Numerical simulation of Al$_2$O$_3$ – water nanofluid effects on the performance of a cross flow micro heat exchanger

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Abstract. The numerical model investigated in this paper was implemented with Finite Element Method (FEM) based Comsol Multiphysics software, starting from a real cross-flow micro heat exchanger manufactured from 50 stainless steel plates for cold and hot fluid transfer. Each plate (foil) had 34 microchannels with rectangular cross-section, at a depth of 100 μm and a width of 200 μm. The working fluids were water and Al$_2$O$_3$-water nanofluids with volumetric concentrations of 0.5%, 1% and 1.5% in the temperature domain of 293 – 333 K. The oxide-based nanofluids are widely investigated in thermal energy transfer applications due to an increased heat transfer performance by comparing with pure water as a cooling fluid. Analytical models for the physical properties of water and nanofluid, referring to density, viscosity, thermal conductivity and specific heat capacity were used here. Microchannel bulk temperature profiles for cold and hot plates of the model with 1% Al$_2$O$_3$-water nanofluid were presented here in order to examine the uniformity of temperature distribution in the channel flow field area. Simulation results indicated that the total heat flow rate in the heat exchanger models increased with the increase in volume concentration of the nanoparticles at mass flow rates of 41, 61, 81, 101 and 150 Kg/h, but with the expense of a bigger pressure drop (or Fanning friction factor) at an increased pumping power required to drive flow through the channels.

1. Introduction

Due to an emerge of microscale devices that require cooling, registered especially in the last decade, microchannel heat exchangers started to be used in industries such as microelectronics, robotics, aerospace, biomedical[1-3], presenting here a heat transfer enhancement with increased heat flux dissipation. However, there are two factors that limit the heat transfer coefficients in a micro heat exchanger: the reduction in the channel dimensions was accompanied by higher pressure drop, and the amount of heat transfer was limited by the heat transfer fluid used.

Several studies demonstrated that nanofluids present a higher effective thermal conductivity than pure base fluids and therefore offer an optimal alternative for heat transfer enhancement [4-6]. In literature survey there are various experimental and numerical studies related to the thermal performance of nanofluids in microchannels. Kalteh et al. [7] experimentally and numerically investigated the convective heat transfer of nanofluids in a microchannel and the results showed that the Nusselt number (and heat transfer) increased with an increase in the particle volumetrical...
concentration and Reynolds number. Another experimental work conducted by using Al₂O₃–water nanofluid was studied by Jung et al. [8] in a microchannel and indicated that the use of the nanofluids caused a slight increase in the pressure drop, but this case was not remarkable with respect to the increase achieved in heat transfer.

C-S Jwo et al. [9] performed a multi-channel heat exchanger (MCHE) experiment using Al₂O₃–water nanofluids of different weight concentrations (0, 0.5, and 1.0 wt.%) and at different flow rate values. Results showed that the overall heat transfer coefficient ratio was higher at higher nanoparticle concentrations but when the mass flow rate was higher, a higher temperature does not provide greater enhancement of the overall heat transfer coefficient ratio.

Numerical investigations using different Computation Fluid Dynamics (CFD) techniques have been performed for microchannel heat exchanger modelling using nanofluids. H.A. Mohammed et al. [10] numerically studied heat transfer in a square microchannel heat exchanger using Finite Volume Method with hybrid differencing scheme and found that nanofluids enhanced the thermal properties and performance of the heat exchanger while slightly increasing the pressure drop and the increasing of the Reynolds number causes the pumping power to increase and the effectiveness to decrease. Pääkkönen et al. [11] performed a numerical evaluation of heat transfer boundary conditions for CFD modeling of heat exchanger geometries using Finite Element Model (FEM) based ANSYS Fluent software. For the flat plate geometry, the heat flux boundary condition was considered to be most suitable, being in good agreement with the results from the literature.

In this research article, with the help of FEM based Comsol Multiphysics software, we implemented numerically four micro heat exchanger models with different volumetric concentrations of Al₂O₃-water nanofluid used as a cooling agent: 0% (pure water), 0.5%, 1% and 1.5% in order to study the total heat flow rate variation at different mass flow rates in correlation with pressure drop, friction coefficient and pumping power.

2. Model set-up

2.1. Geometry of the computational domain and the discretisation network

In figure 1 are presented geometrical details regarding the basic micro-heat exchanger model. Because of existing symmetries in a micro heat exchanger unit, the computational was limited at only two layers. Therefore, the entire micro heat exchanger was modeled as a system of many cross-flow modules connected in parallel [12].

![Figure 1. Geometrical characteristics for the plate-type cross-flow micro heat exchanger model.](image-url)
The channels separated by a fin thickness of \( d = 100 \, \mu m \) have a width of \( c = 200 \, \mu m \) and a depth of \( b = 100 \, \mu m \). A single layer (foil) has the dimensions of 14 mm x 14 mm and a thickness \( t = 200 \, \mu m \). These foils have been stacked together in order to form the heat exchanger core, which led to 25 plates for cold fluid circulation and 25 plates for hot fluid flow. In each foil, the heat transfer take place in the central 10 mm x 10 mm area.

From a preliminary grid sensitivity test performed in order to establish an optimal mesh density that would yield sufficiently accurate results, we considered the Normal mesh discretization for our numerical models (see figure 2).

![Figure 2. Detail of normal mesh discretization used in the numerical simulations.](image)

### 2.2. Governing equations

Several assumptions were made on the operating conditions of the MCHE for the present numerical model [10]:

a. The MCHE operates under steady-state conditions.

b. The fluids remain in single phase along the channel and the flow is laminar.

c. Thermo-physical properties of the fluids and MCHE material are temperature-independent.

d. Flow misdistribution and external heat transfer effects are neglected.

e. The outer walls of the MCHE are considered insulated (adiabatic).

*Heat transfer in solid* interface from Comsol Multiphysics software resolved an equation governing heat transfer through convection and conduction in terms of temperature \( T \) for solid domain (stainless steel plates). The heat transfer equation is defined for this interface in the form of:

\[
\rho C_p u \nabla T - \nabla \cdot (k \nabla T) = Q
\]

(1)

where \( u \) is the velocity vector (m/s), \( \nabla T = T_{amb} - T \) is the temperature gradient, \( C_p = 500 \) J/Kg·K denotes the specific heat capacity of W316L stainless steel plates, \( k = 16.3 \) W/m·K is the thermal conductivity of the plates and \( \rho = 7990 \) Kg/m\(^3\) is the plates density (Kg/m\(^3\))[13]. \( Q \) is a source term representing the internally generated heat (W/m\(^3\)).

*Heat transfer in fluids* feature resolved an equation similar to (1) for water/nanofluid domain. An absolute pressure \( p_a = 1 \) atm and an initial temperature \( T = 293.15 \) K were considered in those two heat transfer features.

The first law of thermodynamics on the heat exchanger leads heat transfer rate expressed by the following equations:

\[
Q_h = \dot{m}_h C_p (T_{h,in} - T_{h,out})
\]

(2)

\[
Q_c = \dot{m}_c C_p (T_{c,out} - T_{c,in})
\]

(3)

where \( C_p \) denotes specific heat at constant pressure and \( T \) denotes temperature. Subscripts \( h, c, in, \) and \( out \) stand for hot passage, cold passage, inlet, and outlet of the microchannel heat exchanger, respectively. Inlet cold stream temperature \( T_{c,in} = 293 \) K and inlet hot stream temperature \( T_{h,in} = 333 \) K are considered for models.

The flow of heating water in the corrugated channel is described by the weakly compressible Navier-Stokes equations at steady-state [14]:
\[
\rho_f \cdot u \cdot \nabla u = \nabla \left[ -p I + \mu \left( \nabla u + (\nabla u)^T \right) - \frac{2}{3} \mu (\nabla \cdot u) I \right]
\]

\[
\nabla \cdot \left( \rho_f \cdot u \right) = 0
\]

(4)

where \( \rho_f \) denotes fluid density (kg/m\(^3\)), \( u \) represents the velocity (m/s), \( \mu \) denotes viscosity (kg/(m \cdot s)), \( I \) is identity matrix and \( p \) equals the pressure in the channels (Pa). The \textit{Laminar Flow} interface sets up and solves the Navier-Stokes equations and is here used to model the fluid flow in the channel.

Fluid flow through the hot/cold plate channels is characterized by the Reynolds number, calculated using the relation:

\[
Re = \frac{\rho_f U_0 D_h}{\mu}
\]

(5)

where hydraulic diameter is: \( D_h = \frac{2 \cdot b \cdot c}{b + c} \), \( \rho_f \) and \( \mu \) are the average fluid density and viscosity in the temperature domain considered and \( U_0 \) is the average inlet velocity (m/s).

The Fanning friction factor for cold and hot fluids was calculated using the following equation for multichannel arrangement [15]:

\[
f_{c,h} = \frac{\Delta p_{c,h} \cdot D_h}{2 L_{c,h} \cdot \rho_f \cdot U_0^2}
\]

(6)

were \( \Delta p_{c,h} \) represent the total pressure drop along the cold and hot channels, respectively and \( L_{c,h} \) is the channel length.

The pumping power \( P_p \) required to drive flow through the channel is [16]:

\[
P_p = \frac{1}{\eta_p \rho_f} \left( \dot{m}_c \Delta p_c + \dot{m}_h \Delta p_h \right)
\]

(7)

Assuming to have the same mass flow rates for both streams, equation (7) can be expressed as:

\[
P_p = \frac{\dot{m} \cdot \Delta p}{\eta_p \rho_f}
\]

(8)

were \( \eta_p = 0.7 \) is assumed for the pump efficiency[16] and \( \Delta p \) is total pressure drop in heat exchange unit containing 850 channels for hot water flow and 850 channels for cold water flow:

\[
\Delta p_c = \Delta p_{c,i} + \Delta p_{c,o} = (p_{ci} - p_{co}) + (p_{ci} - p_{co})
\]

(9)

where \( \Delta p_c \) and \( \Delta p_h \) are pressure drops of hot and cold sides, respectively, expressed as the difference between the channel inlet and outlet pressures.

2.3. \textit{Analytical models for physical properties of water and nanofluids}

Thermo-physical properties of water were obtained as polynomial functions of temperature, based on the tables with water properties at high temperature presented by Wagner et. al [17].

The water density is defined by:

\[
\rho_w(T) = 838.466135 + 1.40050603 \cdot T - 0.0030112376 \cdot T^2 + 3.71822313 \cdot 10^{-7} \cdot T^3
\]

(10)

Thermal conductivity of water is calculated from:

\[
k_w(T) = -0.869083936 + 0.00894880345 \cdot T - 1.58366345 \cdot 10^{-5} \cdot T^2 + 7.97543259 \cdot 10^{-9} \cdot T^3
\]

(11)

Water dynamic viscosity is expressed as:

\[
\mu_w(T) = 1.3799566804 - 0.021224019151 \cdot T + 1.3604562827 \cdot 10^{-4} \cdot T^2 - 4.645409319 \cdot 10^{-7} \cdot T^3 + 8.904273573 \cdot 10^{-10} \cdot T^4 - 9.0790692686 \cdot 10^{-13} \cdot T^5 + 3.8457331488 \cdot 10^{-16} \cdot T^6
\]

(12)

The specific heat capacity of water at constant pressure is evaluated from:
\[
CP_{nf}(T) = -0.869083936 + 0.00894880345 \cdot T - 1.58366345 \times 10^{-5} T^2 + 7.97543259 \times 10^{-9} \cdot T^3
\] (13)

The density of water- \(\text{Al}_2\text{O}_3\) nanofluid is based on the physical principle of the mixture rule, and can be represented as \(\text{[18]}\):

\[
\rho_{nf}(T) = (1 - \phi_s) \rho_{bf}(T) + \phi_s \rho_s
\] (14)

where \(nf, \ bf\) and \(s\) refer to the nanofluid, base fluid and solid nanoparticle, respectively and \(\Phi_s\) is the volume fraction of the nanoparticles.

Density measurements performed by Vajjha et al. \([19]\) on nanofluids containing \(\text{Al}_2\text{O}_3\) nanoparticles (with mean diameter of 44 nm) in water (as a base fluid) over a temperature range of 0 – 50°C for concentration range of 1 – 10% indicated a maximum deviation of 1.2% by comparing with results from equation (14).

Pak and Cho \([20]\) established the following correlation for the specific heat capacity of nanofluids, starting from liquid-particle mixture theory:

\[
CP_{nf}(T) = (1 - \phi_s) CP_{bf}(T) + \phi_s CP_s
\] (15)

Teng and Hung \([21]\) conducted experimental studies to measure the heat capacity and density of \(\text{Al}_2\text{O}_3 – \text{water}\) nanofluids with \(\text{Al}_2\text{O}_3\) average particle size of 20 nm and volume fractions of 0.5%, 1% and 1.5% in the temperature domain of 20 – 70°C. Density deviation was observed to be in the range of -1.5% to 0.06% by comparing with equation (14) and heat capacity predictions offered by equation (15) fits well with their experimental data (with a deviation from -0.07% to 5.88%).

Corcione \([22]\) proposed two empirical correlations for the nanofluid effective dynamic viscosity and thermal conductivity, based on a wide selection of experimental data available in literature, including \(\text{Al}_2\text{O}_3\) nanoparticles with diameters of 38.4 nm, 47 nm and 80 nm suspended in water. The best fit of regression analysis for dynamic viscosity, with 1.84% deviation was:

\[
\mu_{nf}(T) = \frac{\mu_{bf}(T)}{1 - 34.87 \left( \frac{d_s}{d_{bf}} \right)^{-0.3} \cdot \phi_s^{1.03}}
\] (16)

were \(d_{bf} = 0.384\) nm is the equivalent diameter of water molecule; for the \(\text{Al}_2\text{O}_3\) it was selected a particle diameter \(d_s = 44\) nm. This equation is applicable for volume concentrations up to 7.1% in the temperature range of 293 – 333 K.

For nanofluid thermal conductivity, the empirical correlation with a mean deviation of 1.86% was established as \([22]\):

\[
k_{nf}(T) = k_{bf}(T) \left[ 1 + 4.4 \cdot Re^{0.4} \cdot Pr_{bf}^{0.66} \cdot \left( \frac{T}{273} \right)^{10} \cdot \left( \frac{k_s}{k_{bf}} \right)^{0.03} \cdot \phi_s^{0.66} \right]
\] (17)

were nanoparticle Reynolds number is: \(Re = \frac{2 \rho_{bf} k_{bf} T}{\pi \mu_{bf} d_s}\) and Prandl number of the base fluid is:

\[
Pr_{bf} = \frac{CP_{bf} \mu_{bf}}{k_{bf}}. \ Re\ and\ Pr_{bf}\ were\ calculated\ considering\ \rho_{bf} = 997\ \text{m}^3/\text{Kg},\ \mu_{bf} = 0.855 \times 10^{-3} \text{Pa} \cdot \text{s},\ CP_{bf} = 4170\ \text{J}/\text{Kg} \cdot \text{K} and\ k_{bf} = 0.613\ \text{W}/\text{m} \cdot \text{K}\ for\ \text{water} and\ k_s = 42.34\ \text{W}/\text{m} \cdot \text{K}\ for\ \text{Al}_2\text{O}_3\ nanoparticles\ at\ T = 300K\ [23].
\]

Equation (17), applicable for volume concentrations up to 9% was considered also in this paper in the temperature range of 293 – 333 K.

2.4. Boundary conditions

In the case of Laminar Flow interface, the following conditions are considered:

- “Wall” type condition: \(u = 0\) for upper and lower boundary of the cold and hot channels, with “no slip” specification for the walls.
- “Inlet” type condition for cold and hot channels inlet boundary, $L_{entr} = 0.1 \text{ mm}$ and $u = U_0$:

$$L_{entr} \nabla \left[ -pI + \mu \left( \nabla u + \left( \nabla u \right)^T \right) - \frac{2}{3} \mu \left( \nabla \cdot u \right) I \right] \cdot n = -p_{entr}n \quad (18)$$

- “Outlet” type condition for channels outlet boundary, with $p_0 = 0 \text{ Pa}$ and selecting “suppress backflow” option:

$$\left[ -pI + \mu \left( \nabla u + \left( \nabla u \right)^T \right) - \frac{2}{3} \mu \left( \nabla \cdot u \right) I \right] \cdot n = -\hat{p}_0n, \quad \hat{p}_0 \leq p_0 \quad (19)$$

Regarding to boundary conditions in Heat Transfer in Fluids interface, it was used “Outflow” condition: $-n \cdot q = 0$ for channels outlet, “Temperature 1” condition: $T_0 = T_{hot} (333 \text{ K})$ for hot channels inlet and “Temperature 2” condition: $T_0 = T_{cold} (293 \text{ K})$ for cold channels inlet. “Thermal insulation” condition: $-n \cdot q = 0$ was applied to external boundaries of computational domain.

3. Results and discussions

The present numerical model is validated (see figure 3) by comparing the results of FEM simulations with experimental data obtained by Brandner et al. [24] and with other numerical data which take into consideration the mass flow maldistribution along the channels of a real micro-heat exchanger [16]. Comparisons are carried out with reference to the following mass flow rates (assumed to be the same for both streams): $\dot{m} = 41, 61, 81, 101, 150 \text{ kg/h}$. The working fluid was water and the inlet temperatures for the cold and hot streams were $T_{cold} = 283 \text{ K}$ and $T_{hot} = 378 \text{ K}$, respectively [24].

Computational domain represents a repetitive block of the micro heat exchanger and the total heat flow rate $q$ (kW) in the thermal device was obtained by multiplying the numerically computed total net heat rate (W) at the boundary between the hot and the cold layers in a flow block by the number of blocks (25 in this case).

![Figure 3. Comparison of the present model with other numerical data and with experimental data.](image)

In figure 4a was presented the variation of the Reynolds number with mass flow rate for each of the models with different type of fluids, in order to investigate the influence of average density/viscosity ratio on fluid flow intensity. At highest mass flow rate of 150 Kg/h, a 2.5% decrease of the Reynolds number from one model to another was observed, due to a higher density and viscosity under the presence of $\text{Al}_2\text{O}_3$ nanoparticles with increased volumetric concentration.

From figure 4.b we could see that the total heat flow rate $q$ increased with the increasing of $\text{Al}_2\text{O}_3$ concentration in the nanofluid, but only in a small degree. For example, in the case of 1.5% $\text{Al}_2\text{O}_3$ – water nanofluid, $q$ increased with 1.8% at $m = 41\text{ kg/h}$, 2.1% at $m = 61\text{ kg/h}$, 2.04% at $m = 81\text{ kg/h}$, 1.97% at $m = 101\text{ kg/h}$ and 1.85% at $m = 151\text{ kg/h}$.
Figure 4. (a) Variation of Reynolds number and (b) total heat flow rate values at different mass flow rates for each of the four models investigated.

Pressure drop along cold and cold channels increased with increasing of Al$_2$O$_3$ volumetric concentration in the working fluid at different mass flow rates, as we could notice in figure 5.a. At $m = 41$ kg/h and $m = 61$ kg/h it was registered the lowest pressure enhancement, of 18.5% and 11.5%, respectively. The results presented in figure 5.b showed that the friction factor increased with the increase in volume concentration of the nanoparticles for a given mass flow rate and decreases with increase in Reynolds number. A minimum Fanning friction coefficient was identified for cold channels of the 1.5% Al$_2$O$_3$-water nanofluid model at $m = 61$ kg/h, with an increase of only 2.6% by comparing with pure water.

Figure 5. Comparison of: (a) pressure drop and (b) Fanning friction coefficients for hot and cold channels of the micro-heat exchanger models.

C. Nonino and S. Savino reported that the lack of uniformity in the microchannel temperature distributions of the same layer can cause flow maldistribution in streams of fluid with temperature dependent viscosity [16]. Next, we investigated the uniformity degree of the bulk temperature profiles along the micro-channels of the hot and cold layers for the 1.5% Al$_2$O$_3$-water nanofluid model at three mass flow rates (see figure 6). As we can observe in figure 6.a, figure 6.b and figure 6.e, the variation of the temperature along the cold channels became less smoother with the increasing of mass flow rate and the temperature difference between adjacent channels was in the same time reduced for both type of channels (cold and hot). Model M2 presented a temperature variation of 10 K in the central portion of the hot channels (between 4 and 10 mm along the y coordinate), with 2 K lower that in model M1.
Figure 6. Microchannel bulk temperature profiles for cold and hot layers in the case of model with 1.5% Al₂O₃-water nanofluid at \( \dot{m} = 41 \text{ Kg/h} \) (a, b), \( \dot{m} = 61 \text{ Kg/h} \) (c, d) and at \( \dot{m} = 150 \text{ Kg/h} \) (e, f).

In order to illustrate the characteristics of the modelled thermal fields inside the cross flow micro-heat exchanger, temperature gradients along the top and bottom planes of computational domain (in XY coordinates) were presented in figure 7 for the model with 1% Al₂O₃-water nanofluid, at three different fluid flow regimes.

Figure 7. Temperature maps on top and bottom plane of computational domain for the model with 1% Al₂O₃-water nanofluid at three different mass flow rates.

We could notice from here that at high mass flow rates, the temperature gradient was quite uniformly distributed along the X direction for the cold channels and along the Y direction for hot channels (see figure 6.c). A gradual distribution of temperature in the lower left part of the flow filed area, from 334 K to about 305-310 K, was observed for the hot channels layer in figure 7.a.
4. Conclusions
Numerical analysis of the cross-flow micro-heat exchanger model revealed that friction factor and pressure drop increases with the increase in particle volume concentration and with the mass flow rate m grow, the biggest enhancement being registered for the cold channels at high flow rates of 101 and 151 kg/h.
At $m = 61$ kg/h, the pressure drop and Fanning friction factor presented an increase of 6.7% and 1.8%, respectively for fluid flow in cold channels with 1.5% $\text{Al}_2\text{O}_3$ –water nanofluid, by comparing with pure water as a cooling agent.
Total heat flow rate was higher at higher nanoparticle concentrations, when the probability of collision between nanoparticles and the walls of heat exchanger channels increased. Only a maximum total heat flow rate enhancement of 2.1% at $m = 61$ kg/h for 1.5% $\text{Al}_2\text{O}_3$ –water nanofluid was obtained due to a short mean free path length of nanoparticle collisions in microchannels, causing in this way a reduced energy transfer through channel walls.

5. References
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