

INFLUENCE OF TYPES OF NANOPARTICLES, NANOPARTICLES VOLUME CONCENTRATION AND TYPES OF COOLER METALS ON THE HEAT TRANSFER IN A MINI-CHANNEL COOLER

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Abstract

In the present work, we have studied the effect of three different types of nanoparticles, nanoparticles volume concentration and types of cooler metals on heat transfer in a mini channel cooler numerically. In these simulations, we have considered the Cu-H₂O, the Ag-H₂O and the Diamond-H₂O with different volume fractions in the range of 0,02%-0,1% and for two types of cooler materials for cooling an electronic component. In these conditions, the inlet velocity is constant for the three different types of nano-fluids. The power of the electronic component is equal to 130 W. The numerical results are developed for a Reynolds number equal to 1414 and a steady-state.

The simulation was performed using commercial software, ANSYS-Fluent 15.0. The obtained results show that the average heat transfer coefficient increases with the increase of the volume fraction of the nanoparticles (Cu, Ag, Diamond) and with the decrease of the temperature of the electronic component. In these conditions, the average heat transfer coefficient is the highest for the H₂O–diamond nanofluid compared with the other nanofluids the Cu-H₂O and the Ag-H₂O. Furthermore, the types of cooler metals have considerable effects on the amelioration of the temperature of the electronic components.

Keywords: heat transfer; nanofluid; mini channels; cooler; fluent.

Introduction

Heat transfer is a process of great importance in the field of industry and technology, particularly in the field of cooling the electronic components by micro-

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channels to ameliorate the heat exchanger between the coolant and the walls of the cooler and for a discount of temperature gradients in the electronic components. The amelioration of the heat transfer is directly related to several factors such as the thermo-physical properties of the used materials, the volume concentration of fluid, the types of nanoparticles and the flow velocity. In this context, *Abd Elazem et al.* [1] studied the effect of partial slip boundary conditions on the mass transfer and heat of the Ag-water and Cu-water nanofluids over a stretching sheet in the presence of radiation and magnetic field.

Also, among the works in this field and especially with the recent advances in the use of nanomaterials in improving heat transfer performance, *Kalteh* [2] studied the influence of various nine different nanoparticles (Al_2O_3 , CuO, Cu, Fe, Au, Ag, TiO_2 , SiO_2 , and Diamond) and three different base liquid types (ethylene glycol, water and engine oil) on the nanofluids heat and liquid flow in a microchannel numerically. Thermal transfer and pressure drop of different types of nano-fluid are compared at Reynolds number equal to $Re = 100$ and 1% of volume fraction for all nanoparticles and constant inlet flow velocity for different base fluids. The results show that the thermal transfer coefficient is the lowest for the $\text{H}_2\text{O}-\text{SiO}_2$ nano-fluid and is the highest for the H_2O -diamond. A numerical and experimental study of a cooling system for power electronics based on the setting in motion of a fluid metallic confirmed the importance of the choice of the material constituting the cooler with the advantages brought by liquid metal [4]. *Rehena et al.* [3] studied the effect of nanofluids on the heat transfer and cooling system of the photovoltaic thermal (PVT) performance. They concluded that the photovoltaic thermal system worked by nanofluid is more effective than the water-based photovoltaic thermal system. They proved that the heat transfer could be enhanced by 34.2% with the increase of the mass concentration of nanoparticle at the largest cavity ratio and the lowest heating power. They also indicated that the thermal conductivities of $\text{TiO}_2\text{-H}_2\text{O}$ nanofluids could be improved by 5.23%. Moreover, *Gülbanu et al.* [6] numerically investigated the convective heat transfer of two different nanofluids (Al_2O_3 -water and TiO_2 -water nanofluids) through square cross-sectional duct under constant heat flux ($500 \cdot 10^3 \text{ W/m}^2$) in Reynolds number between $Re=3,000$ and $Re= 100,000$. It was concluded that increasing in Reynolds number and the solid volume fraction increases the Nusselt number and pressure drop. In this context, *Fernandoa et al.* [7] found that the higher volume concentrations of Al_2O_3 nanoparticles can achieve higher heat transfer rates.

Balla et al. [8] studied the heat transfer coefficient for $\text{H}_2\text{O}-\text{Al}_2\text{O}_3$, $\text{H}_2\text{O}-\text{CuO}$ and $\text{H}_2\text{O}-\text{TiO}_2$ nanofluids. They concluded that the Nusselt number and heat exchanger coefficient increases with the increase of the volume concentration of nanoparticles. Also, they confirmed that for each investigated fraction value, the heat transfer coefficient is higher for the highest inlet velocity. *Chavda* [9] studied experimentally the effect of various concentrations (0.003%, 0.002%, and 0.004%) of nanoparticle (CuO) mixed in base fluid (water) on heat exchange performance of double pipe thermal exchanger for counter and parallel liquid flow arrangement. The results show that the heat transfer coefficient increases with the increases in the volume fraction of CuO nanoparticle. *Ciloglu et al.* [10] studied the effect of nanoparticle types (silica SiO_2 , alumina Al_2O_3 , titania TiO_2 and copper oxide CuO) on the quenching process with 0.1% particle of volume concentration experimentally. They mentioned that the type of nanoparticle used in nanofluids substantially influenced the cooling, particularly with SiO_2 nanoparticles. *Kannadasan et al.* [11] studied experimentally the effect of CuO- H_2O nanofluids of 0.1%

and 0.2% of volume fraction on the heat exchanger and pressure drop features of helically coiled thermal exchanger held in vertical and horizontal positions. They concluded that, with an increase in the volume fraction of nanoparticle, the Nusselt number increases at turbulent flow. Also, *Kangude et al.* [12] studied the effect of nanoparticles on a single bubble-based nucleate pool boiling experimentally. They used water-silica nanofluids with different concentrations of nanoparticles (0.005% and 0.01%). Among the experimental results, they found that the nanoparticles suspended tend to spread the strength of temperature gradients. *Gherasim et al.* [13] investigated the heat transfer performance of coolants with $\text{Al}_2\text{O}_3\text{-H}_2\text{O}$ nanofluid inside a confined impinging jet cooling system. They noticed that the average Nusselt number increases with an increasing volume concentration of nanoparticle and inlet velocity of nanofluid. However, it decreases with an increase in disk spacing. *Chemloul et Belmiloud* [14] studied the influence of nanofluid type, the variation of the Rayleigh number and the volume concentration of the nanoparticles (TiO_2 ; Cu and Al_2O_3) on the heat transfer performance in a square cavity. They concluded that the thermal transfer increases with the volume concentration and the Rayleigh number. Also, they confirmed that the improvement of the higher heat exchanger is achieved by using Cu nanoparticles. *Khanafer et al.* [15] studied the heat transfer numerically by natural convection of nanofluids in a 2D enclosure. They indicated that the average number of Nusselt increases with the volume concentration of the nanoparticles for various numbers of Grashof. *Khaleduzzaman et al.* [16] presented an analytical study of the thermal performance improvement of three nano-fluids ($\text{Al}_2\text{O}_3\text{-H}_2\text{O}$, $\text{CuO-H}_2\text{O}$, $\text{SiC-H}_2\text{O}$) for a copper rectangular microchannel heat sink for electronic device cooling. They concluded that $\text{CuO-H}_2\text{O}$ nano-fluid is the most appropriate for the cooling of electronics components among three nano-fluids. *Mostafa et al.* [17] studied the heat transfer coefficient in the mini-channel heat sink containing SiC-water and $\text{TiO}_2\text{-water}$ as nanofluids using ANSYS-fluent. They observed that the heat transfer coefficient increases with increasing the volume fraction and the number of Reynolds. *Belahmadi et al.* [18] studied numerically by using Fluent software and a simple algorithm the entropy generation, and the heat transfer of a $\text{Cu-H}_2\text{O}$ nano-fluid in a vertical channel. Their results showed that the increase of Grashof and Reynolds numbers and volume concentration of nanoparticles reduces the entropy generation and improves the heat transfer. In addition, *Mohammed et al.* [19] studied the effect of using various types of base liquids (water, oil, ethylene glycol (EG) and glycerin with 2% volume concentration of diamond nanoparticle) numerically and also as four different types of substrate materials (aluminum, titanium, steel and copper) on a heat exchanger trapezoidal micro-channels heatsink. They found that the heat exchanger performance of water-base nano-fluid can be improved in the steel micro-channel heat-sink substrate. Also, they found that the water-base nanofluid cooled microchannel has the highest value of temperature while the glycerin-base nano-fluid cooled microchannel heat sink has the smallest value. The averaged heat transfer coefficient for other types of base liquids increased with the increase of the Prandtl number. *Saeed and Kim* [20] investigated numerically and experimentally the heat transfer enhancement characteristics using four different channel configurations of a mini-channel heat sink and with three different volume fractions of nanoparticles Al_2O_3 in base fluid (water). They observed an enhancement of 24.9%, 27.6% and 31.1% in the heat transfer coefficient of the heat sink with fin spacing of 1,5 mm, 1 mm and 0,5 mm, respectively. They also observed that the enhancement factor increases by dreading the

fin spacing of the flow channel at the same value of volume fraction and rate of coolant flow. *Abedini et al.* [21] numerically investigated the mixed convection of different nanofluids (Al_2O_3 -water, TiO_2 -water, Cu-water and silver-water nanofluids) in the horizontal annulus. They studied the effect of different parameters such as Richardson, Rayleigh, the volume fraction of nanoparticles and Reynolds number on heat transfer. They indicated that the thermophysical properties of nanoparticles have a direct effect on heat transfer. According to these anterior studies, it is clear that the use of the nanoparticles is very crucial to enhance the heat transfer characteristics.

The objective of this paper is to study the influence of nanoparticles types, the volume concentration of nanoparticles, and cooler materials types on the heat transfer in a proposed mini channel cooler. Particularly, we have chosen two types of cooler materials: the aluminum cooler and the copper cooler.

Thermo-physical properties of the nano-fluid

The calculation of the thermophysical properties of the nanofluids used in our work is considered in this section.

The effective thermal conductivity of the nanofluid is approximated by *Maxwell-Granetts* [22] as follows:

$$K_{nf} = \frac{K_s + 2K_f - 2\phi(K_f - K_s)}{K_s + 2K_f + \phi(K_f - K_s)} K_f \quad 1$$

The dynamic viscosity is approximated by the Brinkman model [22] as follows:

$$\mu_{nf} = \frac{\mu_f}{(1 - \phi)^{2.5}} \quad 2$$

The density of the nanofluid is given as [15]:

$$\rho_{nf} = (1 - \phi)\rho_f + \phi\rho_s \quad 3$$

The heat capacitance of the nanofluid is expressed as given by *Khanafar et al.*[15]:

$$(\rho C_p)_{nf} = (1 - \phi)(\rho C_p)_f + \phi(\rho C_p)_s \quad 4$$

The thermal expansion coefficient of the nanofluid is expressed as given by *Khanafar et al.* [15]:

$$\rho_{nf}\beta_{nf} = (1 - \phi)\rho_f\beta_f + \phi\rho_s\beta_s \quad 5$$

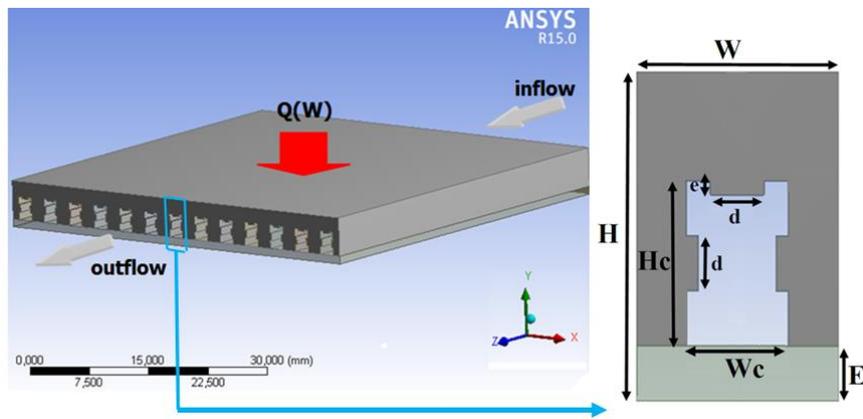
The thermophysical properties of the pure fluid (water) and nanoparticles are grouped in Table 1 [23].

Table 1. Thermo-physical properties of the pure fluid(water) and nanoparticles.

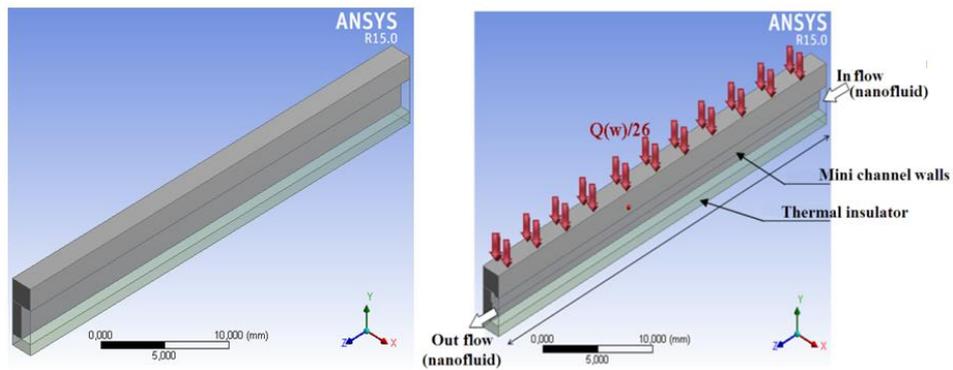
Material	ρ	Cp	K	μ
Water	997	4187	0.613	0.00085
Cu nanoparticles	8933	385	401	-
Ag nanoparticles	10500	235	429	-
Diamond nanoparticles	3500	509	2300	-

Geometrical system

Figure 1 shows the geometry of the cooler mini channels. The dimensions of this cooler are the same that considered by Tawk et al. [4].



(a) Geometrical arrangements of a mini channel



(b) Half a mini channel. (c) Boundary conditions

Fig. 1. CAD model of the cooler mini channels.

Table 2. Dimensions of a unit cell of cooler mini channels (in mm).

H	W	L	E	H _c	W _c	d	e	D _h
6	4	42	1	3	2	1	0,25	1,826

The difference is in the dimensions and shape of mini channels. The 13 channels and 12 fins form this cooler. Indeed, we assume that the bottom of the cooler contains a thermal insulator and that the upper surface of the cooler has an electronic component with a constant value. In these conditions, the thermal insulation is considered on all the outside faces of the cooler.

Due to the symmetry, we have considered only half of the fin and the half of the mini-channel, using the commercial software ANSYS 15.0 Fluent.

Mathematical formulation

In this study, we have assumed that the liquid flow is stationary. Three nanofluids consisting of Cu-H₂O, diamond-H₂O, and Ag-H₂O are used. These nanofluids are supposed incompressible and the thermo-physical characteristics are constant, except for the variation of the density, which is estimated by the Boussinesq hypothesis. The heat transfer by radiation is negligible and the impact of body force and viscosity dissipation is neglected.

The boundary conditions at the inlet are written as follows:

$$u = v = \quad \quad \quad 6$$

$$w = w_{in} = constant \quad \quad \quad 7$$

$$T = T_{in} = constant \quad \quad \quad 8$$

At the outlet, we can write:

$$P = P_{out} = 0 \quad \quad \quad 9$$

At the walls of the mini channel, we can write:

$$u = v = w = 0 \quad \quad \quad 10$$

$$T = T_{wall} \quad \quad \quad 11$$

At the solid/nano-fluid interface, the continuity of the flux can be written as follows:

$$K_s \frac{\partial T}{\partial n} \Big|_w = K_{nf} \frac{\partial T}{\partial n} \Big|_w \quad \quad \quad 12$$

At the superior face of the cooler, the power of the electronic component is equal to 130 W

The numerical model is governed by the equations of conservation of mass (13), the momentum equations (14), (15) and (16), the energy conservation equation (17) and the equation of the heat conduction in the solid (18) [24].

The continuity equation is written as follows:

$$u \frac{\partial u}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial w}{\partial z} = 0 \quad 13$$

The momentum equations along the x-axis, the y-axis, and the z-axis are written as follows:

$$u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} + w \frac{\partial u}{\partial z} = \frac{1}{\rho_{nf}} \left[-\frac{\partial p}{\partial x} + \mu_{nf} \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2} \right) \right] \quad 14$$

$$u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} + w \frac{\partial v}{\partial z} = \frac{1}{\rho_{nf}} \left[-\frac{\partial p}{\partial x} + \mu_{nf} \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2} \right) + (\rho\beta)_{nf} g(T - T_0) \right] \quad 15$$

$$u \frac{\partial w}{\partial x} + v \frac{\partial w}{\partial y} + w \frac{\partial w}{\partial z} = \frac{1}{\rho_{nf}} \left[-\frac{\partial p}{\partial x} + \mu_{nf} \left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2} \right) \right] \quad 16$$

The energy equation is written as follows:

$$u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} + w \frac{\partial T}{\partial z} = \alpha_{nf} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) \quad 17$$

The heat conduction through the solid wall written as follows:

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = 0$$

The hydraulic diameter is defined as the ratio of a cross-sectional area over the wetted parameter and it is calculated as follows [25]:

$$D_h = \frac{4A_c}{P_c} \quad 19$$

The average heat transfer coefficient of the micro-channel is calculated by the following equation [26]:

$$h_{ave} = \frac{Q}{NA_w(T_w - T_m)} \quad 20$$

The Reynolds number is defined as follows [25]:

$$Re = \frac{\rho_{nf} w D_h}{\mu_{nf}} \quad 21$$

Validation and comparison of the results

Figure 2 presents the variation of the surface temperature according to the different values of mass flow. These results are compared with the experimental and numerical results of *Tawk et al.* [4].

The excellent agreement revealed in this comparison confirms the validity of our numerical method.

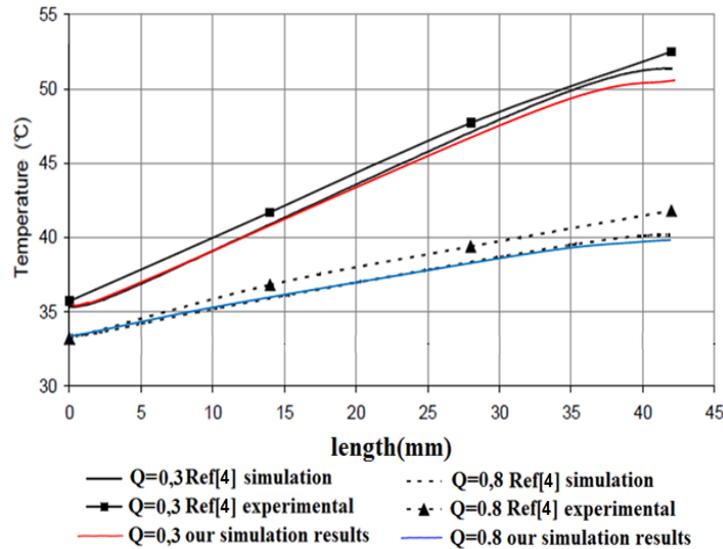


Fig. 2. Variation of the surface temperature for different values of mass flow.

Figure 3 represents a comparison between our numerical results and the results of *Tawk et al.* [4] in terms of the evolution of the surface temperature of the cooler at a dissipated power equal to 200 W and volume flow equal to 0.8 l/min, and for different cases. Case 1 corresponds to the molybdenum cooler and the coolant is the gallium. This case was studied also in the literature [4]. Case 2 corresponds to the molybdenum cooler and the coolant is the Cu-water nanofluid with a volume concentration equal to 0.7%.

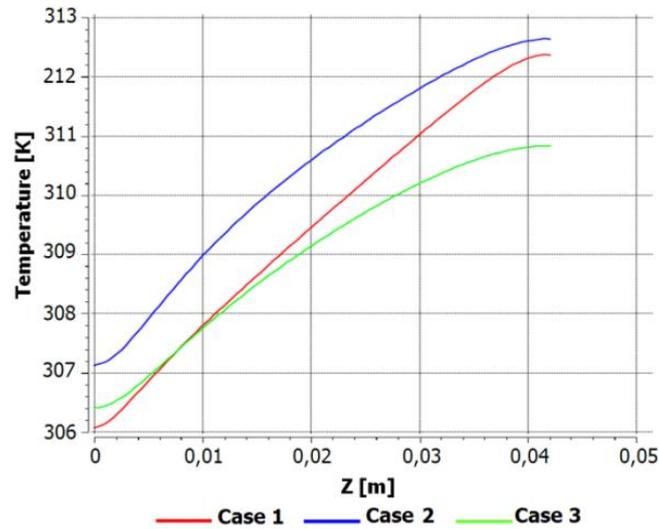


Fig. 3. Comparison of the surface temperature for three cases.

In case 3, we consider in the copper cooler and the coolant is the Cu-H₂O nanofluid with a volume fraction equal to 0.7%. From these results, we find that the temperatures of the upper surface of the cooler in case 3 are less over than the case 1 and case 2. This difference is due to the physical properties and particularly to the value of the thermal conductivity of the metal of the cooler. Also, it can be explained by the type and concentration of the coolant.

Results and discussions

Variation of the temperature of the mini-channels cooler for the three nano-fluids used and two metals of the cooler

Figure 4 shows the variation of the temperature of the upper surface of the mini-channels cooler along the plane of symmetry for the three nano-fluids used and for two metals of the cooler. From these results, it has been observed that the temperature value increases in the three types of nano-fluids. According to these results, the values of the temperature of the upper surface of the copper cooler are lower than those of the aluminum cooler and especially with the coolant H₂O–diamond nanofluid.

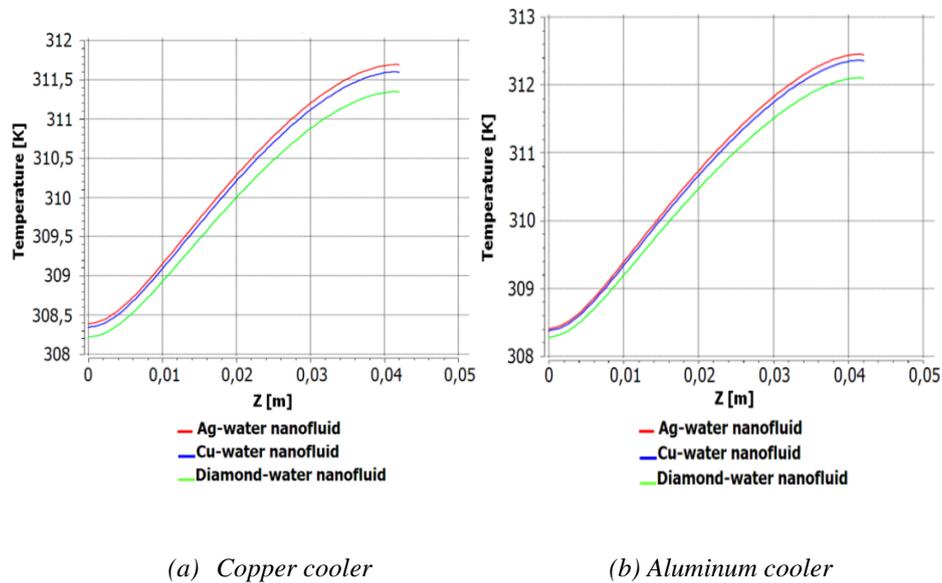


Fig. 4 Profiles of the temperatures for the three nano-fluids and two cooler metals at $Re = 1414$ and $\phi = 0,02$.

Effects of three different types of nano-particles on the temperature of the electronic component and the average heat transfer coefficient

Figures 5 and 6 illustrate the effect of different types of nano-particles and volume fraction of nanoparticles (Cu, Ag, Diamond) on the maximum temperature of the electronic component and the average heat transfer coefficient for the value of the Reynolds number equal to $Re = 1414$. From these results, it is clear that adding a low volume concentration of nanoparticles for the base fluid (water) leads to a significant decrease in the maximal temperature of the electronic component with an increase in the heat transfer coefficient. From figure 6, the results show that among the three used liquids, the liquid containing the diamond nanoparticle and water is the best in terms of the heat transfer coefficient. In these conditions, the electronic component temperature is also reduced. This fact is attributed to the convection facilitated by the higher thermal conductivities of the diamond-water nanofluid compared to that of the Cu-water and Ag-water nanofluid.

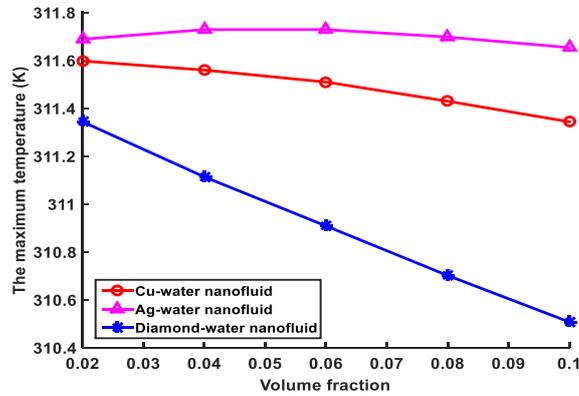


Fig. 5. Variation of the maximum temperature vs. the volume fraction.

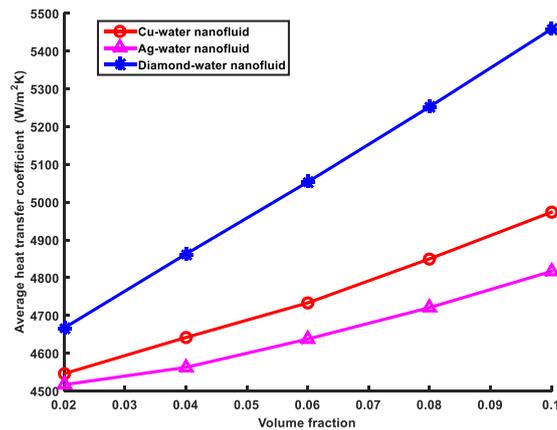


Fig.6. Variation of the average heat transfer coefficient vs. the volume fraction.

Evolution of the temperature of the electronic component for the two metals

Figure 7 shows the evolution of the maximum temperature value of the electronic component in the function of the volume fraction of Cu-H₂O nanofluid for the two metals. From these results, it is clear that the maximum value of the temperature decreases substantially for the two metals when the Reynolds number is constant. Indeed, the temperature value of the electronic component in the copper cooler is higher than the aluminum cooler. The copper cooler allows us to better cool the component with the increase in the volume concentration of Cu-water nanofluid.

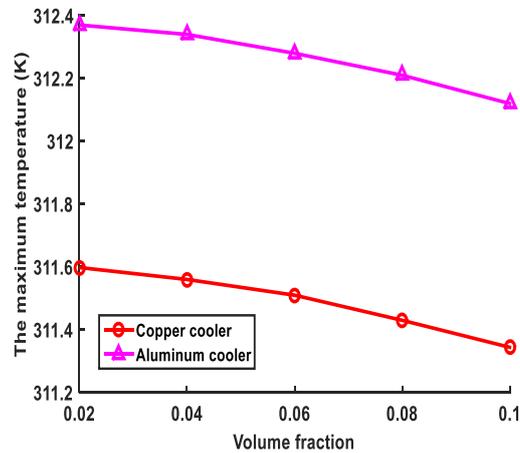
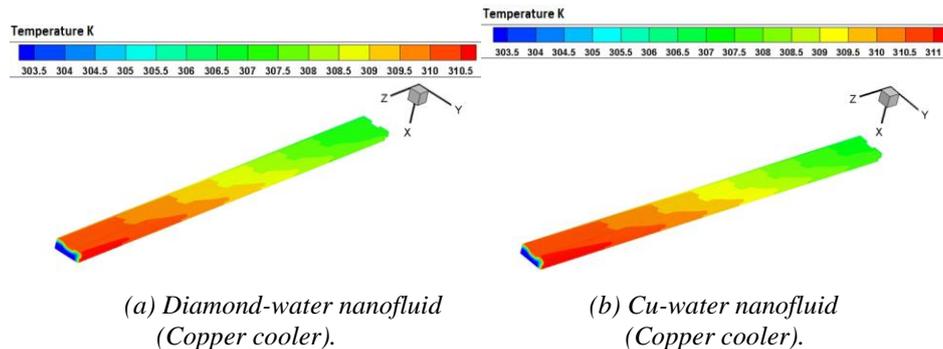


Fig. 7. Variation of the maximum temperature vs. the volume fraction.

Distribution of the temperature of the mini channel for three nanofluid and tow metals of the cooler

Figure 8 shows that for a Reynolds number equal to $Re = 1414$, and an imposed power of the component equal to 130 W, the distribution of the temperature in 3D configuration within the mini channel shows that the maximum temperature of the surface of the mini channel is close to $T = 311.7$ K for the copper cooler and 312.4 K for the aluminum cooler. In comparison between the three nanofluids, we find that the diamond-water nanofluid in which the temperature of the electronic component is low compared to the other liquids and therefore more appropriated for cooling.

Moving from the mini-channels of the cooler inlet to its outlet, we find that the temperatures in the mini-channels increase. This increase indicates the amount of heat transfer between the walls of cooler mini-channels and the coolant (nano-fluid).



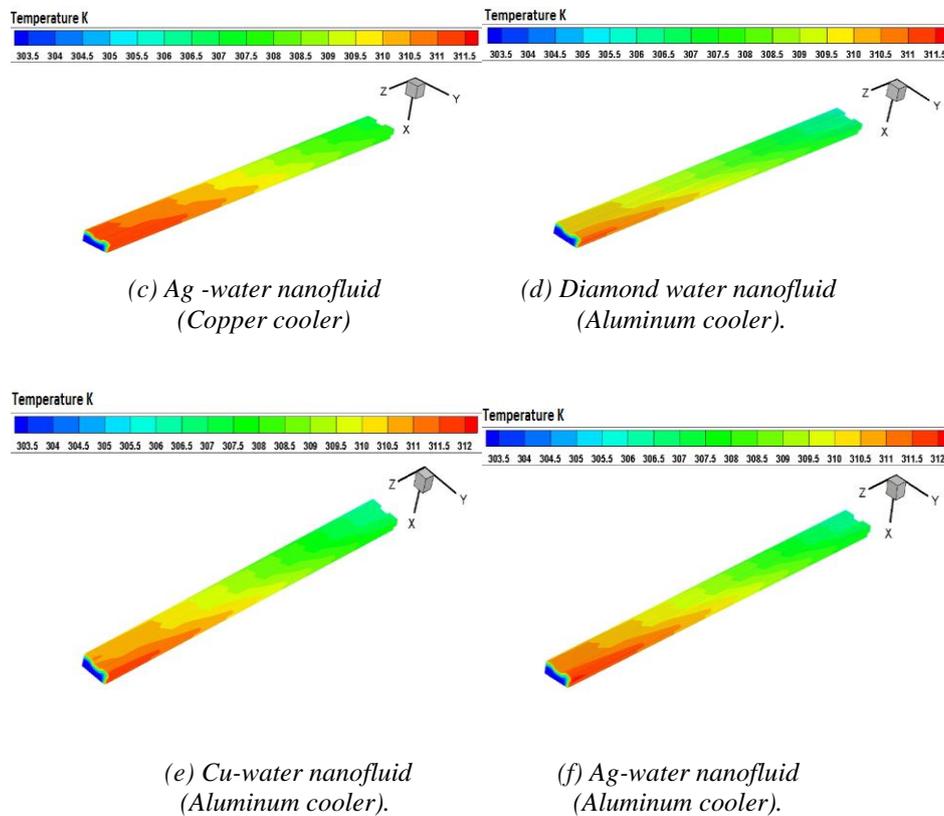


Fig. 8. Distribution of the temperature in the surfaces of the mini-channel for $Re = 1414$ and three different nano-fluids (volume fraction = 0.06).

Conclusions

In the present work, we have studied the effect of different types of nanoparticles numerically, by considering different volume fractions and types of cooler metals on heat transfer in a mini-channel cooler by using the commercial software, ANSYS-Fluent 15.0. The obtained results conclude that:

- For the Reynolds number equal to 1414 and the power of electronic component equal to 130W, the temperature of the electronic component is the lowest for the diamond- H_2O and it is the highest for the Ag- H_2O and Cu- H_2O nanofluids.

- For the two cooler metals types, the copper cooler is better in the reduction of the temperature followed in the aluminum.

- The use of diamond- H_2O nanofluid significantly gives higher heat transfer coefficients than those of the Ag- H_2O and Cu- H_2O nanofluids.

- The increase of the percentage of nanoparticle in basic fluid (water) allows ameliorating the heat transfer coefficient in a mini channels cooler.

These results will be used for the design and the improvements of the mini-channels cooler.

Nomenclature

Symbols

A_c	cross-sectional area of the mini channel (m^2)
A_w	inner wall or fluid contact surface area (m^2)
C_p	specific heat of the fluid ($Jkg^{-1} K^{-1}$)
D_h	hydraulic diameter of the mini channel (m)
E	thickness of thermal insulation (m)
g	acceleration of gravity ($m \cdot s^{-2}$)
H	height (m)
H_c	mini channel height (m)
h_{ave}	average heat transfer coefficient, $W m^{-2}K^{-1}$
K	thermal conductivity ($Wm^{-1}K^{-1}$)
L	mini channel length (m)
N	number of mini-channels
P	wetted perimeter (m)
p	pressure (Pa)
Q	power dissipated in the chip (W)
Re	Reynolds number ($Re = \rho w D_h / \mu$)
T	temperature (K)
T_m	mass-average temperature of the coolant (K)
T_w	area-weighted temperature of the channel wall surface (K)
x, y, z	cartesian coordinate (m)
u, v, w	velocity components ($m s^{-1}$)
W	width (m)
W_c	mini channel width (m)

Greek letters

ρ	coolant density (kg/m^3)
μ	dynamic viscosity of coolant ($kg/m.s$)
ϕ	volume fraction (%)
β	thermal expansion coefficient (K^{-1})
α	thermal diffusivity ($m^2.s^{-1}$)

Indices

c	mini channel
f	base fluid, water
in	inlet
n	outer normal coordinate at the interface between the wall and fluid
nf	nanofluid
out	outlet
s	solid

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