



An Investigation Into the Cooling Performance of Graphene and Copper Oxide Nanofluids in Electronic Devices

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Received: 20 02 2025; Accepted: 17 08 2025

Available: 31 08 2026

Abstract: The primary objective is to evaluate the effectiveness of graphene-based nanofluids as heat transfer fluids for electronic devices. For this investigation, graphene was prepared via the recycling of graphite from used batteries, adding a sustainable dimension to the material's synthesis. The analysis focuses on the effects of the nanofluid's volume fraction and the Reynolds number on the heat transfer rate.

The results demonstrate that both nanofluids significantly improve convective heat transfer compared to the base fluid (water). A comparative analysis reveals that the graphene nanofluid provides superior cooling performance, particularly at low volume fractions, where it shows a 13% increase in heat transfer over the copper oxide-water nanofluid. This enhanced performance is attributed to graphene's high thermal conductivity, which promotes more effective fluid flow and heat dissipation.

Keywords: Recycled graphite, Graphene, Nanofluids, open cavity, cooling, electronic devices

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Peer Review under the responsibility of Universidad Nacional Autónoma de México.

1. Introduction

The rapid increase in power density and miniaturization of electronic components have made thermal management a crucial challenge for the design and reliability of modern electronic devices (Lee et al., 2008). Inefficient heat dissipation can lead to overheating, significantly reducing the performance and lifespan of electronic systems such as microprocessors and batteries (Xiong et al., 2023). Traditional cooling fluids, although widely used, are showing their limitations in meeting increasingly demanding heat transfer requirements (Ghosh et al., 2008). To overcome these limitations, nanofluids, colloidal suspensions of nanoparticles in a base fluid, have emerged as a promising solution (Bae et al., 2010). Nanofluids offer the potential for substantial improvement in the thermal properties of conventional cooling fluids (Asim & Siddiqui, 2022). Graphene, with its exceptionally high thermal conductivity (Gabor et al., 2011), and copper oxide (CuO), known for its good conductivity and stability (Qu et al., 2016; Yan et al., 2015), are two nanomaterials that have been particularly studied in this field (Goossens et al., 2017; Yang et al., 2017). The rheology and viscosity of nanofluids play an essential role in their cooling performance and the pressure losses they induce (Wang et al., 2018). Numerous studies have explored the relationship between the viscosity of nanofluids and factors such as temperature, volume concentration, and particle aggregation (Kovtun et al., 2019; Liu et al., 2019; Awua, 2023; Deshmukh et al., 2017; Pashikanti & Susmitha Priyadarshini, 2023). Although increasing the concentration of nanoparticles tends to increase viscosity (Wright et al., 2013; Salemi et al., 2017), this phenomenon is critical because excessive viscosity can compromise the efficiency of the cooling system (Rudyak, 2013). Understanding these complex interactions remains a challenge, and there is no universal equation that can accurately predict the viscosity of nanofluids (Mahbubul et al., 2012). Existing research has also shown that nanofluids can improve convective heat transfer (Gaudin & Ma, 2020). However, studies on the combined effect of nanofluid properties and flow conditions on the cooling of discrete heat sources, such as those found in electronic circuits, remain limited (Nadooshan et al., 2018). This study is part of efforts to improve cooling systems using nanofluids in a cavity (Djermane & Kadri, 2023). We propose a numerical simulation of mixed laminar convection to evaluate and compare the cooling performance of graphene- and copper oxide CuO-based nanofluids. Our objective is to model the behavior of these fluids and understand the

cooling mechanism of a heat source placed on the wall of a cavity, simulating electronic components. This work aims to make significant contributions to the optimization of cooling solutions for next-generation electronic devices (Akande et al., 2023).

2. Physical Model

The 2D cavity is open at the inlet and outlet, with an orifice width of d . The cavity is bounded by thermally insulated walls. The right wall contains a heat source with a constant heat flux of length S , as shown in Figure 1. The nanofluid enters the cavity through the inlet with a uniform velocity V_i , temperature T_i , and uniform volume fraction ϕ . It then exits through the outlet.

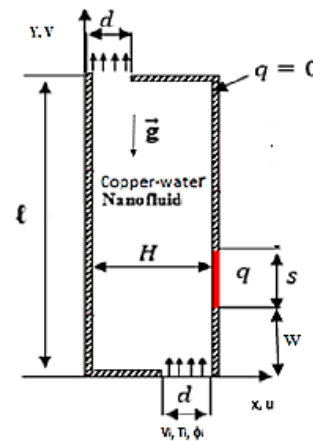


Figure 1. Physical model.

The thermophysical properties of the two nanofluids are considered constant, with the exception of their densities in the expression of the thrust force, for which the Boussinesq approximation is applied (Herwig, 2000; Ahmed et al., 2020; Pantokratoras, 2024). Here, T_0 is the reference temperature, β is the coefficient of thermal expansion at constant pressure, and ρ_0 is the density of the fluid at T_0 . The thermophysical properties and basic equations of the nanofluids used in this study are summarized in Tables 1 and 2.

Table 1. Thermophysical properties of the nanofluid components (Bakli et al., 2022; Makhlof et al., 2024).

Properties	ρ (kg.m ⁻³)	C_p (kg ⁻¹ k ⁻¹)	k (w.m ⁻¹ k ⁻¹)	β (k ⁻¹).10 ⁻⁵
H ₂ O	997.1	4179	0.613	21
CuO	8933	385	400	1.65
Graphene (Gr)	2100	710	5000	3.65

Table 2. Basic equations of the nanofluid (Hassan, 2019).

Density	$\rho_{nf} = (1 - \phi)\rho_f + \phi\rho_p$	(1)
Coefficient of thermal expansion	$(\rho\beta) = (1 - \phi)(\rho\beta)_f + \phi(\rho\beta)_p$	(2)
Heat capacity	$(\rho c_p)_{nf} = (1 - \phi)(\rho c_p)_f + \phi(\rho c_p)_p$	(3)
Thermal conductivity	$\frac{k_{nf}}{k_f} = \frac{k_p + 2k_f - \phi(k_f - k_p)}{k_p + 2k_f + \phi(k_f - k_p)}$	(4)
Dynamic Viscosity	$\mu_{nf} = \frac{\mu_f}{(1-\phi)^{2.5}}$	(5)

2.1. Governing Equations

The flow is steady, two-dimensional, and laminar, and the nanofluid is considered incompressible and Newtonian. Thermal equilibrium is established between the base fluid and the nanoparticles. Viscous dissipation is negligible.

a. Continuity equation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

b. Equations of motion

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

$$\rho_{nf} \left(u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} \right) = -\frac{\partial p}{\partial y} + \mu_{nf} \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) + (\rho\beta)_{nf}(T - T_i)g \quad (3)$$

c. Energy Equation

$$(\rho C_p)_{nf} \left(u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} \right) = K_{nf} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \quad (4)$$

2.2. Boundary Conditions

The boundary conditions for this problem are specified as follows:

$$\begin{aligned} \text{Bottom panel:} & \begin{cases} \text{wall : } u = v = 0, \frac{\partial T}{\partial x} = 0 \\ \text{Input : } v = V_i, u = 0, T = T_i, \phi = \phi_i \end{cases} \\ \text{Right wall:} & \begin{cases} u = v = 0; \left\{ \begin{array}{l} \frac{\partial T}{\partial y} = -\frac{q}{k_{eff}} \\ \frac{\partial T}{\partial y} = 0 \end{array} \right. \end{cases} \\ \text{Upper wall:} & \begin{cases} \text{wall : } u = v = 0, \frac{\partial T}{\partial y} = 0 \\ \text{Output : } \frac{\partial v}{\partial y} = 0; u = 0; \frac{\partial T}{\partial y} = 0 \end{cases} \\ \text{Left wall:} & u = v = 0; \frac{\partial T}{\partial y} = 0 \end{aligned}$$

2.3. Dimensionless Equations

Equations (1)-(4) and the boundary conditions (5)-(8) are made dimensionless using:

$$X = \frac{x}{H}, Y = \frac{y}{H}, U = \frac{u}{v_i}, V = \frac{v}{v_i}, AR = \frac{H}{d}, \theta = \frac{T - T_i}{T_h - T_i}$$

a. Continuity equation

$$\frac{\partial U}{\partial X} + \frac{\partial V}{\partial Y} = 0 \quad (9)$$

b. Equations of motion

$$U \frac{\partial U}{\partial X} + V \frac{\partial U}{\partial Y} = -\frac{\partial P}{\partial X} + \frac{\rho_f}{\text{Re} \rho_{nf}} \frac{\mu_{nf}}{\mu_f} \left(\frac{\partial^2 U}{\partial X^2} + \frac{\partial^2 U}{\partial Y^2} \right) \quad (10)$$

$$U \frac{\partial V}{\partial X} + V \frac{\partial V}{\partial Y} = -\frac{\partial P}{\partial Y} + \frac{\rho_f}{\text{Re} \rho_{nf}} \frac{\mu_{nf}}{\mu_f} \left(\frac{\partial^2 V}{\partial X^2} + \frac{\partial^2 V}{\partial Y^2} \right) + \frac{\beta_{nf}}{\rho_{nf} \beta_f} \text{Ri} \theta \quad (11)$$

c. Energy Equation

$$\left(U \frac{\partial \theta}{\partial X} + V \frac{\partial \theta}{\partial Y} \right) = \frac{\alpha_{nf}}{\alpha_f} \frac{1}{\text{RePr}} \left(\frac{\partial^2 \theta}{\partial X^2} + \frac{\partial^2 \theta}{\partial Y^2} \right) \quad (12)$$

Dimensionless boundary conditions:

$$\begin{aligned} \text{Bottom panel:} & \begin{cases} \text{wall : } U = V = 0, \frac{\partial \theta}{\partial X} = 0 \\ \text{Input : } V = 1, U = 0, \theta = 0, \phi = \phi_i \end{cases} \\ \text{Right wall:} & U = V = 0; \left\{ \begin{array}{l} \frac{\partial \theta}{\partial Y} = \frac{kf}{k_{nf}} \\ \frac{\partial \theta}{\partial Y} = 0 \end{array} \right. \\ \text{Upper wall:} & \begin{cases} \text{wall : } U = V = 0, \frac{\partial \theta}{\partial X} = 0 \\ \text{Output : } \frac{\partial V}{\partial Y} = 0; U = 0 \end{cases} \\ \text{Left wall:} & u = v = 0; \frac{\partial \theta}{\partial Y} = 0 \end{aligned}$$

The Nusselt number reflects the heat exchange that takes place between the heat source and the nanofluid. We define the local Nusselt number as:

$$Nu = \frac{hH}{K_{nf}}$$

where h is the convective heat transfer coefficient given by:

$$h = \frac{q}{T_w - T_m}$$

The average Nusselt number Nu_{avg} is calculated as follows:

$$Nu_{avg} = \frac{1}{s} \int_w^{w+s} Nu_y dy$$

3. Numerical Procedure

The study was carried out using ANSYS Fluent, an established computational fluid dynamics (CFD) software package. The cavity mesh and geometry were created using the ANSYS Meshing module, while the Fluent solver, which uses the finite volume method to discretize the equations, was used to simulate the flow and heat transfer.

4. Results

4.1 Validation of the Numerical Code

To verify the accuracy of the numerical results and the validity of the mathematical model obtained throughout the present study, comparisons with previously published results were made; the outcome of the present numerical

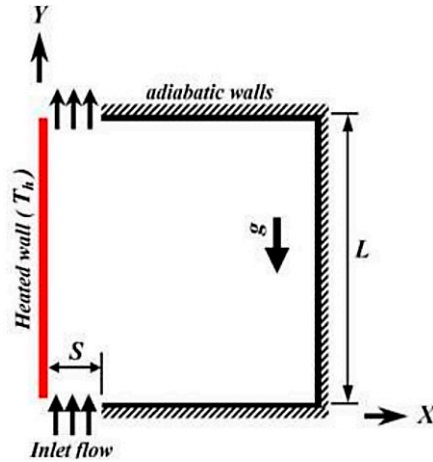


Figure 2. The cavity configuration (Reyadh Al-Zuhairy, et al. (2023)).

code was benchmarked against the numerical results of Reyadh Al-Zuhairy, et al. (2023), who numerically investigated convection heat transfer in a square cavity that has been vented vertically. The streamlines and isotherms for the Reynolds number $Re=150$, $AR=0.1$, $Ri=10$, and $Pr=0.71$ are presented in Figures 2 and 3.

The figures show the isocontours in the channel and demonstrate excellent agreement between the results obtained by our code and those of Reyadh Al-Zuhairy, et al. (2023),

Figure 4 shows the evolution of the mean Nusselt number as a function of the Reynolds number, confirming the accuracy of our calculation code over a wider range of parameters.

	Grid	Isotherms	Streamlines
(Reyadh Al-Zuhairy, et al. (2023))			
Present Study			

Figure 3. Comparison of the present study with the results of Reyadh Al-Zuhairy, et al. (2023), $Re=150$, $AR=0.1$, $Ri=10$, and $Pr=0.71$.

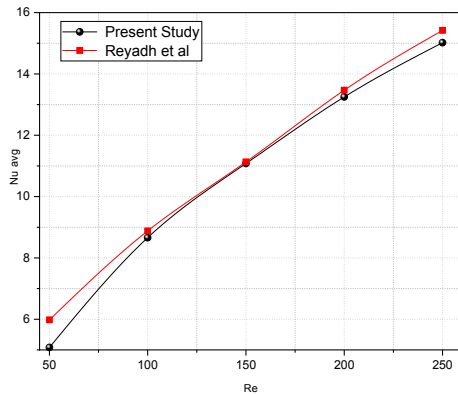


Figure 4. Comparison of the variation of the average Nusselt number as a function of the Re number for a form factor =0.1

The second validation shows a maximum error of 4% in the mean Nusselt number, confirming the model's accuracy. The third validation compares our results with those of Manca et al. (2008) on mixed convection in an open cavity. Figures 5 to 7 illustrate the simulation conditions.

The results of this study show that for the two cases, $Ri=1$ and $Ri=10$, there is good agreement between the results.

4.2. Mesh Sensitivity

A mesh independence study was carried out under typical reference conditions: a nanofluid ($CuO-H_2O$), Richardson number $Ri=3$, aperture ratio $AR=4$, and volume fraction $j=5\%$. The mean Nusselt value was chosen as the reference

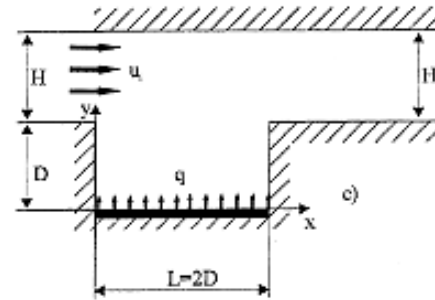


Figure 5. Geometry considered, case (c): heating from below (Manca et al., 2008).

variable to assess the accuracy of the solution. A quadratic mesh was chosen. This is described in Figure 8. It is observed that the magnitude of the average Nusselt number Nu_{avg} for 12,000 nodes shows very little difference from the results obtained for higher node counts. A mesh with 12,000 nodes was chosen for further calculation of the other cases. This grid resolution is therefore used for all subsequent calculations. The convergence criterion requires that the maximum absolute error between two successive iterations be 10^{-6} .

4.3 Results and Discussion

This section presents an analysis of the mixed convection of a $Gr-H_2O$ nanofluid at steady state within an open cavity. Given the large number of control parameters, we have fixed the dimensions of the cavity and heat source for all calculations (Table 3).

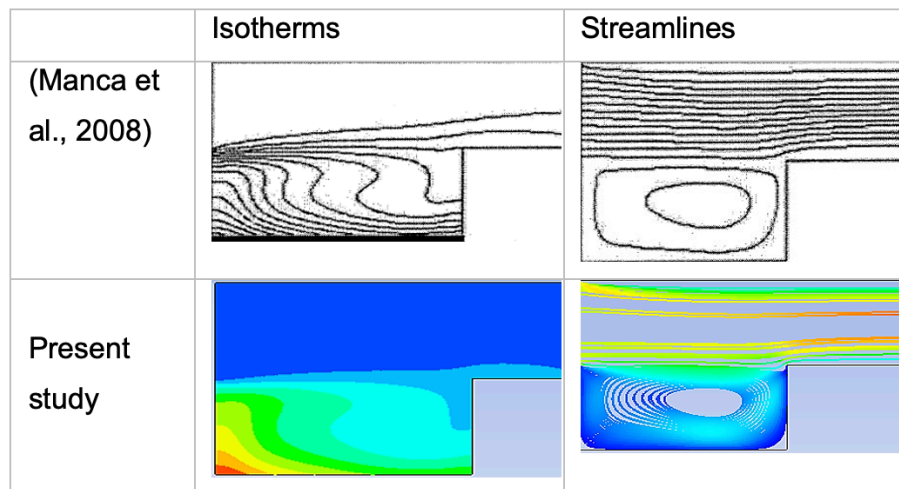


Figure 6. Isocontours a $Re=100$, $H/D=1$, $Ri=1$.

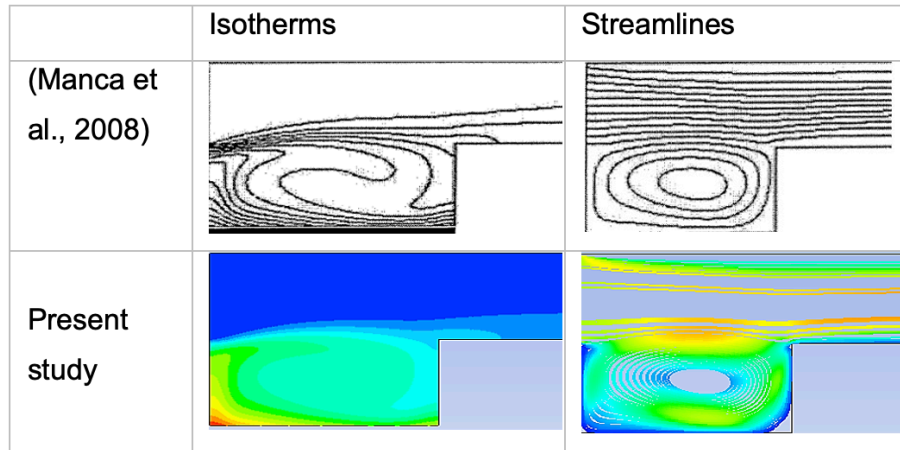


Figure 7. Isocontours a $Re=100$, $H/D=1$, $Ri=10$.

Table 3. Cavity dimensions and heat source.

Cavity height (L)	100 mm
Cavity width (H)	20 mm
Position of source relative to input (w)	20 mm
Length of heat source (S)	20 mm
Cavity opening width (OR)	5 mm

4.3.1 Effect of Volume Fraction ϕ on the Behavior of $(Gr-H_2O)$

The first set of results, presented in Figures 9 to 13, aims to analyze the effect of the volume fraction ($0 \leq \phi \leq 0.005$) of the nanofluid ($Gr-H_2O$) on cooling performance. For this purpose, we adopted a reference configuration already studied in the literature (Selvam et al., 2017). Analysis of the volume fraction (ϕ) is crucial, as it directly influences the thermophysical properties of the nanofluid. Isotherms reveal a direct link between the volume fraction (ϕ) of the $Gr-H_2O$ nanofluid and cooling efficiency. For a low concentration of $\phi=0.001$, the isotherms show a progressive temperature distribution. The maximum temperature near the hot source reaches 324 K, suggesting moderate heat transfer and low cooling efficiency. By increasing the concentration to $\phi=0.003$, the maximum temperature near the hot source decreases to 315 K. This significant reduction of 9 K confirms improved cooling performance.

For the highest concentration tested, $\phi=0.005$, the maximum temperature drops further to 313 K. This continuous improvement indicates that heat transfer is becoming increasingly efficient. As ϕ increases,

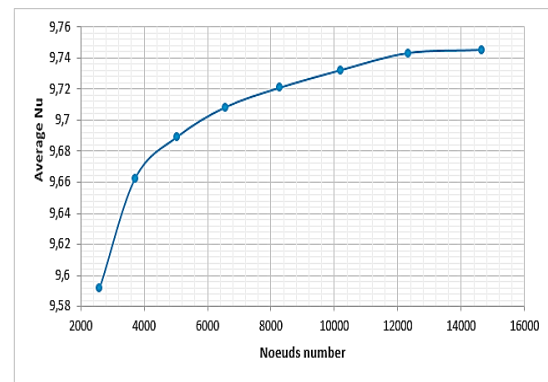


Figure 8. Mesh independence study
 $Ri = 3.2$, $OR = 4$ and $\phi = 5\%$.

the isotherms become more concentrated near the hot source, providing visual evidence of improved heat transfer.

The volume fraction also influences the flow behavior. The main flow is generated by a jet of nanofluid from the inlet, which accelerates in the right-hand region of the cavity. However, recirculation zones form in the left-hand region, sometimes leading to flow inversion. For $\phi=0.001$, the streamlines show several recirculation cells, with large central cells dominating and smaller secondary cells near the walls. When the concentration is increased to $\phi=0.003$, the flow structure changes slightly. Central cells become more elongated, and secondary cells appear to shrink. This phenomenon is the result of improved thermal conductivity, a property directly modified by the addition of graphene nanoparticles.

To quantitatively assess the impact of the volume fraction (ϕ) of the $Gr-H_2O$ nanofluid on cooling performance,

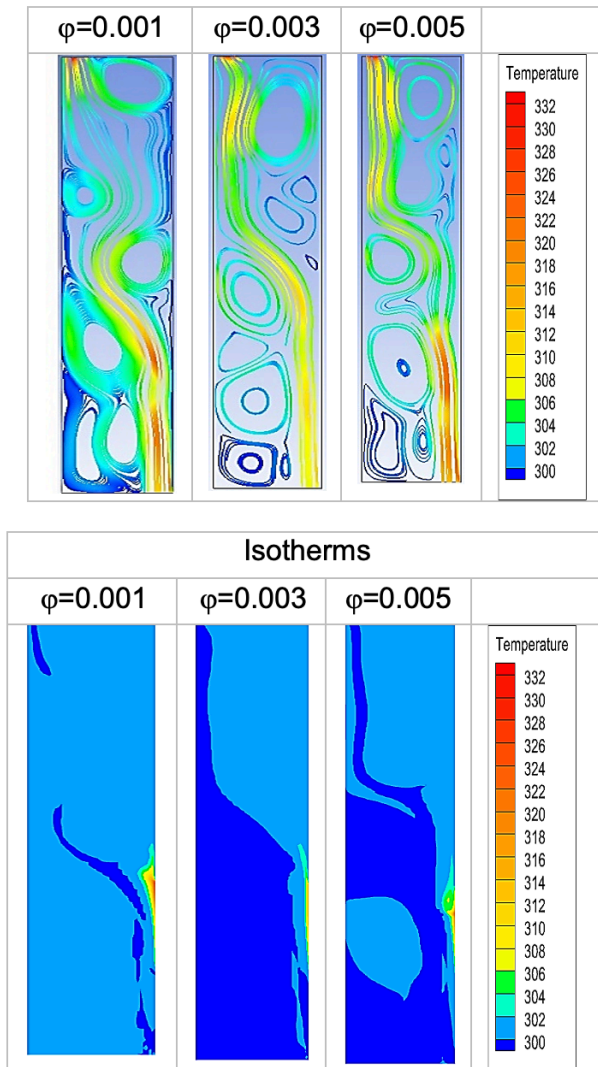


Figure 9. Isotherm and streamline maps of the Gr-H₂O nanofluid at Re=500, R=0.25 for three volume fractions: 0.1%, 0.3% and 0.5%, respectively.

we analyzed the evolution of the local Nusselt number (Nuloc) along the heat source. Figure 10 illustrates the evolution of the local Nusselt number (Nuloc) along the heat source for the Gr-H₂O nanofluid at a fixed Reynolds number of Re=500 and for different volume fractions (ϕ). Nanoparticle concentration has a direct impact on heat transfer efficiency.

For the range of graphene nanofluid concentrations tested, the Nuloc curve follows a similar trend. It starts at a high value near the leading edge of the heat source ($y=0.020$ m), decreases rapidly to around $y=0.035$ m, and finally increases slightly toward $y=0.040$ m, at the end of the source. This variation is typical of convection problems: heat transfer is maximal at the point of entry of the

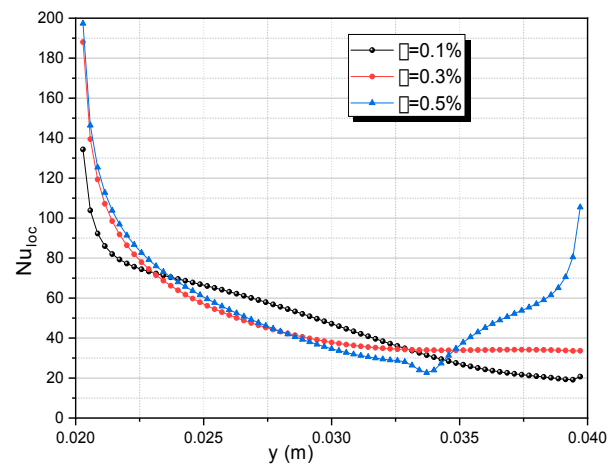


Figure 10. Evolution of the local Nusselt number (Nuloc) at Re=500, for Gr-H₂O at different volume fractions.

fresh fluid, decreases as the fluid warms up by flowing along the hot source, then increases again due to recirculation effects that renew the thermal boundary layer. The significant improvement in heat transfer obtained with increasing nanofluid volume fraction confirms that the addition of graphene nanoparticles enhances the fluid's thermal conductivity and, consequently, its ability to dissipate heat from the source.

4.3.2 Comparative Study Between (CuO-H₂O) and (Gr-H₂O).

A comparative study was also carried out using another widely studied nanofluid, Cu-H₂O (Manca et al., 2008), to establish a benchmark. Our literature search showed that for graphene, effective concentrations for significant cooling are generally in the range from 0.01% to 2% (Selvam et al., 2017). For copper oxide-based nanofluids, on the other hand, this range extends from 1% to 5%. To ensure a relevant and direct comparison between the two types of nanofluids, we performed our calculations on a common and representative volume fraction range: $0\% \leq \phi \leq 0.5\%$.

a. Maps of Isocontours

Reducing the size of inlet openings encourages the formation of vortices and an increase in their size. This phenomenon is explained by the increase in nanofluid injection speed to maintain a constant flow rate, which intensifies fluid shear. The flow, under the effect of the incoming jet, is characterized by acceleration in the right-hand part of the cavity and deceleration in the left-hand part, leading to the formation of recirculation zones. The

streamlines reveal that increasing the nanofluid's volume fraction leads to intensified convection.

A more elongated central cell and less pronounced secondary cells are observed.

This configuration suggests more vigorous fluid movement, the result of improved thermal conductivity and increased nanofluid density. Both properties promote more efficient heat transfer and modify the flow profile inside the cavity.

b. Nusselt Numbers

The above results have a direct effect on the Nusselt numbers shown in Figures 12 and 13.

Clearly, dispersing nanoparticles in the base fluid increases convective heat transfer. The average heat transfer rate of the heat source increases with the volume fraction (ϕ) of nanoparticles. This improvement is attributed to the increase in the nanofluid's effective thermal conductivity, which is directly proportional to the concentration of nanoparticles in the base fluid. This improvement is particularly notable for the Gr-H₂O

nanofluid, which exhibits more favorable heat exchange than the CuO-H₂O nanofluid. For example, at a low volume fraction of $\phi=0.1\%$, the Gr-H₂O nanofluid achieves a 13% improvement in heat transfer compared with the CuO-H₂O nanofluid.

c. Comparative study of the Effect of Re on Isocontour Maps Between (CuO-H₂O) and (Gr-H₂O).

The second set of results (Figures 14 to 17) focuses on the effect of Reynolds number on the behavior of the two nanofluids, keeping the volume fraction constant at $j=0.005$.

A study of the dynamic behavior of nanofluids for different values of the Reynolds number (Re) reveals some interesting trends. For Re=100 and Re=300, the dynamic behavior of Gr-H₂O and CuO-H₂O is similar, with a comparable flow pattern, whereas for Re=500, the Gr-H₂O nanofluid shows an intensification of recirculation zones compared to the two previous cases. This slight difference can be explained by the thermophysical properties of graphene, which affect fluid motion at higher flow

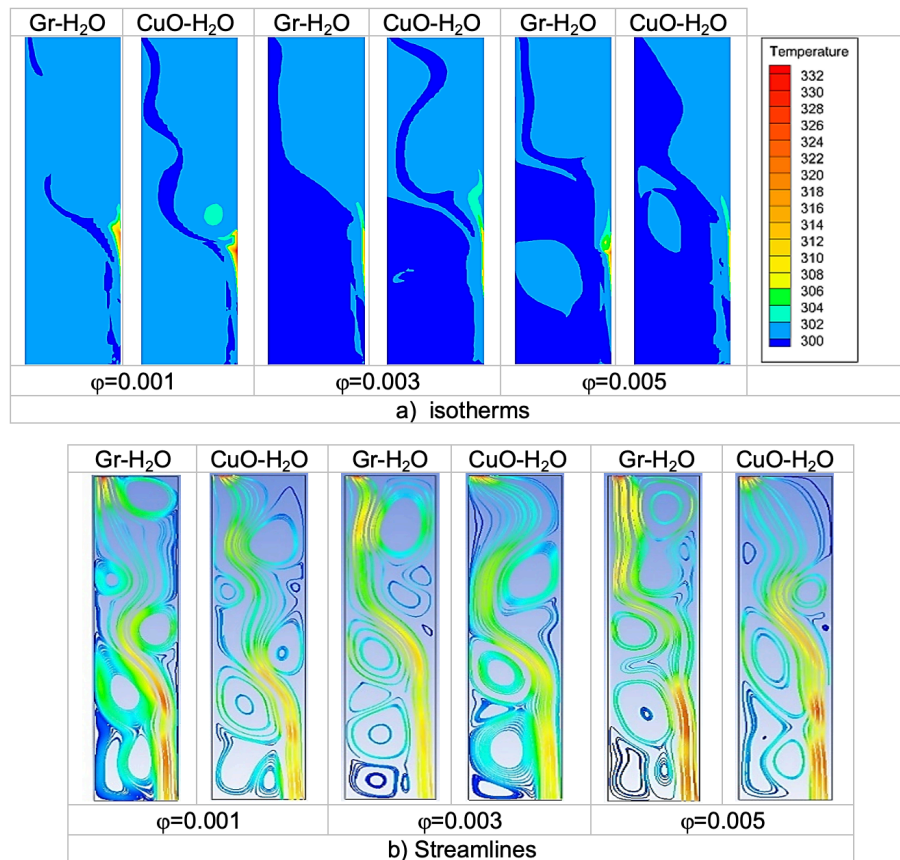


Figure 11. Isocontours at $Re=500$.

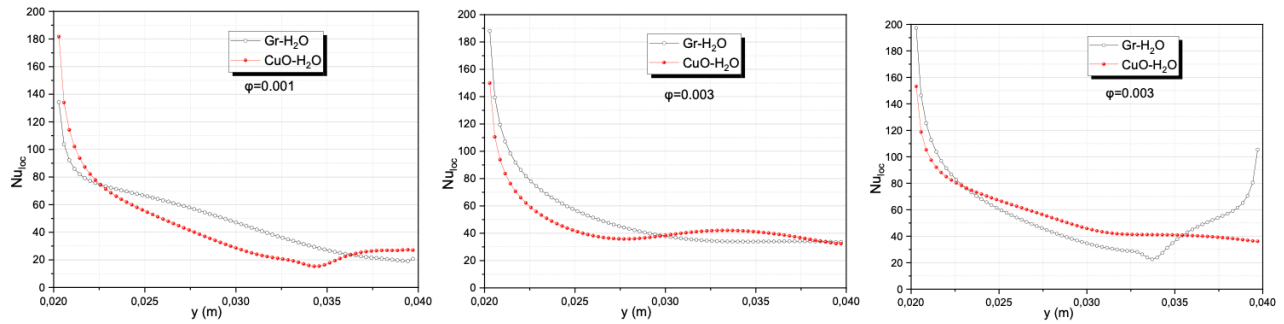


Figure 12. The local Nusselt number evolution at $Re=500$.

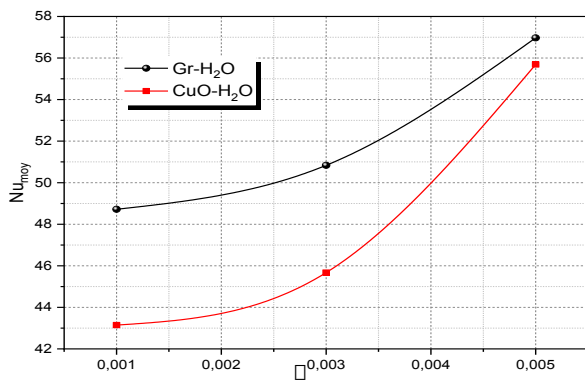


Figure 13. Evolution of the average Nusselt number for different volume fractions.

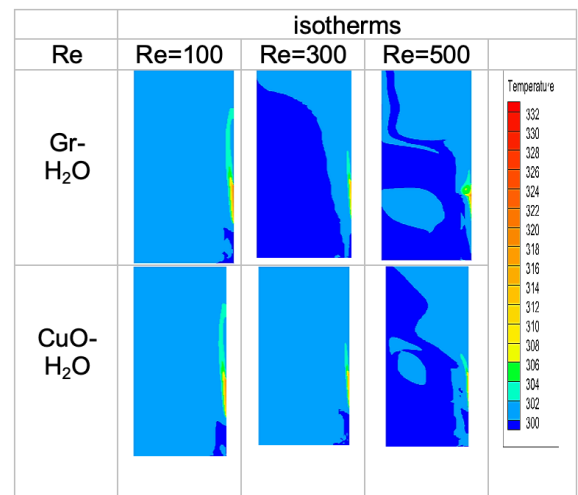
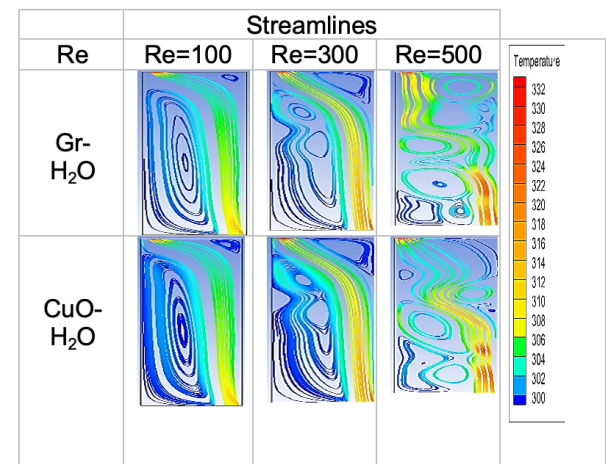


Figure 14. Isotherms and current lines at $j=0.005$.

velocities. The increase in Reynolds number, which corresponds to a higher inlet velocity and flow rate of the nanofluid, has a direct and positive impact on the thermal field. We observe a reduction in the thickness of the thermal boundary layer at the surface of heat sources. This leads to a lower source temperature and a more uniform temperature distribution in the cavity. These dynamic and thermal field structures have a direct impact on the Nusselt number, a measure of heat transfer efficiency.

d. Effect of Re on The Nusselt Number

In convection cooling systems, increasing Re has a direct and significant impact on heat transfer (Figure 15).

Heat transfer is greatest near the leading edge of the hot source, where the incoming fluid is coldest. It then gradually decreases as the fluid warms up and flows along the surface. This trend is observed irrespective of the Reynolds number (Re). However, for the case of $Re=500$, a distinct phenomenon occurs: heat transfer increases again toward the end of the hot source.

This improvement is attributed to the intensification of recirculation effects, which renew the fluid near the surface and disrupt the thermal boundary layer.

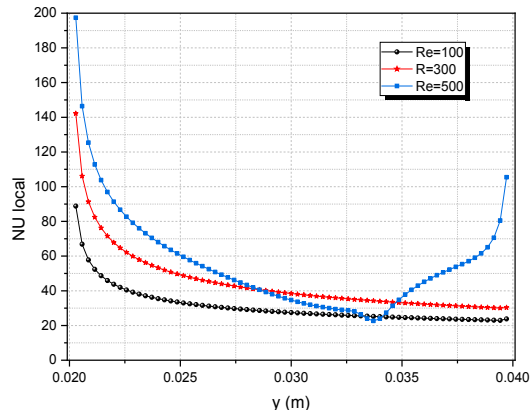


Figure 15. Local Nusselt number (Nu) of Gr-H₂O at $j=0.005$.

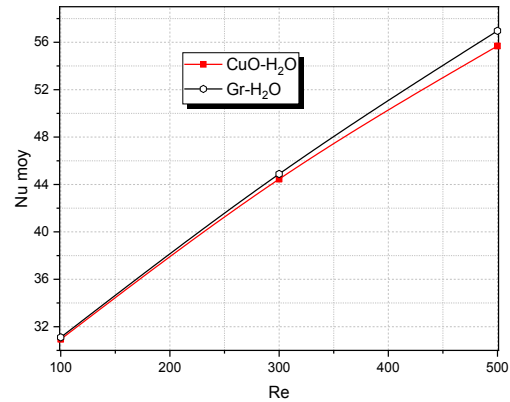


Figure 17. Evolution of the average Nusselt number as a function of Reynolds number.

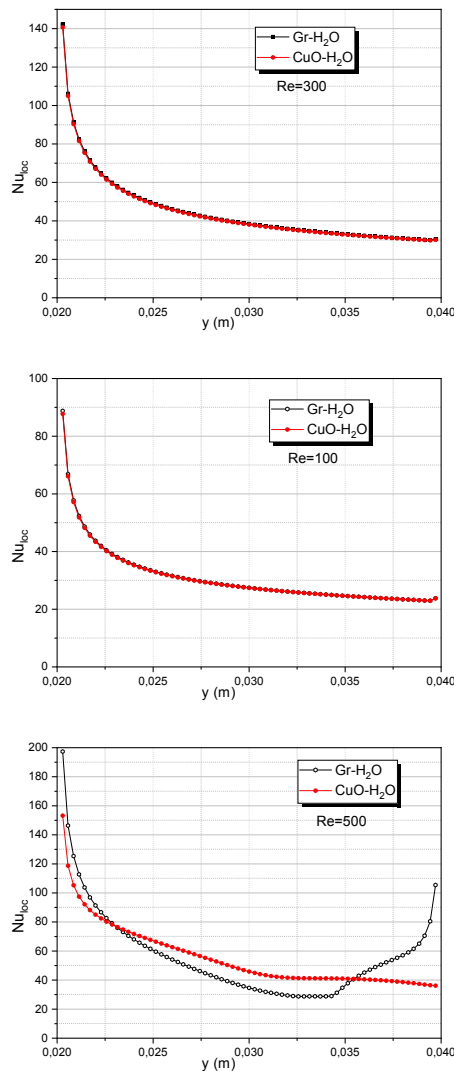


Figure 16. Local Nusselt number as a function of Re.

Analysis of cooling performance reveals distinct trends for both nanofluids as a function of Reynolds number (Re). At $Re=100$ and $Re=300$, the behavior of the Gr-H₂O and CuO-H₂O nanofluids is identical. However, for faster flow at $Re=500$, the Gr-H₂O nanofluid performs better than the CuO-H₂O nanofluid.

This superiority of Gr-H₂O can be explained by its lower density compared to CuO-H₂O. Lower density enables more efficient formation of recirculation effects, which renew the fluid in contact with heat sources more rapidly. Increasing the Reynolds number enhances this transfer by accelerating nanofluid movement near the heat source.

Conclusion

This study conducted a numerical investigation of the cooling performance of graphene (Gr-H₂O) and copper oxide (CuO-H₂O) nanofluids in an open cavity, simulating the cooling of electronic components. The simulation results clearly demonstrated that nanofluids are more efficient than base fluids for convective heat transfer. Analysis of the impact of volume fraction (ϕ) revealed that the average Nusselt number increases significantly with increasing nanoparticle concentration, which is directly linked to the fluid's improved thermal conductivity. Our comparative study highlighted the superiority of the Gr-H₂O nanofluid over the CuO-H₂O nanofluid, particularly at low volume fractions where Gr-H₂O displayed a 13% improvement in heat transfer over CuO-H₂O. This performance is attributed not only to graphene's exceptional thermal conductivity, but also to its low density, which promotes more efficient flow motion and beneficial recirculation zones, particularly at high Reynolds numbers.

Conflict of Interest

The authors have no conflict of interest to declare.

Funding

The authors received no specific funding for this work.

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