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ARTICLE

Numerical Investigation on Heat Transfer Enhancement Inside a Rectangular Microchannel with Vortex Generator using TiO_2, Cuo-water Nanofluids

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ABSTRACT

One of the innovative ways to improve heat transfer properties of heat exchangers, is using nanofluids instead of traditional fluids. Due to presence of metal and oxides of metal particles in nanofluids structure, they have better potential in different environments and conditions than conventional fluids and having higher thermal conductivity causes improvements in heat transfer properties. In this research flow of two different nanofluids through a rectangular microchannel containing a different number of longitudinal vortex generators (lvgs), has been investigated. Simulations conducted under laminar flow boundary condition and for varied Reynolds numbers of 100 to 250. Considered volumetric concentration in this paper is 1, 1/6 and 2/3 %. Results showed, nanofluids and the LVGs notably improve the heat transfer rates within the microchannel. havg improved with increasing the nanoparticles volume concentrations and Reynolds number, while the opposite trends recognized for pressure drop. havg improved for 4 to 12 and 9 to 18% for TiO_2 and CuO nanofluids, respectively for different volume concentrations in simple microchannel. For lvg-enhanced microchannel the amount of improvements is about 9-14 and 5-10% for CuO and TiO_2, respectively. Also using vortex generators alone improved havg for 15-25% for different number of lvgs.

1. Survey

Heat exchangers are a significant part of numerous industrial uses and liable for heat transfer within fluids. Their utilizations involve air conditioning systems, gas turbine coolers, etc. Using of microchannel heat exchangers and nanofluids as an operating fluid has attracted many researchers to improve heat transfer performance in recent years. According to these subjects:

Toh et al. [1] examined heat transfer happenings and fluid flow within a microchannel under constant heat flux; it was noticed that heat inputs decrease the frictional disadvantages, especially at lower Re. Heris et al. [2] experimentally studied the forced flow of Al_2O_3/water inside a tube. The convection heat transfer of nanofluid analyzed. The tube considered to be circular, and the laminar boundary condition was applied for fluid. Outcomes indicated the improvement of heat transfer via the presence of the nanoparticles in the fluid. Bianco et al. [3] investigated nanofluid forced convection in circu-

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lar tubes numerically. Results showed that heat transfer enhances with particle volume concentration, but is followed by growing wall shear stress states. he et al. [4] studied the convective heat transfer of TiO_2 nanofluids inside a vertical tube. effects of nanoparticles concentrations, Re, and numerous nanoparticles aggregates sizes investigated. Duangthongsuk and Wongwises [5] experimentally examined the heat transfer performance and pressure drop of TiO_2-water nanofluids. turbulent regime boundary condition applied inside a flat double tube counter-flow heat exchanger. Results revealed that the heat transfer coefficient rose when using nanofluid instead of conventional fluid and improved with raising the Re and particle concentrations. Demir et al. [6] studied the forced convection flow within a horizontal tube numerically. constant wall temperature condition was applied, and simulations conducted for two different nanofluids. outcomes proved the heat transfer improvement due to the bearing of nanoparticles in the fluid. Kalteh et al. [7] investigated nanofluid forced convection within a wide microchannel heat sink. both numerical and experimental analyses were conducted. For the numerical part of this study, a two-phase Eulerian-Eulerian method using the finite volume approach adopted. Result showed better agreement between experimental and the two-phase method compared to homogeneous modeling. Lotfi et al. [8] experimentally analyzed heat transfer improvement of MWNT-water nanofluid within a heat exchanger. a shell and tube heat exchanger was used in this paper. Carbon nanotubes manufactured by the use of catalytic chemical vapor deposition (CCVD) method over Co-Mo/MgO nanocatalyst. Outcomes indicated that heat transfer improves when using multi-walled nanotubes instead of conventional fluid. Fazeli et al. [9] investigated the heat transfer properties of a small heat sink, which uses SiO_2-water nanofluids as coolant numerically and experimentally. Experimental outcomes revealed that scattering SiO_2 nanoparticles in water improve the overall heat transfer coefficient remarkably while thermal immunity of heat sink decreases up to 10%. Nimafar et al. [10] experimentally investigated the mixing process of three different passive micromixers and presented a new type of micromixers called the H micromixers. in another study Nimafar et al. [11] investigated separation and recombination micromixer in confronting with basic T and O type micromixers experimentally. Kabeel et al. [12] studied the impacts of using nanoparticles on corrugated plate heat exchanger performance. An experimental test loop has formed to study the PHE thermal properties such as heat transfer coefficient and pressure

Nomenclature			
		subscripts	
C	Correction coefficient	μ	Dynamic viscosity(kg/m.s)
C_p	Specific heat(j/kg.k)	φ	Particle volume concentration
C_d	Drag coefficient	ht	Heated zone
D_h	Hydraulic diameter(m)	nf	nanofluid
dlvg	Diameter of longitudinal vortex generators	p	Nanoparticle
N	Normal vector of surface		
Nu	Nusselt number		
P	Pressure(pa)		
Pr	Prandtl number		
F	Fanning friction factor		
G	Gravity(m/s2)		
H	Heat transfer coefficient(w/m2.k)		
K	Thermal conductivity(w/m.k)		
L	Length(m)		
q"	Heat flux(w/m2)		
Re	Reynolds number		
T	Time(s)		
T	Temperature(K)		
υ	Velocity vector(m/s)		
X	Distance from the heated zone inlet		
V	Volume(m3)		

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drop for various concentrated volume fractions of Al_2O_3 nanoparticles. Shkarah et al. [13] studied heat transfer in a microchannel heat sink. graphene, aluminum, and silicon substrates used for this study. Results from this numerical study showed that graphene most effectively reduced the thermal resistance. Viktorov and Nimafar [14] presented a new generation of 3D splitting and recombination (SAR) passive micromixer with microstructures placed on the top, and bottom floors of microchannels called a chain mixer. to analyze the flow structure of this type of passive micromixer, the mixing performance and pressure drop of the microchannel both experimental and numerical methods conducted. Wael aly [15] performed a CFD research to analyze the heat transfer and pressure drop characteristics of water-based Al_2O_3 nanofluid inside coiled tubes in tube heat exchangers. Results presented various behavior depending on the parameter selected for the comparison with the base fluid. Furthermore, analyzing at the same Reynolds or Dn showed the heat transfer coefficient improvements by raising the coil diameter and nanoparticle volume concentration. Bianco et al. [16] analyzed the turbulent convection of Al_2O_3-water nanofluid inside a round tube numerically employing a mixture model. Zhang et al. [17] experimentally studied TiO_2-water nanofluid single-phase flow and heat transfer features in a multiport channel flat tube. Results showed that friction factor and Nu of the nanofluids were higher than water, and both nanofluid's density and particle migration notably influenced the friction factor. Tiwari et al. [18] investigated heat transfer and fluid stream of CeO_2 and Al_2O_3 nanofluids inside corrugated chevron plates, plate heat exchanger as a homogeneous mixture numerically. CFD simulations showed that the corrugation design of the plate amplifies turbulence and vortices of fluid, which results in great heat transfer rates. Sarafraz and Hormozi [19] investigated the heat transfer and pressure drop properties of multi-walled carbon nanotube (MWCNT) aqueous nanofluids within a plate heat exchanger experimentally. Results showed that the heat transfer coefficient could enhance by increasing the flow rate and concentration of nanoparticles. Khajeh arzani et al. [20] investigated thermophysical characteristics, heat transfer and pressure drop of covalent and noncovalent functionalized graphene nanoplatelet based water nanofluids in a terete heat exchanger, experimentally, and numerically. Xia et al. [21] studied the properties of fluid flow and mass transfer in a new micromixer with rifts and baffles. both numerical and experimental methods used to validate the results. The impacts of rifts and baffles examined, considering both mixing performance and pressure drop at different Re. Sakanova et al. [22] investigated the curved channel formation and usage of nanofluids. Results showed that in case of using pure water as the coolant, the heat transfer performance of the channel notably improves compared with a conventional straight channel, while replacement of the pure water by nanofluids, made the effects of curved wall unnoticeable. Behrangzade and Heyhat [23] studied influences and potential of using dispersed nano-silver nanofluid with water as the base fluid inside a plate heat exchanger experimentally. Results showed significant improvements in the overall heat transfer coefficient, but pressure drop growth was not significant. Zarringhalam et al. [24] investigated the flow of CuO nanofluid experimentally. The influences of the Reynolds number and volume fraction on the heat transfer coefficient and pressure drop examined in this article. Like other studies, outcomes showed that heat transfer coefficient rises during using nanofluid instead of conventional fluids. It was observed that heat transfer coefficient and Nu increases with raising the substantial volume fraction and Re. Although the increase rate was lower at higher Reynolds numbers. Ebrahimnia-bajestan et al. [25] studied the effects of using TiO_2-water nanofluid as operating fluid inside heat exchangers in solar systems. Both experimental and numerical methods used to validate the results. Results from both experimental and numerical procedures showed that the heat transfer coefficient rises with an increase in volume concentration and Re. Chen et al. [26] studied micromixers with serpentine microchannels and analyzed their mixing performance experimentally and numerically.

Square-wave, multi-wave, and zigzag structures were picked for study. Results from both numerical and experimental results showed that square wave serpentine micromixers are more efficient and flexible than the other two structures. Sheikholeslami and Nimafar [27] analyzed hydrothermal behavior of CuO water-based nanofluid inside a complex-shaped cavity. Impacts of different parameters such as Ra, volume fraction, and the number of undulations investigated and the homogenous model used to simulate nanofluid. Fsadni et al. [28] studied the turbulent flow of Al_2O_3 nanofluid through a helically coiled rectangular-circular tube heat exchanger numerically. The heat exchanger was under constant wall heat flux, and the heat transfer and pressure drop features of this heat exchanger investigated under different conditions. Zhang et al. [29] used micro fin structure and nanofluid to enhance heat transfer performance of a microchannel. They studied heat transfer and pressure drop inside the microchannel, experimentally. Outcomes revealed that with increasing the number of fins, Nu and friction factor increases. It was found that micro fin and nanofluid techniques are both efficient ways to improve the heat transfer performance of microchannel. Rao et al. [30] studied Al_2O_3 nanofluid forced convective heat transfer coefficient under turbulent flow boundary condition inside a single pass, multi-tube, counterflow shell and tube heat exchanger. Results revealed higher forced convection than water flow at the same conditions. Diao et al. [31] investigated the heat transfer and flow of MWCNT-water nanofluid inside a multiport microchannel with smooth and micro fin structure using experimental methods. Results showed that the heat transfer is higher in the micro fin tubes compared with smooth surface, but the heat transfer enhancement was lower in micro fin tube. Baheri islami et al. [32] studied the mixing performance of non-newtonian nanofluids inside micromixers numerically using the mixture model. They found out that the number of injection flows and baffles can be very efficient in mixing performance. Sheikholeslami and Nimafar [33] studied nanofluid flow and heat transfer within two circular cylinders in the presence of a magnetic field. Multiple active parameters such as aspect ratio, Eckert number, Reynolds number and Hartmann number had examined. They found out that the temperature gradient develops with increasing Ha and Ec, and reduces with increasing Re. in another article [34] they reported the influences of melting heat transfer on nanofluid flowing, in the presence of Lorentz forces. Different shapes of nanoparticles considered. Results showed that the maximum Nusselt number occurs at platelet shape, and the temperature reduces with an increase in melting parameters.

In this research, flow and heat transfer performance of TiO_2-water and CuO-water nanofluids through a longitudinal rectangular microchannel using vortex generators inside investigated numerically. Results reported for different effective parameters such as Re, volume concentration, and the number of vortex generators (lvgs). Variations of heat transfer coefficient, Nu, and pressure drop reported using a single-phase model by simulating the flow with ANSYS-FLUENT software.

2. Problem Description and Mathematical Model

The conferred issue is a four-sided longitudinal microchannel to investigate the effects of using longitudinal vortex generators (lvgs) and nanofluids (TiO_2 and CuO) on heat transfer performance inside this microchannel. In order to do that, at first, a comparative simulation performed between the single-phase and two-phase models at similar conditions to evaluate the difference between results of these two models. Then the rest of simulations done under presented conditions and by using single-phase model using ANSYS-FLUENT software.

In this paper, the Cartesian coordinate system used to describe the flow of the fluid, which in this coordinate, the z-axis represents streamwise direction.

The geometry of the considered issue demonstrated in figure 1. also geometrical parameters used to define the geometry of the microchannel and the vortex generators presented in table 1.

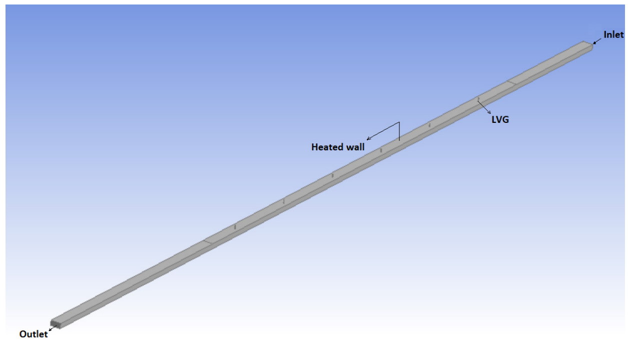

Figure 1. 3D view of investigated geometry

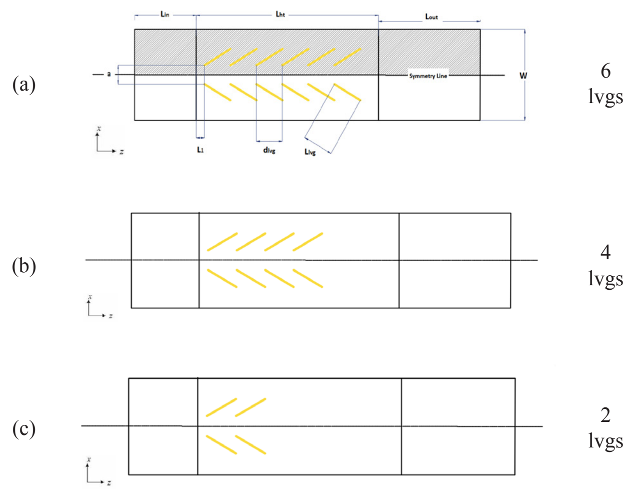

Figure 2. schematic of investigated geometry

Computational domain is divided into three zones:

(1) the inlet area with length of L_{in}, which assumed to be adiabatic and indicates the flow developing zone.

(2) the heated area, which is under constant heat flux and the vortex generators are located in this area.

(3) the outlet area, which eliminates affecting of any backflow on the final results.

Because of symmetric layout of the microchannel and to reduce the computational time and cost, only the schemed area in figure 2 (a) is considered for numerical investigation.

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Table 1. Geometrical parameters of the microchannel and vortex generators

Geometric parameter	amount	Geometric parameter	amount
L_{in}	2500 μm	L1	1000 μm
L_{ht}	10^4 μm	dlvg	1600 μm
L_{out}	5000 μm	Wlvg	10 μm
W	400 μm	Llvg	140 μm
H	100μm	a	80 μm
dh	160 μm	β	30

Microchannel and vortex generators in this paper are made of silicon plates and flow of water and CuO and TiO_2 nanofluids investigated inside this microchannel as coolant. Thermophysical properties of these material presented in table 2.

Table 2. Thermophysical characteristics of materials

	K (w/m.k)	Cp (j/kg.k)	ρ (kg/m3)	μ (Pa.s)
Pure water	-7/843*10^{-6} + 0/0062R – 0/54	6719/637 – 20/86T + 0/0552T^2 – 0/0000462T^3	816/781 + 1/505T – 0/003T2	0/00002414*$10^{247/8/(t-140)}$
TiO2	8/4	710	4157	
CuO	76/5	536/6	6350	
silicon	290-0.4T	390+0.9T	2330	

Some of the other assumptions that considered in this paper are as follow:

(1) surface roughness considered zero for microchannel walls and lvgs.

(2) flow considered steady, Newtonian and laminar because of low velocities of fluid and small dimension of vortex generators.

(3) radiative heat transfer, compressibility and the effects of body forces are neglected.

Fluid flow investigated for 100, 150, 200 and 250 Reynolds numbers and for 3 different volume concentration of 1, 1/6 and 2/3 percent, inside flat microchannel and microchannel with 2, 4 and 6 pairs of vortex generators. The inlet temperature fixed at 298k and the pressure outlet condition considered for outlet. Top wall is under constant heat flux of 20 w/cm^3 and considered the heated zone of the microchannel. Non-slip and adiabatic conditions assumed for other surfaces of the microchannel.

To compare results achieved from single-phase and two-phase model, an Eulerian method assumed for each phase of nanofluid (base fluid and nanoparticles) and equations solved separately using the Eulerian-Eulerian model. Continuity, momentum and energy equations derived from ANSYS-FLUENT software [35] are as follow:

$$Vi=\int \varphi_i \, dV \tag{1}$$

In this equation is volume of phase i. and the relation between volume concentrations of both phases described with next equation:

$$\varphi_f+\varphi_p=1 \tag{2}$$

Continuity and momentum equations are as follow:

$$\nabla.\left(\varphi_i\ \rho_i\,\vec{\vartheta}_i\right)=0 \tag{3}$$

$$\nabla.\left(\varphi_f\rho_f\overrightarrow{\vartheta_f}\overrightarrow{\vartheta_f}\right)=-\varphi_f\nabla p+\nabla.\overline{\overline{\tau_f}}+\overrightarrow{R_{pf}} \tag{4}$$

$$\nabla.\left(\varphi_p\rho_p\overrightarrow{\vartheta_p}\overrightarrow{\vartheta_p}\right)=-\varphi_p\nabla p+\nabla.\overline{\overline{\tau_p}}+\overrightarrow{R_{pf}} \tag{5}$$

Which momentum equation is written for each phase separately in equation (4) and (5) in which in these equations is the i_{th} phase stress strain tensor that defined by equation (6) and is the interaction force between two phases of nanofluid.

$$\overline{\overline{\tau_i}}=\mu_i\nabla\overrightarrow{\vartheta_i} \tag{6}$$

$$\overrightarrow{R_{pf}}=K_{pf}\left(\overrightarrow{\vartheta_p}-\overrightarrow{\vartheta_f}\right) \tag{7}$$

K_{pf} in equation (7) is the interphase momentum exchange coefficient:

$$K_{pf}=(\varphi_f\,\varphi_p\,\rho_p\,C_D\,Re_p)/(24\tau_p) \tag{8}$$

τ_p and C_D in equation (8) are the particulate relaxation time and drag coefficient, respectively which described by:

$$\tau_p=\rho_p d_p^2/18\mu_f \tag{9}$$

$$C_D=\begin{cases}\dfrac{24\left(1+0.15Re_p^{0.687}\right)}{Re_p}, & Re\le 1000\\ 0.44, & Re>1000\end{cases} \tag{10}$$

(Schiller-Naumann model)

Re_p, in equation (10) is the relative Re and written as follows:

$$Re_p=\frac{\rho_f\left|\overrightarrow{\vartheta_p}-\overrightarrow{\vartheta_f}\right|d_p}{\mu_f} \tag{11}$$

In order to apply energy equation in Eulerian-Eulerian model, enthalpy equation should written for each phase separately:

$$\nabla.\left(\varphi_f\rho_f h_f\overrightarrow{\vartheta_f}\right)=\overline{\overline{\tau_f}}:\nabla\overrightarrow{\vartheta_f}-\nabla.\overrightarrow{q_f}+Q_{pf} \tag{12}$$

$$\nabla.\left(\varphi_p\rho_p h_p\overrightarrow{\vartheta_p}\right)=\overline{\overline{\tau_p}}:\nabla\overrightarrow{\vartheta_p}-\nabla.\overrightarrow{q_p}+Q_{pf} \tag{13}$$

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In these equations, and are heat flux and specific enthalpy of phase i, respectively and or the heat exchanger intensity of two phases described with equation (14) as follows:

$$Q_{pf}=h_V\,(T_f-T_p) \tag{14}$$

h_V, or the volumetric heat transfer coefficient described with the next equation:

$$h_V = C \times \frac{(6k_f\varphi_p\varphi_f Nu_p)}{d_p^2} \tag{15}$$

$$C = (|5.505 - 9.606 \times 10^{-6}Re_e^2 + 1.539 \times 10^{-2}Re - 3.073 \times 10^2\varphi|) \times 10^{-9} \tag{16}$$

C in equation (16) is a correction coefficient factor which Ebrahimnia et al[25] proposed to improve the accuracy of the Eulerian-Eulerian model. Also in equation (15), Nu_p is the nanoparticle nusselt number which describes with Ranz-Marshall correlation[36] as follows:

$$Nu_p = 2 + 0.6Re_p^{0.5}Pr_f^{1/3} \tag{17}$$

$$Pr_f = \frac{C_{p,f}\mu_f}{k_f} \tag{18}$$

Pr_f, is the base fluid prandtl number.

3. Numerical Procedure

To estimate the heat transfer performance and fluid flow inside described rectangular microchannel with vortex generators, a single-phase model used to simulate the flow inside ANSYS-FLUENT software. To calculate the corresponding gradians, such as u, ʋ, w and T, a non-uniform tetrahedron structure (Figure 2) used to define and solve the computational domain. The SIMPLEC algorithm applied to model pressure-velocity coupling and the convergence criterion for this problem considered 10^{-6}, which this value is 10^{-8} for the energy equation.

Some of the important parameters of the microchannel defined as follows:

$$Re = \frac{\rho_{in}\vartheta_{in}D_h}{\mu_{in}} \tag{19}$$

$$Nu = \frac{hD_h}{k} \tag{20}$$

$$D_h = \frac{2WH}{W+H} \tag{21}$$

$$h = \frac{q"}{T_{wall,avg} - (T_{in} + T_{out})/2} \tag{22}$$

$$\Delta p = (\overline{p_{in}} - \overline{p_{out}}) \tag{23}$$

In these equations, Re is the Reynolds number, D_h is the hydraulic diameter of the microchannel, Nu and h are the Nusselt number and heat transfer coefficient, respectively and Δp is pressure drop inside microchannel.

4. Validation and Grid Independency

In order to find the least computationally expensive mesh structure, a few number of networks with different cell numbers used. In this paper, the grid independence test has done under Reynolds number of 250 while pure water flows inside the microchannel as coolant. Figure 4 represents the results gained for various mesh sizes and their diversity. Considering the difference between the results are less than 1 percent for structures with 1/8 million cells, then the mesh size of 2 million cells seemed to be suitable for all the other simulations performed in this study.

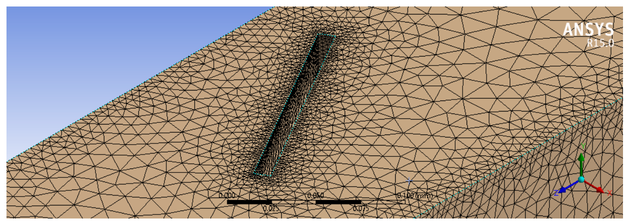

Figure 3. 3d view of mesh network

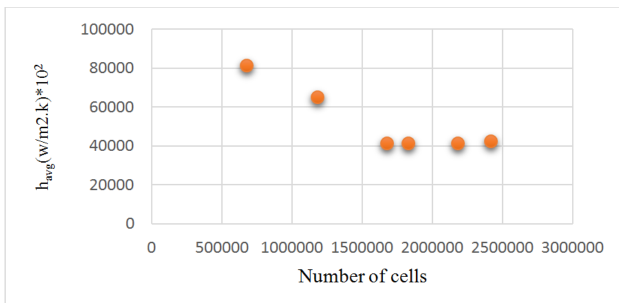

Figure 4. Grid independency test

To verify the obtained outcomes from the presented numerical method, the experimental data from Ebrahimnia et al. [25] for TiO2-water nanofluid is used. Figure 5 represents the comparison between the results of this study and the results of Ebrahimnia et al. [25]. As presented in Figure 5, the deviation between results of this paper and experimental data is about 5 to 7 %, which is acceptable. These negligible differences can be because of several factors such as simplification of physical models, using of single-phase model instead of two-phase model, experimental measurements and non-uniformity of nanoparticle sizes.

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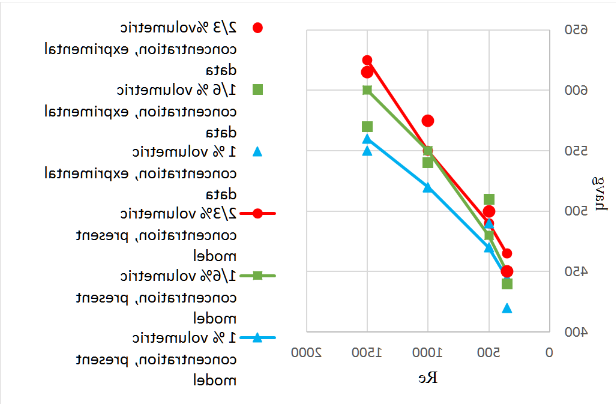

Figure 5. Validation

5. Result and discussions

In this paper, results provided for different parameters such as, convective heat transfer coefficient, Nu and pressure drop. The mentioned parameters are demonstrated as functions of Re, nanoparticle volume concentration and number of lvgs used in the microchannel.

The first figure as was promised, presented the comparison between single-phase and two-phase models at the same conditions at constant volume concentration of 2/3% and different values of Reynolds number while CuO nanofluid flows through microchannel with 6 pair of lvgs. As demonstrated in Figure 6, there is a good proportion between results of the used models at the considered conditions and the disagreement between results are less than 5 percent.

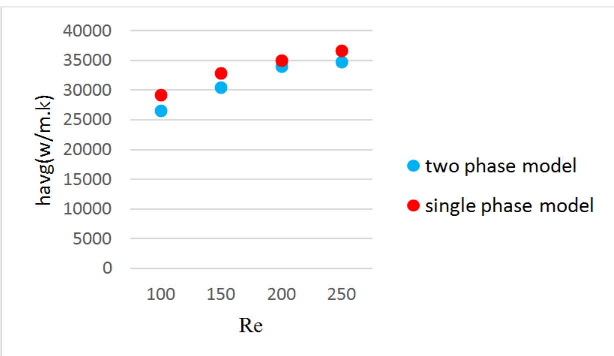

Figure 6. Comparison between two-phase and single-phase models

After comparison between these two models, the rest of simulations done under presented conditions and for single-phase model. Figs 7 describes the varieties of convective heat transfer coefficient against Rer for microchannel with different number of lvgs while pure water flows through microchannel as coolant. As expected using of vortex generators leads to higher heat transfer rates and this value increases by increasing the number of lvgs. Results indicate an improvement in the heat transfer coefficient of 22-35, 18-31 and 15-27 percent for microchannels with 6,4 and 2 pair of lvgs, respectively with variation of Re. small difference between heat transfer coefficient for plain microchannel and microchannel with lvgs at low Reynolds numbers is noticable, but this difference increases at higher Reynolds numbers.

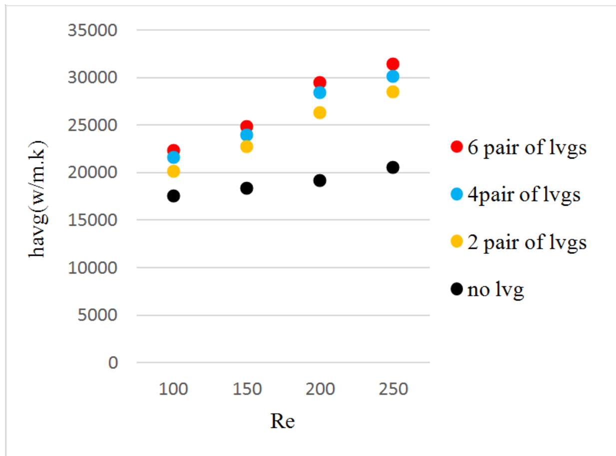

Figure 7. Variations of h_{avg} for different number of lvgs

To investigate the flow inside plain microchannel and lvg-enhanced microchannel more closely, figures 8 and 9 displays velocity and temperature contours inside both microchannels, respectively.

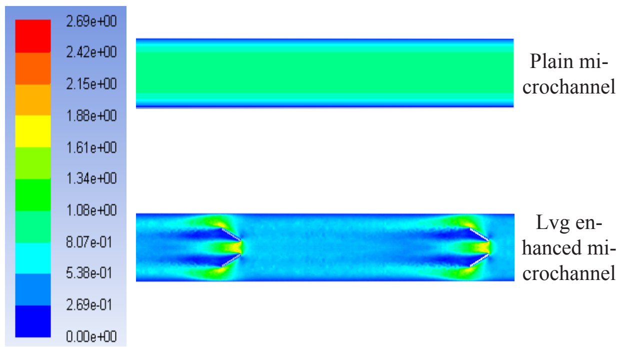

Figure 8. Velocity cantour for different channels

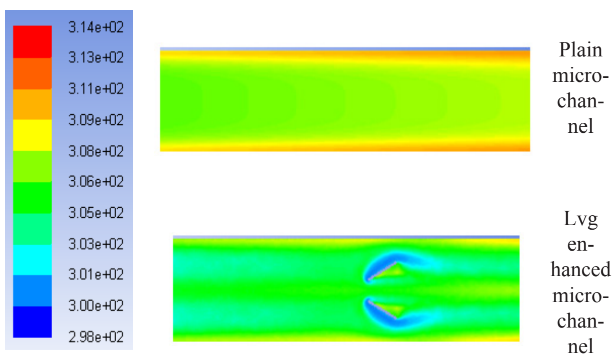

Figure 9. Temprature cantours for different channels

Figures 10 and 11 represent the variation of convective heat transfer coefficient versus the plurality of Re for different volume concentrations inside microchannel with six pair of vortex generators and plain microchannel, while TiO_2 and CuO nanofluids flowing through, respectively. As was displayed in figure 7, Using lvgs increases the h_{avg} inside the microchannel, and as expected, using nanofluids instead of usual fluids such as water causes even greater heat transfer rates, also heat transfer rate rises with an increase in Re and volume concentration of nanoparticles. Using vortex generators inside microchannel creates vortices and wavy patterns back of lvgs, which this phenomenon leads to having even better mixed-flow, which causes greater heat transfer rates. Also, using nanofluids leads to greater heat transfer rates because of their better thermophysical properties, and higher Reynolds numbers reduce thermal boundary layer thickness behind lvgs and stronger and wider vortices, which also causes greater heat transfer rates. At fixed Re, it can be seen that with raising the volume concentration of nanoparticles, the effective thermal conductivity of fluid increases, which causes greater heat transfer rates because of higher thermal energy transfer because of the dispersion of fluid.

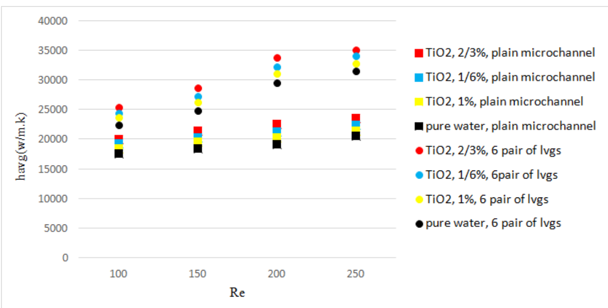

Figure 10. Variations of h_{avg} for diiferent volume concentrations of TiO2 nanofluid inside plain and lvg-enhanced microchannels

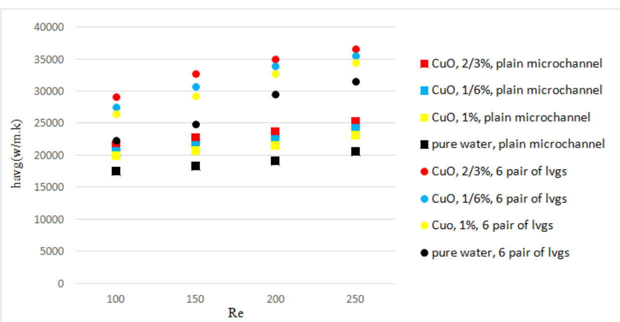

Figure 11. Variations of h_{avg} for diiferent volume concentrations of CuO nanofluid inside plain and lvg-enhanced microchannels

Results from flow of TiO_2 and CuO nanofluid inside plain microchannel and lvg-enhanced microchannel, showed enhancement in heat transfer performance of microchannel. As demonstrated in Figure 10 and 11, using TiO_2 nanofluid alone, increases heat transfer coefficient for about 4-12 percent which this value is about 9-18 percent for CuO nanofluid inside plain microchannel. Using both nanofluid and vortex generators causes even more increase in heat transfer coefficient, which is about 25-40 and 30-43 percent for TiO_2 and CuO nanofluids, respectively.

To compare the flow of three fluids used in this paper, figure 12 describes the varieties of convective h_{avg} for different Re for water and TiO2 and CuO nanofluids at a constant volume concentration of 2/3%. After Evaluating the results, it was found that the best performance happens when using CuO nanofluid as coolant flowing inside microchannel, either for lvg-enhanced or plain microchannel. Results demonstrates about 2-5 percent higher heat transfer coefficient when using CuO nanofluid compared to TiO_2 at different Reynolds number and volume concentrations.

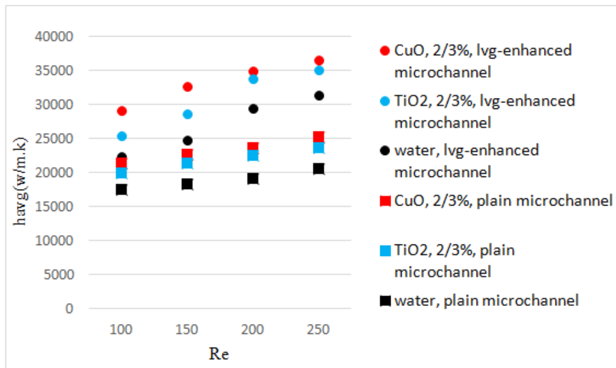

Figure 12. Comparison between results of two different nanofluids for plain and lvg-enhanced microchannels

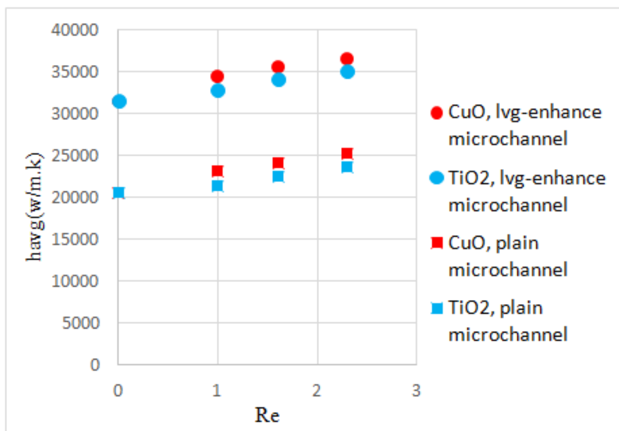

Figure 13. Variations of h_{avg} at constant Reynolds numbers for different volume concentrations

Figure 13 shows the variations of h_{avg} for different volume concentrations at Re=250, while TiO2 and CuO nanofluids flow through plain microchannel and

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microchannel with six pairs of lvgs. As explained before, h_{avg} rises with an increase in nanoparticle volume concentration because of the increase of effective thermal conductivity of the fluid and higher thermal energy transfer. Figure 10 also shows better performance of CuO nanofluid compared to TiO2 nanofluid more obviously.

Another parameter that examined in this paper is the nusselt number. Figure 14 demonstrates varieties of Nu for different Re inside microchannels with a different number of vortex generators while water flows through as coolant. As was expected, Nu increases with an increase in Re and number of lvgs. The enhancement is about 12-30% for variations of Re and lvgs.

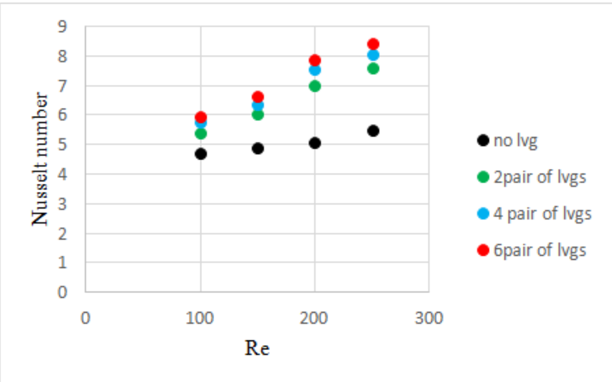

Figure 14. Variations of Nu for different number of lvgs

Figure 15 demonstrates variations of Nu versus different values of nanoparticle volume concentration for microchannel with six pairs of lvgs while TiO2 and CuO nanofluids flow through as coolant. Results show an increase in Nu by increasing the volume concentration of nanoparticles. The Nu increase is about 15% and 19% for TiO2 and CuO nanofluids, respectively.

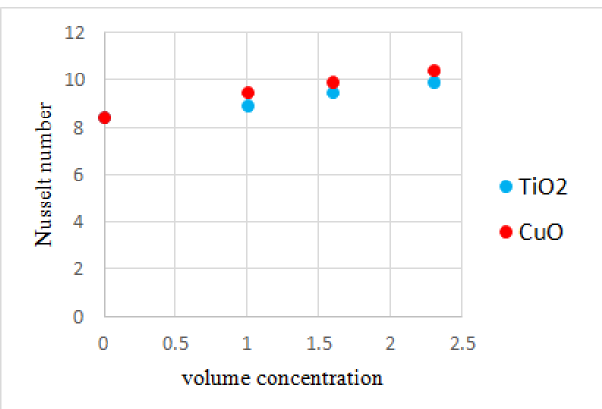

Figure 15. Variations of Nu at constant Re for different volume concentrations

One of the important parameters that should be investigated for flow inside microchannel, is pressure drop. Figure 16 shows the variations of pressure drop versus different nanoparticle volume concentrations inside lvg-enhanced microchannel at constant Reynolds number of 250 for TiO_2 and CuO nanofluids. As results show, by increasing the nanoparticle volume concentration, pressure drop inside microchannel increases for both nanofluids which is an adverse feature inside microchannel. This means using of nanofluids causes higher pressure drop inside microchannel, but due to small amount if increase in pressure drop than heat transfer rate enhancement, it can be neglected. Also results show that using vortex generators can also increase the pressure drop inside microchannel which can be neglected.

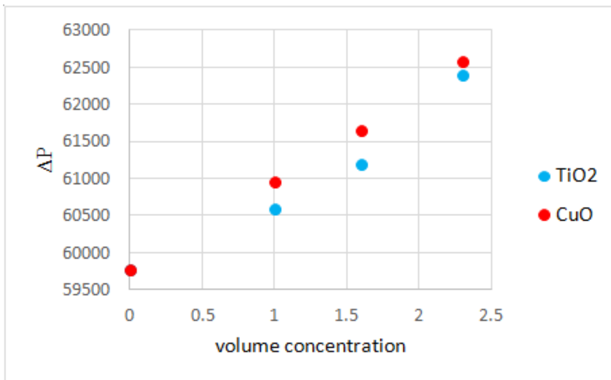

Figure 16. Variations of pressure drop for different volume concentrations

6. Conclusion

In this research, the flow and characteristics of two separate nanofluids inside a four-sided microchannel and the effects of using longitudinal vortex generators on the heat transfer performance of this microchannel investigated. A single-phase model employed to simulate nanofluids flow using ANSYS-FLUENT software. Summary of the findings of this study are as follows:

(1) Using nanofluids instead of common fluids like water, enhances heat transfer performance inside microchannel.

(2) Heat transfer coefficient increases by increasing the nanoparticle volume concentration and Re

(3) Using TiO2 and CuO nanofluids increased the heat transfer coefficient for about 4-12 and 9-18 percent for different volume concentrations, respectively.

(4) CuO nanofluids showed better performance inside microchannel than TiO2 nanofluid. The variation of the measured heat transfer coefficient was about 5%.

(5) Results showed that using vortex generators notably improves heat transfer rates inside the microchannel. It was found out that increasing the number of vortex gener-

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ators increases the heat transfer coefficient. This phenomenon can happen because of having a better-mixed flow, which causes greater heat transfer rates.

(6) Using vortex generators alone increases the heat transfer coefficient for about 15-35 percent for different number of lvgs. This value is about 25-45 percent when using nanofluids as a coolant inside microchannel.

(7) Using nanofluids and lvgs also causes an increase in pressure drop, which can be neglected.

7. Declarations

Authors' contributions

Arash Behaeen was in charge of the whole manuscript and wrote it. Mohammad Nimafar was the corresponding author and assisted with all the scientific issues and analysis.

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Arash Behaeen, born in 1991 in Iran, received his master's degree in mechanical engineering from Central Tehran branch Islamic Azad University in 2017. His research interests include solar combi systems, fluid flow, and heat transfer of nanofluids and microchannels.

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Competing Interests:

The authors claim that they have no competing concerns.

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