Numerical simulation of nanofluids for improved cooling efficiency in a 3D copper microchannel heat sink (MCHS)

L. Snoussi, N. Ouerfelli, K.V. Sharma, N. Vrinceanu, A.J. Chamkha & A. Guizani


To link to this article: https://doi.org/10.1080/00319104.2017.1336237

Published online: 12 Jun 2017.

Article views: 83

View related articles

View Crossmark data

Citing articles: 1 View citing articles
Numerical simulation of nanofluids for improved cooling efficiency in a 3D copper microchannel heat sink (MCHS)

L. Snoussi a, N. Ouerfelli b,c, K.V. Sharma d, N. Vrinceanu e, A.J. Chamkha f and A. Guizani a

aUniversity of Carthage, Thermal Processes Laboratory (LPT), Research and Technology Center of Energy (CRTEn), Borj Cedria, Hammam-Lif, Tunisia; bUniversité de Tunis El Manar, Laboratoire de Biophysique et Technologies Médicales, LR13ES07, Institut Supérieur des Technologies Médicales de Tunis, Tunis, Tunisia; dDepartment of Chemistry, College of Science, University of Dammmam, Dammmam, Saudi Arabia; eCentre for Energy Studies, Department of Mechanical Engineering, JNTUH College of Engineering, Kukatpally, Hyderabad, Telangana, India; fDepartment of Industrial Machinery and Equipment, “Lucian Blaga” University of Sibiu, Sibiu, Romania; gMechanical Engineering Department, Prince Mohammad Bin Fahd University, Al-Khobar, Saudi Arabia

ABSTRACT
In this paper, laminar nanofluid flow in 3D copper microchannel heat sink (MCHS) with rectangular cross section, and a constant heat flux, has been treated numerically using the computational fluid dynamics software (FLUENT). Results for the temperature and velocity distributions in the investigated MCHS are presented. In addition, experimental and numerical values are compared in terms of friction factors, convective heat transfer coefficients, wall temperature and pressure drops, for various particle volume concentrations and Reynolds numbers. The numerical results show that enhancing the heat flux has a very weak effect on the heat transfer coefficient for pure water, but an appreciable effect for the case of a nanofluid. For all considered volume fractions, the sink friction factor decreases by increasing the Reynolds number and slightly increases by increasing the volume fractions, and, with increasing the volume fractions and the Reynolds number, the pressure drop increases.

ARTICLE HISTORY
Received 6 April 2017
Accepted 25 May 2017

KEYWORDS
Rectangular microchannel heat sink (MCHS); nanofluids; volume fraction; heat transfer enhancement; CFD

1. Introduction
In order to drive the development of a compact and efficient thermal management technology for advanced electronic devices, cooling of electronic chip is essential for the proper functioning, in which it is used. At the present time, it has become a major concern to electronic packaging engineers. The microchannel heat sink (MCHS) cooling technology was first proposed by Tuckerman and Pease [1]. In silicon MCHS, the circulated water was able to reach a heat flux of 790 W/cm² without a phase change in a pressure drop of 1.94 bar. Several tests have been conducted by Peng and Peterson [2] in an MCHS with hydraulic diameters ranging from 133 to 367 μm; these tests show a friction factor dependence on hydraulic diameter and the cross-sectional aspect ratio.

New cooling devices modelling approaches have been inspired by the MCHS idea using nanofluids as coolants. A nanofluid is a dispersion of ultra-fine particles in a base fluid such as water, ethylene glycol (EG), oils and others, which tremendously enhances heat transfer characteristics of the original fluid study. The nanoparticles used to produce nanofluids in the reviewed literature are aluminium oxide (Al₂O₃), copper (Cu), copper oxide (CuO), gold (Au), silver (Ag), silica nanoparticles and carbon nanotubes [3].
Koo and Kleinstreuer [4] studied the effect of volume fraction on different parameters of MCHS. Their results proved that using nanoparticles with high thermal conductivity is more advantageous, and a channel with high aspect ratio is desirable. Jang and Choi [5] presented a numerical study in which the MCHS performance cooling was examined using nanofluids. This study showed that the cooling performance was enhanced by approximately 10% with water-based Diamond (1%, 2 nm) at a fixed pump power to 2.25 W compared to an MCHS with water. A literature review shows that limited studies are available on the flow and heat transfer characteristics in MCHS, which motivated the studies [6–8]. The result of this work showed that the convective heat transfer coefficient of a nanofluid is higher than that of the base fluid although there is a slight increase in pressure drop due to the presence of nanoparticles in an MCHS. Several factors are at responsible for this enhancement namely the Reynolds number, particle volume fraction, and particle size and shape.

A three-dimensional (3D) numerical model of the fluid flow and heat transfer for a rectangular MCHS has been developed by Qu and Mudawar [9,10]. This model finds that the Reynolds number would influence the length of the developing flow region, and it was also found that the highest temperature was typically encountered at the heated base surface of the heat sink immediately adjacent to the channel outlet. This was similar to that proposed by both Kawano et al. [11] and Fedorov and Viskanta [12]. Lee and Mudawar [13] have done an experimental work to explore the microchannel cooling benefits of Al$_2$O$_3$/water nanofluid. This experimental work was performed with small concentrations of nanofluid (1% and 2% concentration). Their theoretical results show that the friction factor was slightly changed, but the pressure drop increased. The heat transfer enhancement increased with the particle volume concentration particularly in the entrance region than the downstream fully developed region.

In addition, Chamkha et al. [14] presented the physical phenomena regarding heat transfer enhancement, cooling and control involving the use of nanofluids. Much attention has been given to the measurement and the development of models for the nanofluid’s thermal conductivity, viscosity and other properties and their effects on the heat transfer characteristics. The authors review some of the available nanofluids’ physical properties models and focus on presenting the various research work done on magnetohydrodynamic (MHD) convection of nanofluids in various geometries and applications [14–24].

It noted that more research needs are to be performed for the true understanding of the physical mechanisms at the nanoscale level due to the lack of agreement in some cases between experimental measurements and theoretical studies dealing with nanofluids. Accurate nanofluids’ property models based on experimental work are still needed and many factors still need to be explored and optimised and more promising potential applications need to be identified. In this context and in next works, we will introduce the effects of nanofluids’ characteristics through some nanofluids’ features, that is, taking into consideration the reports highlighting the synthesis, textural and morphological characterisation, and thermal stability of nanoparticles [25–32]. In this way, new correlation will be developed for the nanofluids’ attributes in terms of particle volumetric concentration and temperature measurements.

In addition, based on the fact that the introduction of nanoparticles deteriorates the heat transfer and decreases convection because of the high increase of fluid viscosity, semi-empirical equations will be suggested and discussed to correlating some parameters with some physico-chemical, thermophysical and temperature effects on some pure and mixed nanofluids, and electrolyte and non-electrolyte nanoparticles solutions [33–42].

Alumina (Al$_2$O$_3$) and copper are the most common and inexpensive nanoparticles used by many researchers in their numerical and analytical and experimental investigations [19,28]. All results have demonstrated the enhancement of the thermal conductivity by addition of nanoparticles. In most of the nanofluids’ research, some literature works have been performed using the single-phase and two-phase approaches. In contrast to single-phase methods, due to the complexity involved in
two-phase flow, using the two-phase modelling approach for nanofluid has been limited. Consequently, most of the numerical solutions have been done using a single-phase model [43–46]. Sundar and Sharma [47] obtained thermal conductivity enhancement of 24.6% with CuO nanofluid and 6.52% with Al₂O₃ nanofluid at 0.8% volume concentration compared to water. Sharma et al. [48] investigated experimentally the friction factor and the heat transfer coefficient in the transition flow with low volume concentration of nanofluid flowing in a circular tube and with twisted tape insert. The results showed a considerable enhancement of heat transfer coefficient by an amount of 23.7% when compared with water at a volume fraction of 0.1%. The thermal conductivities for certain base fluids and nanofluids were reported [49].

In this study, computational fluid dynamics (CFD) simulations are performed to predict heat transfer and laminar nanofluid flow characteristics in an MCHS. The calculated variation of wall temperature, friction factors and the pressure drop in the channel of a single-phase MCHS can well predict the experimental data. The results are presented for different flow rates of pure water, and nanofluids at different power inputs are also discussed in this work.

2. Formulation of the problem

2.1. MCHS model

A schematic diagram of the MCHS with key dimensions is shown in Figure 1. The fluid is flowing through a rectangular microchannel embedded in a test module. The dimensions of each...
The physical dimensions of the MCHS are presented in Table 1, whereas the thermophysical properties of the fluid and the solid nanoparticles are presented in Table 2.

### 2.2. Governing equations

For the specific case of heated flow through an MCHS, the conventional Navier–Stokes and energy equations are solved with the following assumptions:

1. Both the fluid flow and heat transfer are three-dimensional and at steady-state condition.
2. Fluid is a single phase and incompressible, and the flow is laminar.
3. Properties of both the fluid and the heat sink material are temperature-independent.
4. The gravitational force and radiation heat transfer are negligible.

The dimensionless governing equation forms which are used in present study for heated MCHS can be written as

Continuity: \[ \frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} + \frac{\partial W}{\partial Z} = 0 \] (1)

Momentum:

X-Momentum: \[ U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} + W \frac{\partial U}{\partial Z} = -\frac{\partial P}{\partial X} + \frac{1}{Re_{nf}} \left( \frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} + \frac{\partial^2 U}{\partial Z^2} \right) \] (2)

Y-Momentum: \[ U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} + W \frac{\partial V}{\partial Z} = -\frac{\partial P}{\partial Y} + \frac{1}{Re_{nf}} \left( \frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} + \frac{\partial^2 V}{\partial Z^2} \right) \] (3)

Z-Momentum: \[ U \frac{\partial W}{\partial X} + V \frac{\partial W}{\partial Y} + W \frac{\partial W}{\partial Z} = -\frac{\partial P}{\partial Z} + \frac{1}{Re_{nf}} \left( \frac{\partial^2 W}{\partial X^2} + \frac{\partial^2 W}{\partial Y^2} + \frac{\partial^2 W}{\partial Z^2} \right) \] (4)

Energy: \[ U \frac{\partial \Theta}{\partial X} + V \frac{\partial \Theta}{\partial Y} + W \frac{\partial \Theta}{\partial Z} = \frac{1}{Pe_{nf}} \left( \frac{\partial^2 \Theta}{\partial X^2} + \frac{\partial^2 \Theta}{\partial Y^2} + \frac{\partial^2 \Theta}{\partial Z^2} \right) \] (5)
Energy (solid regions): \[ \frac{\partial^2 \Theta}{\partial X^2} + \frac{\partial^2 \Theta}{\partial Y^2} + \frac{\partial^2 \Theta}{\partial Z^2} = 0 \] (6)

where \( \text{Pe}_{nf} = \frac{\text{Pr}_{nf} \mu_{in} D_h}{\mu_{nf}} \) and \( \text{Re}_{nf} = \frac{\rho_{nf} u_{in} D_h}{\mu_{nf}} \) are the Peclet number, the Reynolds number, and the Prandtl number, respectively.

In the aforementioned equations, the following dimensionless parameters are used:

\[
X = \frac{x}{D_h}; \quad Y = \frac{y}{D_h}; \quad Z = \frac{z}{D_h}
\]

\[
U = \frac{u}{u_{in}}; \quad V = \frac{v}{u_{in}}; \quad W = \frac{w}{u_{in}}
\]

\[
P = \frac{p}{\rho u_{in}^2}; \quad \Theta = \frac{T - T_{in}}{T_{wall} - T_{in}}
\]

The pressure drop is given as [50]:

\[ \Delta p = f \left( \frac{L}{D_h} \right) \left( \frac{\rho_{nf} u_{in}^2}{2} \right) \] (8)

where \( f \) is the Darcy friction factor and \( D_h \) the hydraulic diameter:

\[ D_h = \frac{4S}{P_{ch}} = \frac{2bd}{b + d} \] (9)

The heat flux to channel is given by Qu and Mudawar [9,10]

\[ q''_{ch} = \frac{q''_{bot} \times F}{d + 2b} \] (10)

The single phase local heat transfer coefficient along microchannel defined as

\[ h = \frac{q''_{ch}}{T_w - T_m} \] (11)

where \( T_w \) is the wall temperature and \( T_m \) is the mean fluid temperature evaluated by an energy balance around the channel.

2.3. Boundary conditions

In order to compare the simulation results to the existing experimental results, the flow is assumed fully developed at the channel outlet. The boundary condition of velocity inlet is assumed uniform in this simulation, and the velocity is zero along all other solid boundaries. All the surfaces of the heat sink exposed to the surroundings are assumed to be insulated except the bottom plate of the heat sink where a constant heat flux boundary condition simulating the heat generation from electronic chip is specified. Detailed description of boundary conditions in mathematical expressions is listed as follows:

At the inlet:

\[ U = 1; \quad \Theta = 1; \quad P = P_{in} \] (12)

At the outlet:

\[ P = P_{out}; \quad \frac{\partial \Theta}{\partial n} = 0 \] (13)
At the fluid–solid interface:

\[ U = 0 \ ; \ \Theta_s, \Gamma = \Theta_f, \Gamma ; \ -k_s \frac{\partial \Theta_s}{\partial n} \bigg|_\Gamma = -k_f \frac{\partial \Theta_f}{\partial n} \bigg|_\Gamma \]  

(14)

At the bottom plate (heating area):

\[ -k_s \frac{\partial \Theta_s}{\partial n} \bigg|_\Gamma = q''_{\text{bot}} \]  

(15)

At the top of the heat sink:

\[ U = 0 ; \ \frac{\partial \Theta_s}{\partial n} \bigg|_\Gamma = 0 \]  

(16)

### 2.4. Thermophysical properties of the nanofluid

In this study, the nanofluid is used as the working fluid. The physical properties of the nanofluid are calculated using the following equations [51–53]:

Density:

\[ \rho_{nf} = (1 - \varphi)\rho_f + \varphi\rho_s \]  

(17)

Heat capacity:

\[ (\rho C_p)_{nf} = (1 - \varphi)(\rho C_p)_f + \varphi(\rho C_p)_s \]  

(18)

In the present study, the empirical correlation for the thermal conductivity of Al₂O₃ nanofluid is specified in literature [54,55] as follows:

\[ \frac{k_{nf}}{k_f} = 1 + 64.7 \ \varphi^{0.746} \left( \frac{d_i}{d_s} \right)^{0.369} \left( \frac{k_s}{k_f} \right)^{0.7476} \ \text{Pr}^{0.9955} \ \text{Re}^{1.2321} \]  

(19)

where \( \text{Pr}_f \) is the Prandtl number of water and \( \text{Re}_f \) is the Reynolds number in Equation (19) or are defined as

\[ \text{Pr}_f = \frac{\mu_f}{\rho_f \alpha_f} \]  

(20)

\[ \text{Re}_f = \frac{\rho_f v_{Br} d_s}{\mu_f} = \frac{\rho k_B T}{3\pi\mu_f^2 l_f} \]  

(21)

where \( v_{Br} \) is the Brownian velocity of nanoparticles, \( l_f \) is mean-free path of water molecular (\( l_f = 0.17 \) nm), \( k_B \) is Boltzmann constant (\( k_B = 1.3807 \times 10^{-23} \) J/K) and \( \mu_f \) has been calculated by the following equation:

\[ \mu_f = 2.414 \times 10^{-5} \times 10^{\frac{2n-1}{2n+3}} \]  

(22)

The Hamilton–Crosser’s (H-C) [56] model of thermal conductivity of Cu–Water nanofluid introduces the model which holds for non-spherical particles and takes account the shape factor \( n \) as an experimental parameter for different types on nanoparticles which is given by

\[ k_{nf} = k[k_s + (n-1)k_f - (n-1)\varphi(k_f - k_s)][k_s + (n-1)k_f + \varphi(k_f - k_s)]^{-1} \]  

(23)

where \( n = 3/\psi \) is the empirical shape factor and \( \psi \) is the particle sphericity. Sphericity is the ratio of the surface area of a sphere with a volume equal to that of the particle to the surface area of the
particle; for a spherical particle \( (n = 3) \) and in that case the H-C model becomes identical to the Maxwell model \([57, 58]\). In this context, a correlation proposed by Maiga et al. \([59]\) for dynamic viscosity based on fitting curves through regression analysis of experimental data is used:

\[
\mu_{nf} = (123 \varphi^2 + 7.3 \varphi + 1) \mu_f
\]  

(24)

3. Numerical solution using CFD

A FLUENT CFD code was devised to solve the governing integral equations for the conservation of mass and momentum with the prescribed boundary conditions \([60]\). Also, for this case, the two heat transfer modes are coupled by the continuities of temperature and heat flux at the interface between the solid and fluid. The SIMPLEC algorithm \([61, 62]\) was chosen for the pressure–velocity coupling with a first-order upwind scheme for discretisation of convective transport terms. Double-precision solver is selected. The convergence criteria are set at \(10^{-6}\) for continuity and momentum equations, while for energy equation the convergence is set at \(10^{-8}\). The novelty in the present numerical work is that the governing equations are expressed in the dimensionless form, which dominates the convergence problems for complex geometry or for natural convection problem \([60, 62]\).

4. Validation and grid independence study

In CFD analysis, mesh quality has a significant impact on the solution time and accuracy as well as the rate of convergence. In order to check the sensitivity of the numerical results to mesh size, several different grid systems are used to test the grid independence of the solution. The regions near to the solid–fluid interface are meshed denser with successive ratio of 1.016. This implementation is expected to give more accurate results than equally meshed geometries because the velocity and the heat flux gradients are higher at the fluid–solid interface. On the other hand, the meshes where adiabatic or symmetry boundaries are present are coarser. Thereby, the solution is obtained for three non-uniform grid configurations corresponding to 187.900, 312.748 and

![Figure 2](image_url)

**Figure 2.** Comparison of the experimental data and numerical results \([64]\) for \( \text{Al}_2\text{O}_3/\text{H}_2\text{O} \) nanofluid \((\varphi = 1\%)\) with the present analysis.
Figure 3. Variation of heat transfer coefficient along MCHS of Al$_2$O$_3$/H$_2$O nanofluid at different power inputs for: (a) Re = 639; $\phi = 0\%$ (pure water); (b) Re = 582; $\phi = 1\%$; (c) Re = 532; $\phi = 2\%$. 
454,327 hexahedral elements. Beyond 312,748 elements, the numerically predicted results are grid independent.

In order to validate this CFD results obtained with use of the FLUENT commercial software, the predictions of the fluid flow of Al₂O₃/H₂O nanofluid (φ = 1%) in a smooth microtube with

Figure 4. Comparison of experimental [13] and the numerically predicted friction factor variations with Reynolds number of: (a) pure water; (b) 1% Al₂O₃/H₂O nanofluid; (c) 2% Al₂O₃/H₂O nanofluid.
Reynolds number equal to 600 is compared with numerical and experimental data [63]. The model validation was found to be in excellent agreement with experimental and numerical results from other previous studies (Figure 2). Therefore, numerical code is credible to predict the thermal and hydrodynamic behaviour of a nanofluid flow in an MCHS.

5. Results and discussion

5.1. Local heat transfer coefficient profile

Figure 3(a)–(c) shows the variation of the heat transfer coefficient along the length of the channel. There is a large drop in the value of the heat flux computed at the inlet of the MCHS, and then it reduces at a slow rate as the fluid moves towards the exit of the MCHS. This is due to the high temperature gradient at the inlet as the fluid temperature entering the MCHS is fixed at 303 K and as the fluid moves along the MCHS, the average bulk fluid temperature increases, thus reducing the heat transfer rate to the fluid. Increasing the heat flux has a very weak effect on the heat transfer coefficient for pure water, but an appreciable effect for the nanofluid. Notice also the stronger effect of heat flux for 2% Al$_2$O$_3$ in Figure 3(c), compared to 1% Al$_2$O$_3$ in Figure 3(b).

5.2. Friction factor profile

A comparison between measured [13] and predicted friction factor versus the Reynolds number is shown in Figure 4(a)–(c). These figures show that the friction factor is similar for all particle volume fractions where the friction factor decreases with the increase of Reynolds number. It can be stated that the increment in dynamic viscosity due to the presence of the nanoparticles in water only appears to give a slight rise in friction factor especially for low Reynolds number.

In Figure 5, the friction factor variation versus Reynolds number for various particle volume fractions is presented. It is observed that adding nanoparticles to the base fluid does not remarkably increase the friction factor. The friction factor tends to slightly augment with an increase in the volume concentration of the nanofluid.

![Friction factor variations versus Reynolds number for various particle volume fractions: $\phi = 0\%, 1\%, 2\%$.](image)
Figure 6. Comparison of the experimental [13] and the numerically predicted variations pressure drop profile with Reynolds of (a) pure water; (b) 1% Al₂O₃/H₂O nanofluid; (c) 2% Al₂O₃/H₂O nanofluid.
5.3. Pressure drop profile

A comparison between measured [13] and predicted pressure drop values with the Reynolds number is shown in Figure 6(a)–(c). It can be seen that the pressure drop rises linearly with the increase of the Reynolds number for various studied nanoparticle volume fractions because of increased viscosity of the nanofluids. Figure 6(a)–(c) shows that both the computational model and the experimental results can adequately predict the pressure drop across the MCHS.

The variations of pressure drop with the Reynolds number for various volume fractions of nanoparticles are shown in Figure 7. The results indicate that the pressure drop of the nanofluids increases with the Reynolds number. The calculated results showed 24% augmentation with increase in the volume fraction of nanofluids from 0% to 2%.

![Figure 7](image-url)

**Figure 7.** Variation of pressure drop with Reynolds number for various volume fractions.

![Figure 8](image-url)

**Figure 8.** Comparison of the experimental [13] and the numerically predicted wall temperature profile for pure water at mass flow rate of 2.15 g/s and two heat inputs of 100 W and 300 W.
5.4. Wall temperature profile

The variation of the wall temperature along the microchannel for different flow rates and different concentration level of nanoparticles is shown through Figures 8–13. Along the MCHS wall, the temperature increases as the fluid goes from the inlet to the outlet for all cases. The CFD results are also in good agreement with the experimental results [13].

The numerical results for the variation of wall temperature along MCHS for different flow rates and Al₂O₃ concentrations and heat inputs are compared with the experimental data of Lee and Mudawar [13]. Figure 14(a) and 14(b) shows a dependence of the wall temperature on the mass flow rate. In fact, for the same heat flux, the decrease of the mass flow rate increases the local wall temperature. Therefore, the only noticeable decrease in wall temperature is realised near the inlet for the lower mass flow rate at the lower heat input. As the heat flux is increased to 300 W, a
decrease in the slope of the temperatures is observed for the lower mass flow rate, which indicates an enhanced heat transfer rate. The comparison shows that CFD results can predict well the experimental data.

5.5. Heat transfer coefficient for various nanofluids

Figure 15 illustrates the variation of the heat transfer coefficient along the MCHS for two types of nanofluids. It is observed that the nanofluids-cooled MCHS could be able to enhance the heat transfer compared with pure water. That, with 2% concentration of Al$_2$O$_3$/water nanofluid, 14% increase in heat transfer coefficient was observed. Also, the heat transfer rate when using Cu-
Figure 13. Comparison of the experimental [13] and the numerically predicted wall temperature profile for 2% Al₂O₃/H₂O nanofluid at mass flow rate of 5.49 g/s and two heat inputs of 100 W and 300 W.

Figure 14. Variation of wall temperature along MCHS for different flow rates and Al₂O₃ concentrations and heat inputs of (a) 100 W and (b) 300 W.
water is greater by 4% compared with Al\(_2\)O\(_3\)–water which may be due to higher thermal conductivity of Cu–water.

### 5.6. Temperature and velocity contours

Figure 16 shows that the outlet temperature keeps on increasing as the heat flux augments because the nanofluid heats up more and more due to the convective heat transfer. Moreover, the shown temperature profiles are due to the assumption of hydrodynamic fully developed flow. Along the flow direction, the temperature rises in the fluid and the solid regions of the MCHS; so at the heated surface of the MCHS and in correspondence of the outlet, the highest temperature point is observed.
The velocity contours at two Reynolds numbers and two values of heat inputs using a nanofluid ($\varphi = 2\%$) as the coolant are shown in Figure 17. The figure shows that the entrance length increases with the increase of the Reynolds number.

6. Conclusion

The cooling efficiency of MCHS is numerically investigated using CFD simulations in this study. The results obtained were compared with the available experimental results, and the numerical simulation results showed good agreement with the experimental data. The MCHS performance was evaluated in terms of the temperature and velocity profile, friction factors, convective heat transfer coefficients, wall temperature and pressure drops. Based on the presented results, the following conclusions can be drawn:

- The MCHS wall temperature decreased with the increase of nanoparticles volume fraction. There was no apparent difference in the results between nanofluids and pure water at lower heat flux conditions.
- A small effect was noted for increased heat flow for pure water. Inversely, an increase in the nanoparticles volume fraction indicated a stronger effect of heat flux.
- The pressure drop of the nanofluids increased with the increase of Reynolds number. The calculated results showed the 24% augmentation with increase of the volume fraction of nanofluids from 0% to 2%.
- The presence of the nanoparticles in water flowing through the MCHS appeared to give only a slight rise in the friction factor values. As the Reynolds number increased, the friction factor decreased. At the entrance of the MCHS, a remarkable drop in the value of the heat flow was noted.
- The presence of nanoparticles substantially increased the heat transfer coefficient by increasing the heat flux. In the case of pure water, increasing the heat flux did not affect the heat transfer coefficient.
The calculated results showed that the heat transfer performance of Al₂O₃-water and Cu-water nanofluids was about 14–20% better than that of pure water.

Comparison of the numerical results with the experimental data available in the literature indicated that the hydrodynamic assumption, developed for fully laminar flow, was valid.

**Acknowledgements**

The authors thank Professor Mounir Baccar and Dr Wahid Massmoudi, National School of Engineers of Sfax, Tunisia, Mechanical Engineering Department and Head of the Research Unit, Computational Fluid Dynamics and Transfer Phenomena, for authorising us to use the commercial CFD code FLUENT.

**Disclosure statement**

No potential conflict of interest was reported by the authors.

**ORCID**

L. Snoussi [http://orcid.org/0000-0002-5136-9911](http://orcid.org/0000-0002-5136-9911)

N. Ouerfelli [http://orcid.org/0000-0002-8343-0510](http://orcid.org/0000-0002-8343-0510)

**Nomenclature**

- **L** Microchannel length (m)
- **b** Microchannel depth (m)
- **Cₚ** Specific heat capacity (J kg⁻¹ K⁻¹)
- **d** Microchannel width (m)
- **dᵢ** Molecular diameter of the base fluid (m)
- **dₛ** Nanoparticle diameter (m)
- **Dₜ** Hydraulic diameter (m)
- **F** Width of unit cell microchannel (m)
- **f** Friction factor
- **g** Gravitational acceleration (m s⁻²)
- **h** Heat transfer coefficient (W.m⁻².K⁻¹)
- **k** Thermal conductivity (W.m⁻².K⁻¹)
- **p** Pressure (Pa)
- **P** Dimensionless pressure
- **S** Channel flow area (m²)
- **Δp** Pressure drop (Pa)
- **Pᶜʰ** Channel wet perimeter (m)
- **Pₑ** Total electric power input
- **Pr** Prandtl number
- **Q** Total heat transfer (W)
- **qᵢ** Heat flux through heat sink base area (W/m²)
- **Re** Reynolds number, Re = \( \frac{Dₜ \times u \times \rho}{\mu} \)
- **u, v, w** Dimensional velocity component (m/s⁻¹)
- **U, V, W** Dimensionless velocity component
- **x, y, z** Cartesian coordinates (m)
- **X, Y, Z** Dimensionless coordinates

**Greek Symbols**

- **α** Thermal diffusivity (m²s⁻¹)
- **β** Thermal expansion coefficient (K⁻¹)
- **Θ** Dimensionless temperature
- **μ** Dynamic viscosity (kg.m⁻¹.s⁻¹)
- **φ** Volume fraction of nanofluids
- **ρ** Density (kg.m⁻³)
Kinematic viscosity ($m^2s^{-1}$)

**Subscripts**
- $f$: Fluid
- $s$: Solis
- $h$: Hot
- $c$: Cold
- $nf$: Nanofluid
- $bot$: bottom area
- $ch$: Channel
- $in$: inlet

**References**


