion of the tube)

where q_o is the heat flux based on the outer heat transfer area of the tube wall.

$z = L_{tot}$, $0 < r < R_o$: $\frac{\partial T}{\partial z}=0$

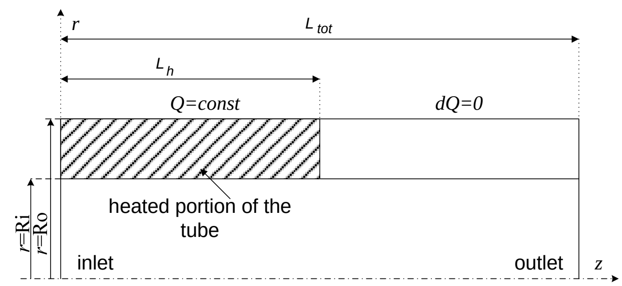

Fig. (1). The calculation domain.

$r = R_i$: $T_s|_{Ri+} = T_f|_{Ri-}$

$$k_S\left(\frac{\partial T_S}{\partial r}\right)_{Ri+}=k_f\left(\frac{\partial T_f}{\partial r}\right)_{Ri-}$$

The fluid properties of the Novec-7600 were considered as temperature dependent with the following:

Dynamic viscosity:

$$\mu(t)=(1587.5-1.755\cdot t)\cdot 10^{-6}\cdot e^{\frac{464.403382}{t+133}-2.881482}$$

Density:

$$\rho(t)=1587.5-1.755\cdot t$$

Thermal conductivity:

$$k(t)=0.078-0.0003\cdot t$$

Specific heat:

$$c_p(t)=3.1631\cdot t+1240.2$$

The partial differential equations (1)–(3) together with boundary conditions are solved using the finite volume method described in [12].

First, the parabolic flow field condition is considered and the velocity field is solved. The temperature field, as a conjugate heat transfer problem, was then solved as the elliptic problem using the obtained velocity field.

Because of the temperature dependent fluid properties, iterative procedure is needed to obtain the convergence of the fluid properties (viscosity, thermal conductivity, density and specific heat capacity) through the successive solution of the flow and temperature field. A tube wall and water flow inside the tube, are considered as one domain and harmonic mean values for the thermal conductivity are calculated at the solid-liquid interface [12]:

$$k_m = \left(\frac{1-f}{k_s} + \frac{f}{k_f} \right)^{-1}$$

The viscosity in the solid region was set to a very large value, in order to handle discontinuities between these two domains. Consequently the convective terms in the energy equation will be zero. So in this case the separate numerical treatment of the temperature field in the solid and fluid region is avoided. In order to test the grid sensitivity, two grids have been used. The coarser one with 250 cells in radial direction and 400 cells in axial direction and finer grid with 500 and 800 cells in *z*- and *r*- direction respectively. Differences obtained for Po and Nu were smaller than 0.1 %, so the coarser grid has been used for further calculations. Further details regarding the numerical code are presented in [13].

RESULTS AND DISCUSSION

The microtube conjugate heat transfer analysis was made for tubes with following ratio D_i/D_o = 125.4/300 μm and silicon substrate (k = 198 W/m K). The laminar flow regime was considered with Re < 400. The input heat transfer rate was constant for all the runs Q_0 = 0.75 W. Also, the upstream heating is considered as presented in Fig. (**1**).

In Fig. (**2**) the local Nu variation versus axial distance is presented for Re=86.5 and Pr=23.9. Three different cases were considered, cooling, heating and constant thermal conductivity. For the cases of the constant thermal properties the local Nu decreases and approaches the constant value depending on cross-section, Re or Pr.

In the case of heating the local Nu is constantly decreasing along the heating portion of the tube. For the cooling case the Nu is decreasing and after the half of the length the Nu is increasing. In the case of k=const. the Nu follows the usual behavior approaching the well-known fully developed value Nu = 4.36.

In Fig. (**3**) the local Nu distribution versus axial distance is presented for Re = 139.8 and Pr = 27.3. In the case of heating the Nu decreases similar to the previous case. For cooling case Nu decreases and approaches the constant fully developed value Nu = 4.36.

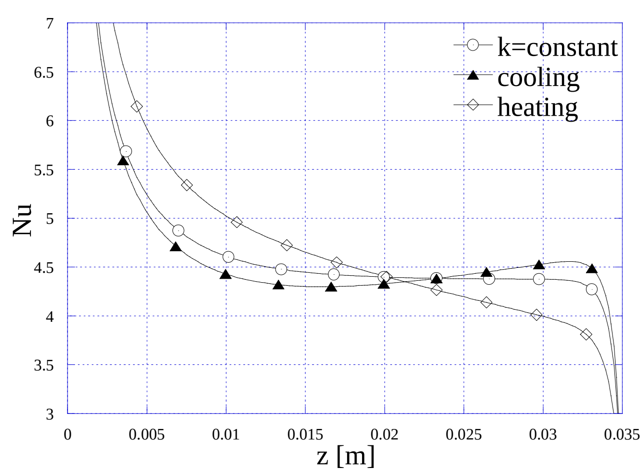

Fig. (2). The local Nu distribution versus axial distance for Re = 86.5 and Pr = 23.9.

Finally in Fig. (**4**) local Nu versus axial distance is presented for Re = 354.5 and Pr = 30.6. Due to the higher velocities and higher Pr the thermal entrance length is longer so the Nu does not approach the fully developed value.

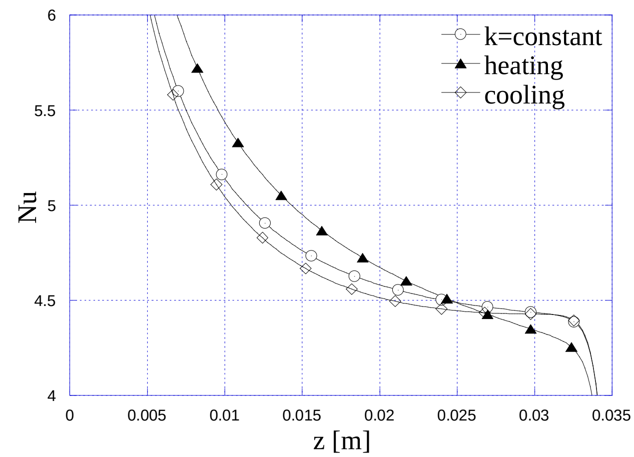

Fig. (3). The local Nu distribution versus axial distance for Re = 139.8 and Pr = 27.3.

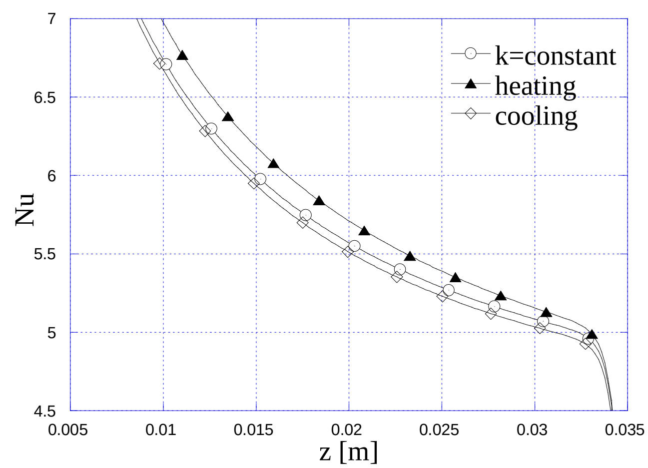

Fig. (4). The local Nu distribution versus axial distance for Re = 354.5 and Pr = 30.6.

CONCLUSIONS

The numerical modeling of the microtube heat transfer and fluid flow is analyzed. A dielectric fluid Novec-7600 is considered with variable thermophysical properties. The special attention is put on a Nu number behavior for two different heat transfer applications (heating and cooling). The following outcomes might be emphasized from the results presented in the Figs. (**2-4**):

- Both the temperature dependent fluid properties and heat flux direction (heating or cooling) have a strong influence on Nu number variation;
- For lower Re = 86.5 the Nu number has a different variation depending on the heat flux direction. After the initial decreasing the Nu increases in the case of cooling and decreases in the case of heating. If the thermal conductivity is considered constant, the Nu is approaching the fully developed constant value Nu=4.36;

- On the other hand for Re = 139.8 and 354.5 the Nu decreases for all cases due to the higher Re and Pr number.

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NOMENCLATURE

A, m^2	=	Cross-section area
D , m	=	TUBE diameter
k , W/mK	=	Thermal conductivity
L , m	=	Length
Δp , Pa	=	Pressure drop
Po -,	=	Poiseuille number
Pe - ,	=	Peclet number
Q , W	=	Heat transfer rate
q , W/m^2	=	Heat flux
R, m	=	Tube radius
Re	=	Reynolds number
T , K	=	Temperature
u , v , m/s	=	Velocity components
V , m^3/s	=	Volume flow rate
x, y, z	=	Spatial coordinates

Greek Symbols

δ m,	=	Tube wall thickness
μ Pa s,	=	Viscosity
ρ kg/m^3,	=	Density

Subscripts

i	=	Inner
in	=	Inlet
out	=	Outlet
o	=	Outer
m	=	Mean
s	=	Solid
f	=	Fluid

REFERENCES

[1] Poiseuille, "Recherches experimetales sur la mouvement des liquids dans les tubes de tres petits diametres", Academie des Sciences, *Comptes Rendus*, Vol. 12, pp. 112-115, 1841.

[2] D. B. Tuckerman and R. F. W. Pease, "High-performance heat sinking for VLSI", *IEEE Electron. Device Lett.*, vol. EDL-2, pp. 126-129, May 1981.

[3] P. Y. Wu and W. A. Little, "Measurement of friction factor for the flow of gases in very fine channels used for micro miniature joule thompson refrigerators", *Cryogenics*, Vol. 23, pp. 273-277, May 1983.

[4] P. Y. Wu and W. A. Little, "Measurement of the heat transfer characteristics of gas flow in fine channel heat exchangers for micro miniature refrigerators", *Cryogenics*, Vol. 24, pp. 415-420, August 1984.

[5] G. L. Morini, "Single-phase convective heat transfer in microchannels: A review of experimental results", *Int. J. Thermal Sci.*, Vol. 43, pp. 631—651, July 2004.

[6] D. Lelea, S.Nishio, K.Takano, "The experimental research on microtube heat transfer and fluid flow of distilled water", *Int. J. Heat Mass Transfer*, Vol. 47, pp. 2817 - 2830, June 2004.

[7] I. Tiselj, G. Hetsroni, B. Mavko, A. Mosyak, E. Pogrebnyak Z. Segal, "Effect of axial conduction on the heat transfer in microchannels", *Int. J. Heat Mass Transfer*, Vol. 47, pp. 2551 - 2565, June 2004.

[8] D. Lelea, "Some considerations on frictional losses evaluation of a water flow in microtubes", *Int. Commun. Heat Mass Transfer*, Vol. 32, pp. 964-973, July 2005.

[9] J. Koo, C. Kleinstreuer, "Viscous dissipation effects in microtubes and microchannels", *Int. J. Heat Mass Transfer*, Vol. 47, pp. 3159 - 3169, July 2004.

[10] G. Maranzana, I. Perry, D. Maillet, "Mini- and micro-channels: influence of axial conduction in the walls", *Int. J. Heat Mass Transfer*, Vol. 47, pp. 3993-4004, August 2004.

[11] *** 3M™ Novec™ 7600 Engineered Fluid, 3M Electronics Markets Materials Division, 2008.

[12] S.V. Patankar, *Numerical Heat Transfer and Fluid Flow*, New York: McGraw Hill, 1980.

[13] D.Lelea, "The conjugate heat transfer of the partially heated microchannels", *Heat Mass Transfer*, Vol. 44, pp. 33-41, November 2007.

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