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INVESTIGATION OF COMBINED EFFECT OF NANOFLUID AND SWIRLING JET ON HEAT TRANSFER

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ABSTRACT

The present study is focused on the numerical investigation of heat transfer from a heated surface by using nanofluids and swirling jets. Effects of different Reynolds number and different inlet temperature on heat transfer and fluid flow were studied numerically. Al_2O_3-H_2O nanofluid was used as a base coolant in all parameters. k-ω turbulent model of PHOENICS computational fluid dynamics code was used for numerical analysis. It is obtained that increasing Reynolds number from Re=12000 to 21000 causes an increase of 51.3% on average Nusselt Number. Increasing inlet temperature from T_{inlet}=5°C to 30°C has not a significant effect on average Nusselt number.

1. INTRODUCTION

One of the main problems of design parameter for new technological devices are heat loads. Using swirling flow with nanofluids can be a key solution to solve this heat loads problem. So, swirling flows have gained an increasing interest in the last decade in fluid dynamics research, since in many technical applications swirl is an essential phenomenon. Swirling flows are generally used in the industry for separation, mixing and flame stabilization. The main characteristic of swirling flow is the combination of axial and tangential velocities. Swirling flows are used to minimize pressure drop and prevent particle deposition (Kharoua et al.[1]). In swirling flows, part of the fluid enters axially while the remainder is injected tangentially at various locations along the tube axis. The radial pressure gradient results in thinning of thermal boundary later with an accompanying improvement in heta transfer (Chang et al. [2]). A nanofluid is defined as a suspension of solid particles which have 1-100 nm size in a base fluid. In heat transfer applications using nanofluid, the particles suspended in the base fluid, expand thermal capacity of the fluid. Interactions and collisions between particles cause to increase in turbulence and turbulence intensity of the transition surface. Turbulence intensity and large surface area enables more heat transfer. Nanoparticles carry 20% of their atoms at the surface that makes them ready to heat transfer. Another advantage of using nanofluids is the particle agitation cause micro-convection in the fluid due to its very small size and therefore increases the heat transfer in particular heat transfer from surfaces with high heat flux, as well as industry, medicine and space research. But main disadvanteges of using nanofluid may be pressure drop an particle deposition in pipes or ducts. So using nanofuids with swirling jets may be a key solution for these disadvanteges.

Many studies on enhancing heat transfer technique as impinging jets or swirling jets with different coolant can be found in the literature. Kilic et al. [3, 4] surveyed the cooling of a flat plate with the support of the impinging fluid air jet for different Reynolds numbers and dimensionless channel heights. The average Nusselt number was found to increase by 49.5% in *Re* = 4000-10000 and 17.9% in H/D_h = 4-10. Teamah et al. [5] investigated heat transfer and flow structure formed by Al_2O_3 nanofluid to flat plate by experimentally and numerically with various Re number (*Re*=3000-32000) and different volume ratio of nanofluids (φ=0-10%). As the nanoparticles in the base fluid increases, the heat transfer from the surface increases and heat transfer coefficient can be enhanced by 62% according to the

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water is used base fluid only. They observed that heat transfer can be increased of 8.9% by using CuO-water with respect to Al_2O_3-water. Sun et al. [6] researched the effect of a single impinging jet using Cu-water nanofluids as working fluid on heat transfer. It has been determined that when the nanofluid is used, important enhancement can be achieved in heat transfer with respect to the use of water only, no significant change in pressure drop, a higher heat transfer coefficient is obtained when a circular nozzle is used, and a higher heat transfer coefficient is obtained when the jet angle is 90°. Kilic et al. [7] investigated the heat transfer from a high heat flux surfaces for different parameters using nanofluids and multiple impinging jets. It was found that increase on Re number and decrease on particle diameter causes an increase on heat transfer. The use of Cu-water nanofluid causes an increase of 9.3% and 8.4% on heat transfer according to the use of Al_2O_3-water and TiO-water nanofluid.Sekrani et al. [8] investigated turbulent convective heat transfer of Al_2O_3 nanofluid flowing in a circular tube with uniform heat flux for different turbulenc model numerically. Their results show that SST k-ω model perfoms best for predicting average Nusselt number and friction factor according to the other turbulence models. Wongcharee et al. [9] investigated effects of swirling impinging jets with TiO_2-water nanofluid on heat transfer for different jet-to-target speacing ratio, Reynolds number and volume consantration. Their results show that increasing volume ratio up to the 2% causes an increase on average Nusselt number but while the one with concentration of 2.5% shows opposite results. Akyurek et al. [10] studied on turbulent forced convection heat transfer and pressure drop characteristic of Al_2O_3-water nanofluid experimentally by using wire coil turbulators. They obtained that average Nusselt number increases with increasing the Reynolds number and particle concentration. The pressure drop of the Al_2O_3-water nanofluid showed neqarly equal to that of oure water at the same Reynolds number rage. Using wire coil turbulators increase pressure drop as well as the heat transfer coefficient.

This study is different from the studies at literature by evaluating combined effect of swirling jets and nanofluids on heat transfer for different parameters to prevent the disadvanteges (pressure drop and particle deposition) of nanofluids. By using combined effect of nanofluids and swirling flows heat transfer enhancement was evaluated for different Reynolds number and different inlet temperature. Numerical results was also validated with the experimental results in literature.

2. NUMERICAL MODEL

Compuatitional domain consists of a rectangular channel of which dimensions are 10x10x50 mm. Two jet inlets of which dimensions are 3x3 mm were located at inverse direction to inject tangentially. There is one outlet at the end of the channel. All walls has a constant wall temperature. Inlet velocity of swirling jets are applied according to the Reynolds number. Inlet temperature of fluid is changed according to the different inlet temperature. k-ω turbulence model of PHOENICS CFD code was used for this numerical analysis. CFD simulation domain is shown in Fig.1 and mesh structure is shown in Fig.2.

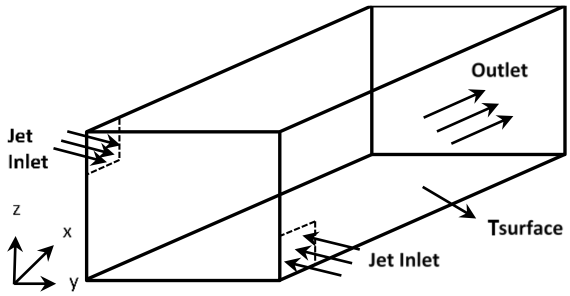

Fig.1. *CFD Simulation model*

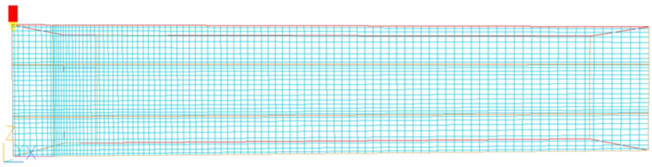

Fig.2. *Mesh Structure*

The continuity, Reynolds averaged momentum and time averaged energy equations governing 3-dimensional steady, flow of fluid with constant properties used for turbulent solutions can be written in the Cartesian coordinate system as follows;

Continuity equation:

$$\frac{\partial U_i}{\partial x_i} = 0 \tag{1}$$

Momentum equation:

$$\rho U_i \frac{\partial U_j}{\partial x_i} = -\frac{\partial P}{\partial x_j} + \frac{\partial}{\partial x_i}\left[\mu\left(\frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i}\right) - \rho\overline{u_i' u_j'}\right] \tag{2}$$

Energy equation:

$$\rho c_P U_i \frac{\partial T}{\partial x_i} = \frac{\partial}{\partial x_i}\left[k\frac{\partial T}{\partial x_i} - \rho c_P \overline{u_i' T'}\right] \tag{3}$$

where Uτ is the resultant fricon velocity ($U\tau = \sqrt{(\tau_w/\rho)}$, τ_w is the wall shear stress, δ is the normal distance of the first grid point from the wall, and k is von Karman's constant. Aditionally, Vinlet is the bulk inlet velocity, I is the turbulent intensity (typically in the range 0.01<I<0.05), ε is the dissipaon rate, C_d is the constant (C_d=0.09) and the mixing length L_m~0.1H, where H is a characterisc inlet dimension.

Table 1. *Boundary Conditions*

	U (m/s)	V (m/s)	W (m/s)	T °C	k	ω
Inlet	$U = 0$	$V = V_{inlet}$	$W = 0$	$T = T_{inlet}$	$k = (I \cdot V_{inlet})$	$\omega = \varepsilon/(C_d \cdot k)$
Outlet	$\frac{\partial U}{\partial x} = 0$	$\frac{\partial V}{\partial y} = 0$	$\frac{\partial W}{\partial z} = 0$	$T = T_{outlet}$	$\frac{\partial k}{\partial z} = 0$	$\frac{\partial \omega}{\partial z} = 0$
Bottom Wall	$U = 0$	$V = 0$	$W = 0$	$T = T_{surface}$	$k = U\tau^2/\sqrt{C_d}$	$\omega = U\tau/(\sqrt{C_d}\kappa\delta)$
Top Wall	$U = 0$	$V = 0$	$W = 0$	$T = T_{surface}$	$k = U\tau^2/\sqrt{C_d}$	$\omega = U\tau/(\sqrt{C_d}\kappa\delta)$
Front and Backwall	$U = 0$	$V = 0$	$W = 0$	$T = T_{surface}$	$k = U\tau^2/\sqrt{C_d}$	$\omega = U\tau/(\sqrt{C_d}\kappa\delta)$

3. DATA REDUCTION

The heat transfer from the surfaces to the fluid will take place by convection, conduction and radiation.

$$\dot{Q}_{convection} = \dot{Q}_{total} - \dot{Q}_{conduction} - \dot{Q}_{radiaiton} \tag{4}$$

Heat transfer occurs from high temperature walls to the nanaofluid. So heat transfer rate gained by nanaofluid equals to the heat transfer rate lost by walls. As a result, walls surfaces will be cooled by using nanofluid with swirling jet. It is assume that channel walls has constant temperature. So heat conduction through the walls (from outer surface to the inner surface) is neglected. Heat transfer with radiation is negligible in this study because surface temperature is under 573.15 K.

Heat transfer rate gained by the fluid from wall surfaces;

$$\dot{Q} = \dot{m}C_{p_{nf}}\,(T_{outlet} - T_{inlet}) \tag{5}$$

Where $\dot{m}$ is mass flow rate of nanaofluid, $C_{P_{nf}}$. is nanaofluid specific heat, T_{outlet} is outlet temperature of nanaofluid and T_{inlet} is inlet temperature of nanofluid. Nusselt number (Nu) is a dimensionless parameter indicating the ratio of heat transfer with convection to heat transfer with conduction. Averaege Nusselt number can be presented as ratio of average heat transfer coefficient times characteristic length (D_h) to the coefficient of thermal conductivity of the nanofluid.

$$Nu_{avg} = \frac{h_{avg} \cdot D_h}{k_{nf}} \tag{6}$$

Where h_{avg} is the average heat transfer coefficient, measured, D_h is the hydraulic diameter, and k_{nf} is the coefficient of thermal conductivity of the nanofluid. Averaege heat transfer coefficient can be presented as;

$$h_{avg} = \dot{Q}/(A_s.T_{lm}) \tag{7}$$

Where A_s is the convection surface area and T_{lm} is logarithmic mean temperature of nanaofluid.

$$T_{lm} = (\Delta T_e - \Delta T_i)/\ln\left(\frac{\Delta T_e}{\Delta T_i}\right) \tag{8}$$

Where ΔT_e is temperature difference between surface temperature and nanaofluid exit temperature $\Delta T_e = (T_s - T_e)$ and ΔT_i is temperature difference between surface temperature and nanaofluid inlet temperature $(\Delta T_i = (T_s - T_i)$. Reynolds number ($Re$) is used to determine for forced convection whether the flow is laminar or turbulent. Reynolds number based on turbulent flow;

$$Re = \frac{\rho_{nf} V_{jet} D_h}{\mu_{nf}} \tag{9}$$

Where ρ_{nf} is the nanofluid density, V_{jet} is the jet velocity, and μ_{nf} is the nanofluid dynamic viscosity. The density of nanofluids is;

$$\rho_{nf} = (1-\varphi)\rho_{bf} + \varphi\rho_p \tag{10}$$

Where ρ_{bf} is the base fluid (water) density, φ is the volumetric ratio of the nanofluid, and ρ_p is the density of the solid particles in the nanofluid. The volumetric ratio of nanoparticles is;

$$\varphi = \frac{1}{(1/\omega)(\rho_p - \rho_{bf})} \tag{11}$$

Where ω is the density difference between the fluid and the main fluid (water). The nanofluid specific heat is calculated from;

$$C_{p_{nf}} = \frac{\varphi(\rho C_p)_p + (1-\varphi)(\rho C_P)_f}{\rho_{nf}} \tag{12}$$

Where $(C_p)_p$ is specific heat of particle $(C_p)_p$ is specific heat of base fluid. The effective thermal conductivity of nanofluid is calculated according to Corcione [11];

$$\frac{k_{eff}}{k_f} = 1 + 4.4Re^{0.4}Pr^{0.66}\left(\frac{T_{nf}}{T_{fr}}\right)^{10}\left(\frac{k_p}{k_f}\right)^{0.03}\varphi^{0.66} \tag{13}$$

Where Re is the nanoparticle Reynolds number, Pr is the Prandtl number of the base liquid. k_p is the nanoparticle thermal conductivity, φ is the volume ratio of the suspended nanoparticles, T_{nf} is the nanofluid temperature (K), T_{fr} is the freezing point of the base liquid. Nanoparticle Reynolds number is defined as;

$$Re = \frac{2\rho k_b f T}{\pi\mu f^2 d_p} \tag{14}$$

k_b is the Boltzmann's constant. The effective dynamic viscosity of nanofluids defined as;

$$\mu_{nf} = \mu_{bf}(1 + 2.5\varphi + 4.698\varphi^2) \quad (15)$$

4. RESULTS AND DISCUSSIONS

In this section, numerical results were prepared for two parameters. Effects of Al_2O_3-water nanofluid with 20 nm particle diameter and volume ratio of φ=4% on heat transfer for different Reynolds number (*Re*=12000, 15000, 18000, 21000) and for different inlet temperature (T_{inlet} =5, 10, 20, 30°C) were obtained.

4.1. Effects of different Reynolds number on heat transfer

To understand effect of fluid velocity on heat transfer, Reynolds number was increased from *Re*=12000 to 21000. Fig.3 and Fig.4 show velocity vector and temperature contours of fluid flow at the midpoint of inlet of swirling jet, in the middle of channel and outlet of the channel at *x*-direction (x/D_h=0.03, 8, 16) for *Re*=12000 and 21000 respectively.

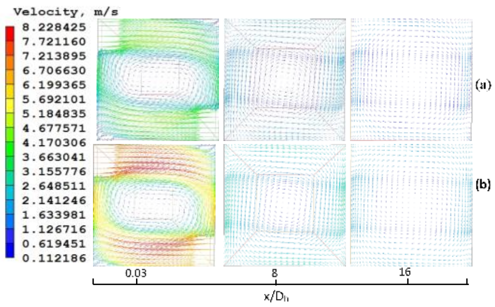

Fig.3. *Velocity vectors for different Reynolds Number (a) Re=12000 and (b) Re= 21000*

Thermal boundary region does not occure at the inlet of swirling jet because of the high turbulence intensity at the surfaces of the channel. Temperature increase can be seen only at the impinging region of swirling jet because fluid velocity decreases at these region. Velocity vectory occurs in a shape of flattened sphere. So effect of swirling jet can be seen significantly at the inlet region fo swirling jets. At the middle of the channel thermal boundary layer is thickening because of the velocity decrease (decrease of hydrodynamic boundary layer) at the corner of the rectangular channel.

This boundary layer thickness increase can be seen at the direction of swirling flow. Because separation of fluid flow is more evident at these region. At the end of the channel, decrease of fluid velocity can be seen easily. Seperation of the fluid flow is more important at the corner of the channel at this region. Decreasing hydrodynamic boundary layer thickness causes an increase on thermal boundary layer thickness and it causes an increase on surface temperature. So increasing Reynolds number causes an increase on hydrodynamic boundary layer and decrease on themal boundary layer and surface temperature. Variation of average Nusselt number for different Reynolds number is shown in Fig.5.

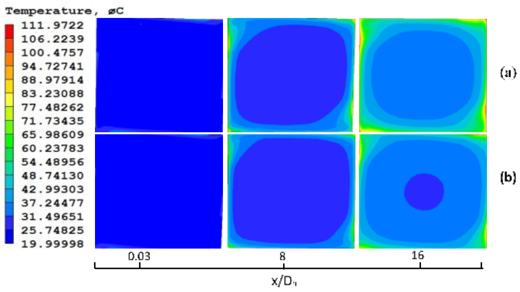

Fig.4. *Temperature contours for different Reynolds Number (a) Re=12000 and (b) Re= 21000*

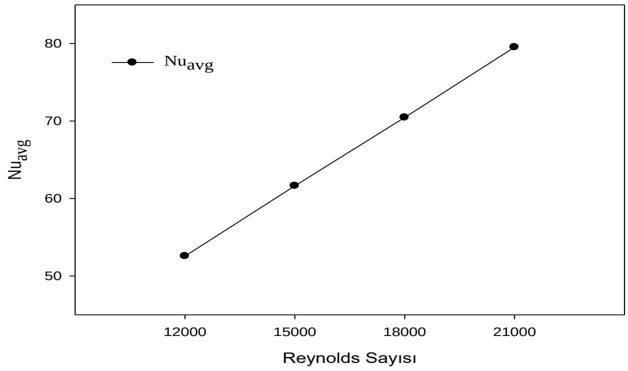

Fig.5. *Average Nusselt Number of Al_2O_3-H_2O nanofluid for different Reynolds Number*

It is obtained that increasing Reynolds number causes an increase on average Nusselt number and decrease on surface temperature. So increasing Reynolds number from *Re*=12000 to 21000 causes an increase of 51.3% on average Nusselt Number. But this increase decreases gradually. Increasing Reynolds number from *Re*=12000 to 15000 causes an increase of 17.2% on average Nusselt number. But this increase decreases to 14.3% and 12.9% for Reynolds number increase from *Re*=15000 to 18000 and *Re*=18000 to 21000.

4.2. Effects of Different Inlet Temperature on Heat Transfer

Fig.6 shows average Nusselt number for different inlet temperature (T_{inlet}= 5, 10, 20, 30°C) of Al_2O_3-H_2O when it is used at 20 nm particle size and φ=4% at *Re*=18000. It can be seen that increasing inlet temperature has not a considerable effect on average Nusselt number. The reason of this is increasing or decreasing the inlet temperature causes a change on heat convection coefficient.

This means that heat flux from the surface to the fluid changes temperature difference between bulk fluid temperature and avarege surface temperature. So surface temperature can be change but it does not cause a significant change on average Nusselt number. Temperature contours for different inlet temperature are shown in Fig.7.

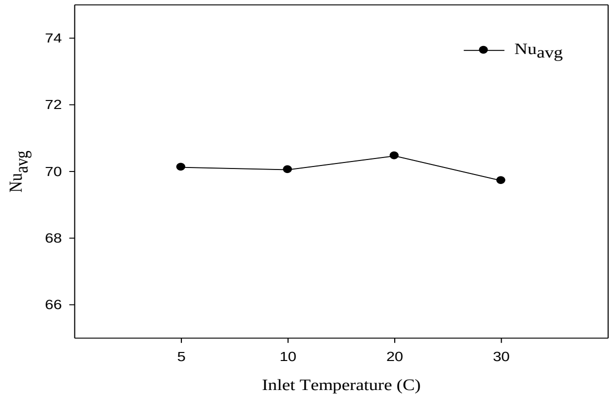

Fig. 6. *Average Nusselt numbers of Al_2O_3-H_2O nanofluid for different inlet temperature*

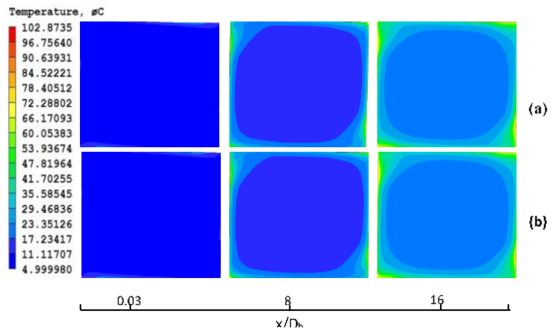

Fig.7. *Temperature contours of Al_2O_3-H_2O nanofluid for different inlet temperature (a) T_{inlet}= 5°C and (b) T_{inlet}= 30°C*

5. CONCLUSIONS

The present study is focused on the numerical investigation of heat transfer from a heated surface by using nanofluids and swirling jets. Effects of different Reynolds number, different inlet temperature on heat transfer and fluid flow were studied numerically. It is obtained that increasing Reynolds number from *Re*=12000 to 21000 causes an increase of 51.3% on average Nusselt Number. Increasing inlet temperature from T_{inlet}=5°C to 30°C has not a significant effect on average Nusselt number. Difference between numerical results of this study and experimental results is less than 11% for *Re*=12000-18000. Research areas for future investigations can be using different particle diameter, different volume ratio and different types of nanofluids with different application geometries and cooling techniques to obtain combined effect of nanofluid and swirling jets.

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