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Conference paper

Improvement of heat transfer within solar water heater's tubes (SWH) using nanofluids

Toufik Benmalek [a,*], Abdelkader Boutra [b], Moustafa Boukraa [c]

[a] *LMFTA laboratory, USTHB University , Algiers*

[b] *bNational School of Advanced Technologies, ENSTA, Algiers*

[c] *Research Center in Industrial Technologies CRTI, ,Algiers*

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ABSTRACT

This research offers a numerical study of steady and laminar mixed convection flow in a circular pipe as part of a flat plate solar collector, which is crossed by nanofluids. The pipe is kept at a constant wall temperature and then at a constant heat flux, which represents the solar radiation received by the pipe. The governing coupled equations are solved with the finite volume approach. Computations are carried out using an in-house computer code, which has been satisfactorily validated by comparison to previous investigations. Empirical relations are used to predict the effective thermal conductivity and viscosity of nanofluids. The results are investigated using dynamic and thermal fields, with a special emphasis on the Nusselt number calculated along the active wall. They demonstrate that increasing the volume fraction of nanoparticles and Reynolds number improves heat transfer.

1. INTRODUCTION

Industrial growth emphasizes the importance of improving heat transport. Many researchers are interested in emerging technologies that might improve the thermal performance of fluids. Nanofluids outperform traditional fluids in terms of heat transmission. This innovative class of fluids is formed by adding nanoparticles to a basic fluid. The inclusion of these particles in the fluid greatly improves thermal performance when compared to the base fluid. Abbasian et al. (2012) investigated heat transmission via mixed convection in a square chamber filled with nanofluid. The vertical walls are kept at a separate and constant temperature, whereas the remaining walls are adiabatic. They investigated the effects of Richardson number and nanoparticle volume fraction modifications on the hydrodynamics.

[*] *Corresponding author, E-mail address: t.benmalek@yahoo.fr*

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Rana & Bhargava. (2011) conducted a numerical investigation of the effect of nanoparticle shape and type on laminar mixed convection of nanofluid into a vertical plate exposed to a constant heat flux. Nanoparticles were analysed in spherical and cylindrical geometries, with volume percentages of up to 4%. Various nanofluids (Cu, Ag, Al2O3, and TiO2) were tested for different Richardson numbers. The results show that when the Richardson number and nanoparticle volume fractions rise, they correspondingly increase the average Nusselt number. Furthermore, the type of nanoparticles was discovered to be a crucial element influencing heat transfer enhancement. Shahi et al (2010) used the finite volume approach to solve a numerically natural convection of nanofluid (copper-water) in an asymmetric tube. The investigation included volume fractions ranging from 0 to 0.05 and heat fluxes ranging from 1000 to 700 W/m2. They evaluated the effect of tube dimensions and inclination angle on circulation within the tube, and found that maximum mass flow at the exit rises with inclination angle and solid concentration. Recent research indicates that nanofluids might significantly enhance the efficiency of solar collectors, photovoltaic-thermal systems, and solar stills, among other uses (García-Rinc´on & Flores-Prieto, 2023), (Tiko 2023). As a result, replacing traditional working fluids (such as Al2O3, Ag, SiO2, CuO, TiO2, and MgO) with nanofluids improves the thermal characteristics of solar collectors, resulting in up to 35% efficiency gains. Nanofluids often improve thermal efficiency in solar water heating applications as well.

Incorporating nanoparticles into standard working fluids can improve their thermophysical characteristics, notably heat conductivity. Furthermore, research on the thermal stability of nanofluids shows that low flow rates are critical for enhancing stability (Al-Mamun et al, 2023), (Chekifi & Boukraa, 2022, 2023).

The purpose of this work is to analyse numerical heat transfer within a circular enclosure traversed by nanofluid, replicating the tube of a solar water heater. The research focuses on the combined impact of Reynolds and Richardson numbers, as well as the effect of nanoparticle volume fraction, on thermal performance within the enclosure.

2. PHYSICAL MODEL

Consider a cylindrical pipe with a length of L and a radius of r_w that is filled with nanofluid. The wall of a pipe exposed to solar radiation is supposed to be isothermal, or susceptible to a constant heat flux.

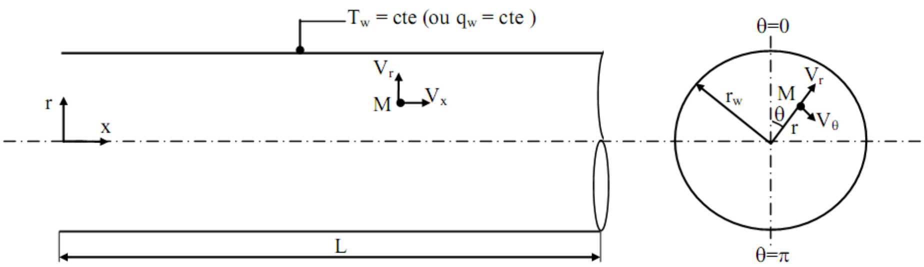


Fig 1. Geometry of the present study

3. MATHEMATICAL FORMULATION

The dimensionless transport equations of continuity (1), momentum (2,3 & 4) and energy (5) are:

$$\frac{1}{R}\frac{\partial(RV)}{\partial R}+\frac{1}{R}\frac{\partial W}{\partial \theta}+\frac{\partial U}{\partial X}=0 \qquad (1)$$

$$\frac{1}{R}\frac{\partial(R\,V\,V)}{\partial R}+\frac{1}{R}\frac{\partial(W\,V)}{\partial\theta}+\frac{\partial(U\,V)}{\partial X}-\frac{W^2}{R}=-\frac{\partial P^*}{\partial R}+\frac{1}{\text{Re}}\frac{\mu_{nf}}{\rho_f\nu_f}\left[\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial V}{\partial R}\right)\right.$$
$$\left.+\frac{1}{R^2}\frac{\partial}{\partial\theta}\left(\frac{\partial V}{\partial\theta}\right)+\frac{\partial}{\partial X}\left(\frac{\partial V}{\partial X}\right)\right]+\frac{1}{\text{Re}}\frac{\mu_{nf}}{\rho_f\nu_f}\left[-\frac{V}{R^2}-\frac{2}{R^2}\frac{\partial W}{\partial\theta}\right]-\frac{\text{Gr}}{\text{Re}^2}\frac{(\rho\beta)_{nf}}{\rho_{nf}\beta_f}T^*\cos\theta \tag{2}$$

$$\frac{1}{R}\frac{\partial(R\,V\,W)}{\partial R}+\frac{1}{R}\frac{\partial(W\,W)}{\partial\theta}+\frac{\partial(U\,W)}{\partial X}+\frac{V\,W}{R}=-\frac{1}{R}\frac{\partial P^*}{\partial\theta}+\frac{1}{\text{Re}}\frac{\mu_{nf}}{\rho_f\nu_f}\left[\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial W}{\partial R}\right)\right.$$
$$\left.+\frac{1}{R^2}\frac{\partial}{\partial\theta}\left(\frac{\partial W}{\partial\theta}\right)+\frac{\partial}{\partial X}\left(\frac{\partial W}{\partial X}\right)\right]+\frac{1}{\text{Re}}\frac{\mu_{nf}}{\rho_{nf}\nu_f}\left[\frac{2}{R^2}\frac{\partial V}{\partial\theta}-\frac{W}{R^2}\right]+\frac{\text{Gr}}{\text{Re}^2}\frac{(\rho\beta)_{nf}}{\rho_{nf}\beta_f}T^*\sin\theta \tag{3}$$

$$\frac{1}{R}\frac{\partial(R\,V\,U)}{\partial R}+\frac{1}{R}\frac{\partial(W\,U)}{\partial\theta}+\frac{\partial(U\,U)}{\partial X}=-\frac{\partial P^*}{\partial X}+\frac{1}{\text{Re}}\frac{\mu_{nf}}{\rho_f\nu_f}\left[\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial U}{\partial R}\right)+\frac{1}{R^2}\frac{\partial}{\partial\theta}\left(\frac{\partial U}{\partial\theta}\right)+\frac{\partial}{\partial X}\left(\frac{\partial U}{\partial X}\right)\right] \tag{4}$$

$$\frac{1}{R}\frac{\partial(R\,V\,T^*)}{\partial R}+\frac{1}{R}\frac{\partial(WT^*)}{\partial\theta}+\frac{\partial(U\,T^*)}{\partial X}=\frac{\alpha_{nf}}{\alpha_f}\frac{1}{\text{Pr}\,\text{Re}}\left[\frac{1}{R}\frac{\partial}{\partial R}\left(R\frac{\partial T^*}{\partial R}\right)+\frac{1}{R^2}\frac{\partial^2T^*}{\partial\theta^2}+\frac{\partial^2T^*}{\partial X^2}\right] \tag{5}$$

3.1 Numerical method

The governing equations (eqs. 1 to 5) are discretized using Patankar's finite volume approach [9]. The power-law technique is used to discretize the advection-diffusion components in the momentum and energy equations. The SIMPLER method is used to solve the checkerboard pressure problem on a staggered mesh, with under-relaxation variables added to control convergence. Linear algebraic equations generated by discretization are solved iteratively using the TDMA line-by-line approach. The numerical technique is implemented using the FORTRAN programming language. The convergence of the numerical results is achieved when the sum of residuals for each equation (u, v, w, and T*) falls below 10^{-6}.

3.2 Code validation

An air flow via a horizontal cylindrical pipe was validated with Nguyen et al. [10]. Table 1 demonstrates that the maximum variation between these outcomes is less than 4.5%.

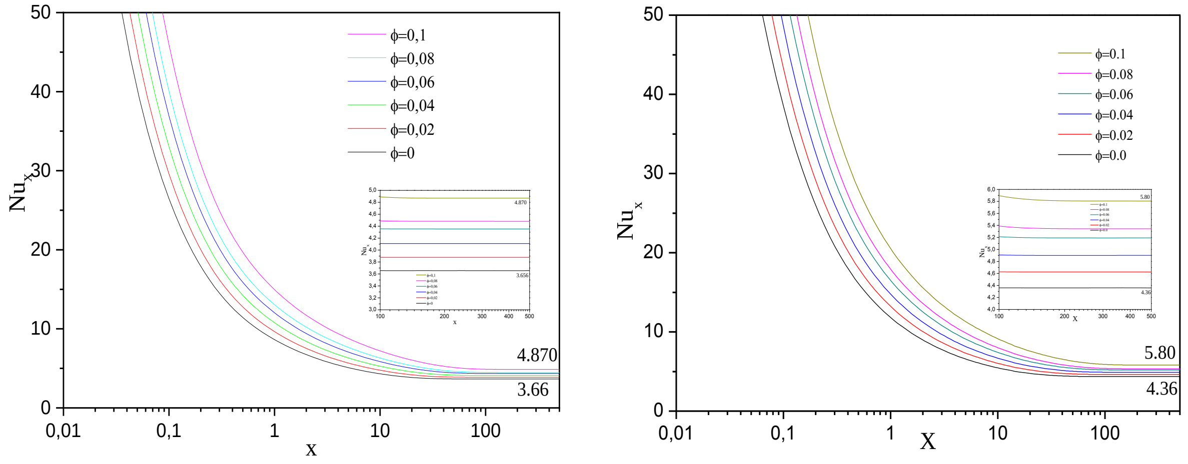
Table 1. Comparison of the radial distribution of the axial velocity 'u' with [10]. X = 19.5, Pr =0.71 and Re =500, Gr = 10^5

R	**0**	**0.05**	**0.10**	**0.15**	**0.20**	**0.25**	**0.30**	**0.35**	**0.40**	**0.45**	**0.50**
Ref [10]	2.22	2.20	2.13	2.00	1.79	1.68	1.13	0.97	0.62	0.34	00
Our results	2.17	2.15	2.09	1.96	1.73	1.63	1.20	1.02	0.64	0.34	00

4. RESULTS AND DISCUSSION

4.1 Influence of the volume fraction

Figure (2) shows how the local Nusselt number varies with varying volume fractions (ϕ) under two conditions: constant heat flux and constant wall temperature. The Nusselt number increases as the volume fraction (ϕ) of nanoparticles in the base fluid increases. This suggests that "Ag" nanoparticles can improve convective heat exchange by increasing the fluid's thermal conductivity. Furthermore, when these two situations (constant heat flux and constant wall temperature) are compared, the constant heat flux at the wall has a greater Nusselt number. Specifically, inside the region where the thermal regime is created, this value rises by 33.06% for the constant wall temperature and 33.03% for the constant heat flux conditions.



(a) (b)

Fig 2. Evolution of the Nusselt number for different values of the volume fractionϕ; Re=100
(a) Constant wall temperature. b) Constant wall heat flux

Figure (3) illustrates the influence of volume fraction on axial temperature distribution. From the entry to the exit of the pipe, it is observed that each curve exhibits a consistent profile. The temperature changes linearly due to the heat flux entering the fluid.

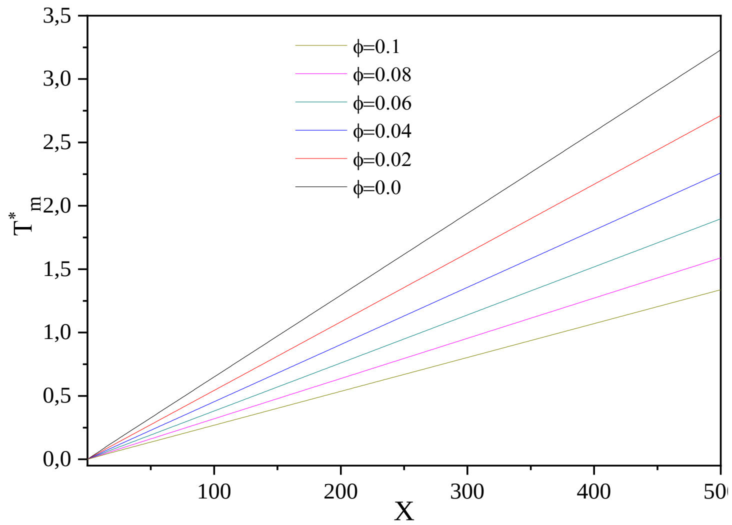
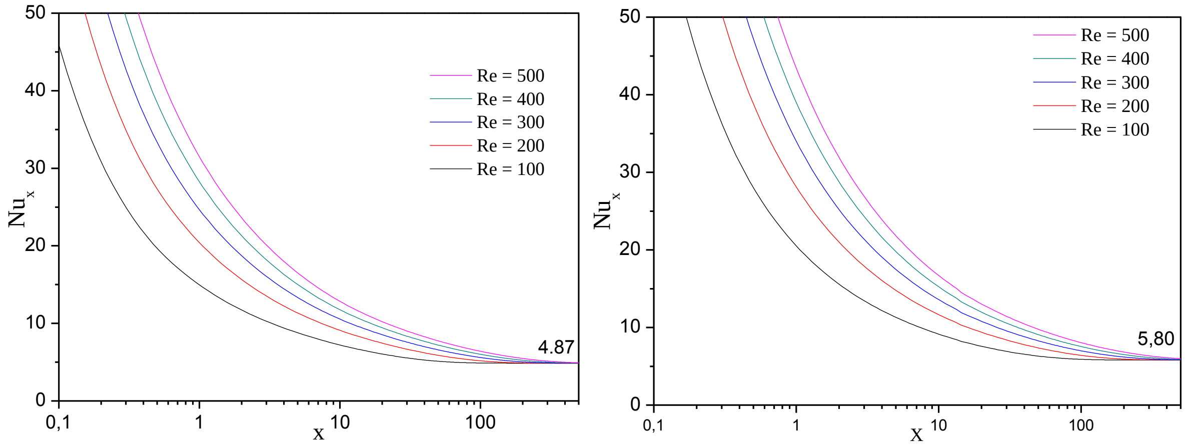


Fig 3. Temperature profiles for different values of ϕ at Re = 100

4.2 Reynolds number effect

Figure (4) depicts the development of the Nusselt number at various Reynolds values. All curves follow a regular pattern: a steep fall in the entrance zone, accompanied by a strong temperature gradient, followed by a progression to a limiting value, indicating the formation of the thermal regime. We find that Reynolds number variation has only a detectable effect in the inlet zone, where an increase in

Reynolds number accelerates heat exchange and expands the impacted zone. However, Reynolds number modifications have no effect on the asymptotic value of the Nusselt number for the established thermal regime (Nu∞ = 4.87 for constant wall temperature and Nu∞ = 5.80 for constant heat flux).



(a) (b)

Fig 4. Local Nusselt number for different values of the Reynolds number at ϕ = 0.1;
(a) Constant wall temperature. (b) Constant heat flux

Figure 5 shows the Nusselt number ratio for varying volume fractions of nanofluids, relative to a pure base fluid (ϕ = 0). Increasing the volume fraction of nanoparticles greatly improves heat transmission, with a maximum at ϕ=0.1 (47%), compared to ϕ=0.

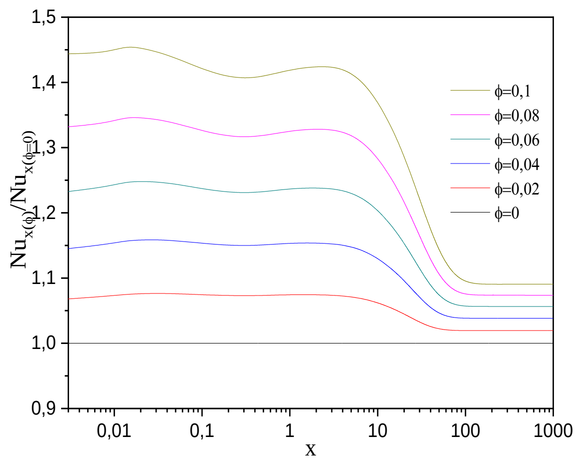


Fig 5. Report of the local Nusselt number as a function of the volume fraction ϕ.Re = 100 and Gr = 10^5

5. CONCLUSION

Following the previous studies, it appears that the presence of nanoparticles in the fluid has a major influence on heat transport. Indeed, increasing the volume percentage improves heat transfer by increasing the fluid temperature and hence the Nusselt number. This enhancement in heat transport is more pronounced at low Reynolds numbers than at high ones. Heat transmission improves significantly at ϕ = 0.1 (47%) compared to ϕ = 0. These discoveries appear to be valuable for engineers and manufacturers that want to create more efficient solar collectors.

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