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The particle thermal conductivity influence of nanofluids on thermal performance of the microtubes☆

Dorin Lelea *, Ioan Laza

University Politehnica Timisoara, Faculty of Mechanical Engineering, Department of Thermal Machines and Transportation, B-dul Mihai Viteazu nr. 1, 300222 Timisoara, Romania

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ABSTRACT

The paper presents the numerical analysis on microchannel laminar heat transfer and fluid flow of nanofluids in order to evaluate the suitable thermal conductivity of the nanoparticles that results in superior thermal performances compared to the base fluid. The diameter ratio of the micro-tube was $D_i/D_o = 0.3/0.5$ mm with a tube length $L = 100$ mm in order to avoid the heat dissipation effect. The heat transfer rate was fixed to $Q = 2$ W. The water based Al_2O_3, TiO_2 and Cu nanofluids were considered with various volume concentrations $\phi = 1,3$ and 5% and two diameters of the particles $d_p = 13$ nm and 36 nm. The analysis is based on a fixed Re and pumping power Π, in terms of average heat transfer coefficient and maximum temperature of the substrate. The results reveal that only the nanofluids with particles having very high thermal conductivity ($\lambda_{Cu} = 401$ W/m K) are justified for using in microcooling systems. Moreover, the analysis is sensitive to both the comparison criteria (Re or Π) and heat transfer parameters (h_{ave} or t_{max}).

1. Introduction

The microchannel heat transfer and fluid flow are basic phenomena in various micro-thermal devices. The cooling of VLSI devices, biomedical applications, and micro-heat-exchangers is among the examples where the fundamentals of the microchannel heat transfer and fluid flow are essential for a proper design of these devices. Moreover, the nanofluids [1] might be considered as a suitable solution for enhancement of the cooling capabilities if the proper nanoparticles are selected.

Yang et al. [2] investigated experimentally the laminar heat transfer and fluid flow of graphite nanofluids in horizontal tubes. The experimental results show that the nanoparticles increase the heat transfer coefficient of the fluid system in laminar flow, but the increase is much less than that predicted by current correlation based on static thermal conductivity measurements.

Experimental research on developing laminar and turbulent heat transfer of water-based FMWNT nanofluid in a uniformly heated horizontal tube was made by Amrollahi et al. [3] while the experimental research on heat transfer of Al_2O_3/propanol nanofluid was made by Sommers and Yerkes [4]. Both reports indicated the convective heat transfer coefficient enhancement if nanofluid is used.

Rea et al. [5] analyzed experimentally the laminar convective heat transfer of alumina–water and zirconia–water nanofluids. The data expressed in form of dimensionless numbers (Nu and x+), show good agreement with the predictions of the traditional models/correlations for laminar flow. This suggests that the nanofluids behave as homogeneous mixtures.

Heris et al. [6] analyzed numerically laminar-flow convective heat transfer of nanofluid in a circular tube with constant wall temperature boundary condition. The numerical results indicate that the addition of nanoparticles to base liquid produces considerable enhancement of heat transfer. The comparison of different approach for numerical modeling on heat transfer of nanofluids was presented by Hejazian at al [7] and Akbari et al. [8]. The later report considered that with appropriate selection of the relations for the effective nanofluid properties the single-phase approach gives better results.

The turbulent heat transfer of CuO/water nanofluids inside circular tubes was investigated experimentally by Fotukian and Esfahany [9], while Chon et al. [10] proposed correlation for effective thermal conductivity of nanofluids.

Lee and Mudawar [11] analyzed experimentally the effectiveness of the nanofluids for single-phase and two-phase heat transfer in micro-channels. Higher heat transfer coefficients were achieved in the entrance region of micro-channels proving that nanoparticles have an appreciable effect on thermal boundary layer development. Ghasemi et al. [12] analyzed natural convection heat transfer in an inclined enclosure filled with a CuO/water nanofluid. The results indicate that adding nanoparticles into pure water improves its heat transfer performance.

Also a new model for assessment of the effective viscosity of water based nanofluids was developed by Masoumi et al. [13]. Finally an

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* Corresponding author.
E-mail address: dorin.lelea@upt.ro (D. Lelea).

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Nomenclature

c_p	J/kg K, specific heat
D	m, tube diameter
D_p	nm, particle diameter
D_{bf}	nm, molecular diameter of the base fluid
h	W/m^{2}K, heat transfer coefficient
k	W/mK, thermal conductivity
k_b	J/K, Boltzmann constant
L	m, length
l	m, mean free path
Q	W, heat transfer rate
q	W/m^2, heat flux
Pr	–, Prandtl number
R	m, tube radius
Re	Reynolds number
T	K, temperature
u	v, m/s, velocity components
M	kg/s, mass flow rate
r	z, spatial coordinates

Greek symbols

α	m^2/s, thermal diffusivity
ϕ	%, particle volume fraction
μ	Pa s, viscosity
ρ	kg/m^3, density
Π	W, pumping power

Subscripts

ave	average
b	bulk
bf	base fluid
eff	effective
f	fluid
i	inner
in	inlet
nf	nanofluid
out	outlet
o	outer
p	particle
s	solid
w	wall

overview on enhancement of heat transfer using nanofluids was presented by Godson et al. [14].

In order to evaluate the suitable thermal conductivity of the particle that gives better thermal performance of the microscale device compared with the base fluid, three different nanofluid types are considered: TiO_2–water, Al_2O_3–water and Cu–water. Additionally, in order to emphasize the differences between various approaches, the analysis is made on a fixed Re and pumping power basis. The large majority of the research reports considered the analysis based on a fixed Re. As it was revealed by Haghighi et al. [15] and Lelea et al. [16] this is not a very suitable comparison criteria considering that larger viscosity implies higher velocity for the fixed Re.

2. Numerical details

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 following set of partial differential equations is used to describe the phenomena, considering the effective thermophysical properties of fluids:

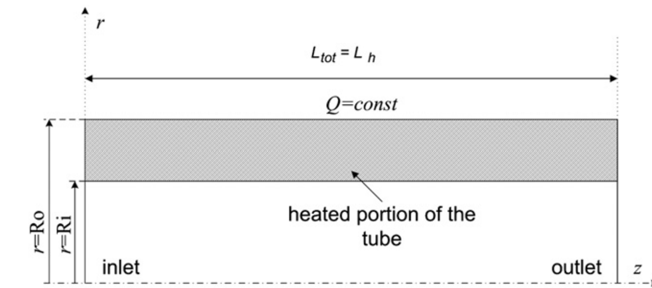

Fig. 1. The calculation domain.

Continuity equation:

$$\frac{\partial(\rho_{\mathrm{eff}} \cdot u)}{\partial z} + \frac{1}{r}\frac{\partial(r \cdot \rho_{\mathrm{eff}} \cdot v)}{\partial r} = 0 \tag{1}$$

Momentum equation:

$$\frac{\partial(\rho_{\mathrm{eff}} v u)}{\partial r} + \frac{\partial(\rho_{\mathrm{eff}} u u)}{\partial z} = -\frac{dp}{dz} + \frac{1}{r}\frac{\partial}{\partial r}\left(\mu_{\mathrm{eff}} r \frac{\partial u}{\partial r}\right) \tag{2}$$

Energy equation:

$$\frac{\partial\left(\rho_{\mathrm{eff}} c_{p\,\mathrm{eff}} v T\right)}{\partial r} + \frac{\partial\left(\rho_{\mathrm{eff}} \cdot c_{p\,\mathrm{eff}} u T\right)}{\partial z} = \left[\frac{1}{r}\frac{\partial}{\partial r}\left(k_{\mathrm{eff}} \cdot r \frac{\partial T}{\partial r}\right) + \frac{\partial}{\partial z}\left(k_{\mathrm{eff}} \frac{\partial T}{\partial z}\right)\right] \tag{3}$$

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

The boundary conditions are:

$$z = 0\,,\ 0 < r < R_o: \quad u = u_0\,, T = T_w = T_0 \tag{4}$$

$$0 < z < L_{\mathrm{tot}}: \quad \begin{array}{l} r = 0\,, \dfrac{\partial u}{\partial r} = 0\,,\ \dfrac{\partial T}{\partial r} = 0\,,\ v = 0 \\ r = R_i, u = v = 0 \end{array} \tag{5}$$

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,

$$r = R_o : q_o = k_s \frac{\partial T}{\partial r} \tag{6}$$

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

$$z = L_{tot},\ 0 < r < R_o: \quad \frac{\partial T}{\partial z} = 0. \tag{7}$$

The conjugate heat transfer procedure implies the continuity of the temperature and heat flux at the solid–liquid interface defined as,

$$r = R_i : T_s|_{Ri+} = T_f|_{Ri-} \tag{8}$$

$$k_s\left(\frac{\partial T_s}{\partial r}\right)_{Ri+} = k_{\mathrm{eff}}\left(\frac{\partial T_f}{\partial r}\right)_{Ri-}. \tag{9}$$

The partial differential Eqs. (1)–(3) together with boundary conditions are solved using the finite volume method described in [17].