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NUMERICAL INVESTIGATION OF FORCED CONVECTION HEAT TRANSFER PERFORMANCE OF AL_2O_3/WATER NANOFLUIDS IN PLATE HEAT EXCHANGER CHANNEL

Murat Unverdi , Yasar Islamoglu

***Abstract*-The aim of this study is to investigate forced convection heat transfer and flow characteristics in a herringbone type plate heat exchanger channels using Al_2O_3/water nanofluids with numerical analysis method. The numerical analysis comprises the analysis of two dimensional model of the plate heat exchanger channel. Constant heat flux is applied on the walls. The heat transfer characteristics were simulated using finite volume method with the standard k–ε turbulence model and solved iteratively by using the SIMPLE algorithm. The numerical analysis was carried out in different Reynolds (600-1900) numbers and volumetric (0.25%, 0.5%, 0.75%, 1%) concentrations for single phase approach and results got from the numerical analysis were compared with the results obtained experimentally. It was concluded that there is a good consistency between the numerical analysis and results obtained experimentally. According to the experimental study and numerical analysis (this research), heat transfer coefficient increased with increasing the volume fraction of nanofluid and Reynolds number. This study utility when comes to selecting a suitable model for a similar type study.**

***Keywords*-Nanofluids, Plate heat exchanger, Heat transfer, CFD analysis**

I. Introduction

Nanofluids have recently drawn attention in terms of their enhancement of heat transfer of conventional working fluids. Because fluids that are called conventional fluids (water, glycol ethylene, oil, etc.) and are used until today have low heat transfer coefficients, they prevent the dimensions and thermal efficiency of heat exchangers **[1].** As a result of the utilization of solid particles of nano sizes with base fluids to become a suspension, enhancements in heat transfer occurred. Thus, the enhancements in thermo-physical properties of conventional fluids were performed. As a result of this, an increase in the performance of heat exchangers and their thermal performance occurred **[2, 3, 4]**. These enhancements in the heat exchange have been presented by some researchers in their experimental and numerical studies carried out. Bajestan et al. investigated laminar convective heat transfer of water-based TiO_2 nanofluid (%1, %1.5, %1.6, and % 2 and % 2.3) flowing through a uniformly heated tube via experiments and numerical modeling. The numerical section, the single-phase model was compared with the common two-phase numerical approaches. The numerical investigation indicated that the predicted heat transfer coefficients using single-phase and common two-phase approaches, even based on experimental thermophysical properties of nanofluids, underestimate and overestimate the experimental data, respectively. For the single-phase and two phase Eulerian–Eulerian model is the maximum deviation of 11.8% and 12.8%, respectively. Based on the results, a maximum enhancement of 21% in average heat transfer coefficient has been obtained using TiO_2/water nanofluids **[5].** Tiwari et al. investigated numerical of heat transfer and fluid flow in a single pass counter flow chevron corrugated plate heat exchanger considering nanofluids (CeO_2 and Al_2O_3) as homogeneous mixtures. The results of numerical simulation were compared with experimental data in order to verify the accuracy of the homogeneous model. Validation of the CFD model suggests that considering nanofluid a homogeneous mixture, simulation the maximum difference between the experimental and CFD data is 3.75%, which can be qualified as quite satisfactory, simulation can be performed to predict the plate heat exchanger performance with reasonable accuracy **[6].** In this study, a gasket plate heat exchanger with herringbone type was used as a nanofluids plate heat exchanger. Experiments in 4 different volumetric concentrations (0% - 1%) and different mass flow rates (90-300 kg/h) in the channel of the heat exchanger were carried out. Afterwards, the consistency between experimental and numerical studies was shown by making the numerical analysis of nanofluids with numerical analysis technique for the channel geometry.

II. Determining thermo-physical properties of nanofluids

To obtain reliable experimental and numerical results, expressions suggested in the literature for the calculation of thermo-physical properties were used. These properties comprise density, viscosity, specific heat, and heat transfer coefficient equations previously derived. After having assumed that nanoparticles are homogenously dispersed in distilled water, thermo-physical properties of nanofluid were calculated by using average temperature rates. In this study, equations below were used.

$$\rho_{nf} = (1 - \emptyset_{np})\rho_s + \emptyset_{np}\rho_p \quad (1)$$

Pak and Cho, Ho et al. showed the validity of the Equation (1) density of the Al_2O_3-water nanofluid at room temperature. Density of the nano particle, which is 3700 kg/m^3 in the calculations, has been taken from **Ref. [7-12].**

$$\mu_{nf} = (1 + 2.5\emptyset_{np})\mu_s \quad (2)$$

Equation (2), an Einstein equation, can be accepted for less spherical particles that are of 2 % volumetric concentration **[13].**

$$c_{nf} = \frac{(1-\emptyset_{np})\rho_s c_s + \emptyset_{np}\rho_p c_p}{\rho_{nf}} \quad (3)$$

Zhou and Ni showed the validity of the Equation (3) special heat of Al_2O_3-water nanofluids at room temperature with their experiments **[12].** The specific heat of nanoparticle is 880 J/kgK, and it is taken from **Ref. [14]**. Yu and Choi employed the equation below to determine the thermal conductivity of the nanofluid **[15].**

$$k_{nf} = \left[\frac{k_{np} + 2k_s + 2(k_{np} - k_s)(1+\beta_l)^3\emptyset_{np}}{k_{np} + 2k_s - (k_{np} - k_s)(1+\beta_l)^3\emptyset_{np}}\right]k_s \quad (4)$$

In the Equation (4), β_l is layer thickness and is considered to be between 19 % and 22 % of the radius of nanoparticle in general **[16].** In the calculation of thermal conductivity, it's taken as $\beta_l = 0.2$ and thermal conductivity of nanoparticle to be $k_{np} = 35\ W/mK$ from **Ref. [14]**. Also, as for the thermophysical properties of distilled water depending on the temperature, the equations from **Ref. [17]** depending on the properties of water between 0-100 ℃ are obtained as below.

For density,

$$\rho_s = 1001.07 - 0.0885789 \times T$$
$$- 0.00346617 \times T^2 \quad (R^2 = \%99.94) \quad (5)$$

For dynamic viscosity,

$$\mu_s = 0.00175015 - 0.0000517503 \times T$$
$$+8.65854 \times 10^{-7} \times T^2 - 7.53662 \times 10^{-9} \times T^3$$
$$+2.58918 \times 10^{-11} \times T^4 \quad (R^2 = \%99.99) \quad (6)$$

For specific heat,

$$c_{p,s} = 4217.25 - 3.01528 \times T$$
$$+0.0780849 \times T^2 - 8.10854 \times 10^{-4} \times T^3$$
$$+3.32379 \times 10^{-6} \times T^4 \quad (R^2 = \%99.61) \quad (7)$$

For thermal conductivity,

$$k_s = 0.559434 + 0.00215742 \times T$$
$$-0.00000965824 \times T^2 \ (R^2 = \%99.98) \quad (8)$$

Here subscripts, nf, np and s represents mean nanofluid, nanoparticle and distilled water respectively.

III. Analysis of the Data

By the help of the data obtained from experimental and numerical studies, properties of heat transfer and pressure drop are calculated by the means of the statements below. Reynold numbers of the water and nanofluid,

$$Re = GD_h/\mu \quad (9)$$

Here D_h hydraulic diameter, G mass velocity of the channel for water and nanofluid.

$$G = \dot{m}/bL_w \quad (10)$$

The constant heat flux in the plates was calculated resorting to an energy balance to the nanofluids and the total area of the plates:

$$q = \frac{\dot{m}c_p(T_i - T_o)}{2A} \quad (11)$$

Here $\dot{m}$ mass flow rate of fluid, c_p specific heat, T_i and T_o inlet and outlet temperature, A heat transfer area. The friction factor of the nanofluid is calculated with the statement below **[18]**.

$$f = (-\frac{dP}{dx}D_h)/\left(\frac{1}{2}\rho V^2\right) \quad (12)$$

where ρ is density and V is mean velocity. According to Equations (9)-(12), numerical outlet temperature and friction factor are obtained for different volumetric concentrations and Reynolds numbers.

IV. CFD analysis

The flow developing between corrugated surfaces geometric is called to be a periodic fully developed flow. It is accepted that the flow develops at a shorter distance compared to the straight channels, and the flow fully develops after three or five channel pitches approximately in the channels that have a periodic cross-sectional area in plate heat exchangers **[19]**. For this reason, modelling was performed by taking a periodic channel pitch that cold fluid reached to their fully developed flow conditions (hydrodynamic and thermal). The out temperature in the heat exchanger is important for the calculation of the heat transfer coefficient for cold fluid. Because, by using this temperature value, expressions related to the fluids can be calculated. The inlet temperatures for 2D geometric model are the inlet temperatures in the experimental study and they are taken as 290 K for cold fluid. In the model, exit temperatures were numerically calculated by using Standard k-ε Enhanced Wall treatment turbulence model approach for the estimation of the out temperature of fluids. Numerical analysis is under incompressible flow and steady flow conditions and its governing equations are as below. Continuity

$$\nabla(\rho_{nf}\vec{V}) = 0 \quad (13)$$

momentum and energy equations,

$$\nabla(\rho_{nf}\vec{V}\vec{V}) = -\nabla P + \nabla(\mu_{nf}\vec{V}) \quad (14)$$

$$\nabla(\vec{V}(\rho c_p)_{nf}T) = \nabla(k_{nf}\nabla T) \quad (15)$$

Two-equation turbulence models allow the determination of both, a turbulent length and time scale by solving two separate transport equations. The standard k-ε model in

ANSYS Fluent falls within this class of models and has become the workhorse of practical engineering flow calculations in the time since it was proposed by Launder and Spalding. Robustness, economy, and reasonable accuracy for a wide range of turbulent flows explain its popularity in industrial flow and heat transfer simulations. It is a semi-empirical model, and the derivation of the model equations relies on phenomenological considerations and empiricism. The standard model is a model based on model transport equations for the turbulence kinetic energy (k) and its dissipation rate (ε). In the derivation of the k-ε model, the assumption is that the flow is fully turbulent, and the effects of molecular viscosity are negligible. The standard k-ε model is therefore valid only for fully turbulent flows **[20, 21, 22].**

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right]$$

$$+G_k + G_b - \rho\varepsilon - Y_M + S_k \quad (16)$$

$$\frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) = \frac{\partial}{\partial x_j}\left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right]$$

$$+C_{1\varepsilon}\frac{\varepsilon}{k}(G_k + C_{3\varepsilon}G_b) - C_{2\varepsilon}\rho\frac{\varepsilon^2}{k} + S_\varepsilon \quad (17)$$

The turbulent (or eddy) viscosity, μ_t is computed by combining k and ε as follows:

$$\mu_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad (18)$$

where C_μ is a constant.

The model constants,

$$C_{1\varepsilon} = 1.44, C_{2\varepsilon} = 1.92, C_\mu = 0.09, \sigma_k = 1, \sigma_\varepsilon = 1.3$$

Thermophysical properties of the nanofluids are calculated with the equation (2)-(5). During the numerical analysis for the nanofluids, the single phase fluid approach was taken as a basis. Because, due to the fact that particles that are smaller 100 nm (infinitesimally small) comply with this approach, single phase fluid was accepted in the nanofluid models and there is no important difference between the application of classical correlations used for the heat transfer for nanofluids in this approach. **[23, 24].** The liquid phase and nanoparticles the relative velocity between the two phases is negligible. Actually, such an approach cannot take into account the slip motion occurring between suspended nanoparticles and base liquid, whose consequent non-uniform distribution of the solid-phase concentration may have non-negligible effects on heat and momentum transfer. The presence of the nanoparticles in water is only used to determine thermophysical properties of the nanofluids in this approach. Also, while it is a simple approach, accuracy of the results depends on the proper equations used for the calculation of thermo-physical properties of the nanofluids **[25, 26, 27, 28].** The single phase model is simpler and computationally more efficient. However, the error rate in this approach increases in concentrations that are higher than 1% volumetric concentrations **[6].**

V. Physical Model

2D physical model of the plate heat exchanger with corrugated is given in Figure 1. Valid optimum number of mesh (structured) for the model is 13250 elements and is divided into 250 equal parts in the direction of the x-axis, 50 equal parts in the direction of the y-axis. Also, bias factor (bias: 5) from the center to the surface was chosen for the surfaces with heat transfer. In model, firstly, to water heat transfer for cold fluid sides was numerically analysed. The corrugated channels have mean flow channel gap b=2.5 mm and θ=38^o in this investigation.

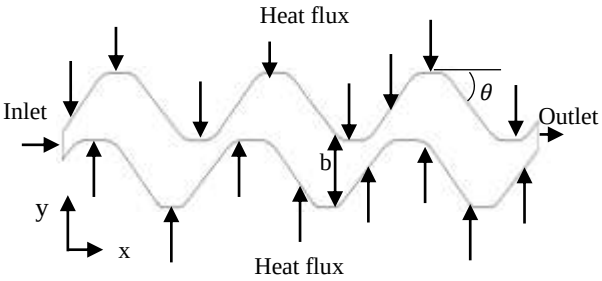

Figure 1. Dimensional physical model

Thermophysical properties of water are obtained with the expressions in the experimental study calculated from the equations (6)-(9). In the result of to water heat transfer, after the validity of model and its consistency with the experimental study have been obtained, numerical analysis for nanofluids in different concentrations same in the experimental study and Reynolds range was carried out.

VI. Results and Discussions

With the experimental setup for the enhancement of heat transfer by using Al_2O_3/water nanofluids, Reynolds number of the heat transfer coefficient and their variations depending on the concentrations prepared were obtained. In this study, k-ε turbulence model was chosen in the numerical analysis for the case that four different volumetric concentrations of the nanofluid (0.25%, 0.5%, 0.75%, and 1%) and Reynolds number varying between 600 and 1900. 2D model was created by making simplifications in the plate heat exchanger experimental studied for this reason. Numerical analysis of nanofluid heat transfer was carried out by showing the validity of the model studied with the numerical analysis done for the water different Reynold numbers. Numeric estimation for heat transfer coefficient, Al_2O_3/water nanofluids with corrugated surface was compared with the results of the heat transfer coefficient obtained experimentally. The out temperature is an important magnitude in the heat exchangers because the heat transfer also depends on the temperature change, the temperature is a measurable parameter in the heat transfer and thermodynamic properties of the fluids can be calculated by using the inlet and out average temperatures of the plate heat exchanger. In Figure 2 constant heat flux out temperatures of the water of the cold fluid side, the flow rate of which was changed, depending on Reynolds number both numerically and experimentally were compared. The results of the exit temperatures between numerical studies and experimental studies showed a similar trend.

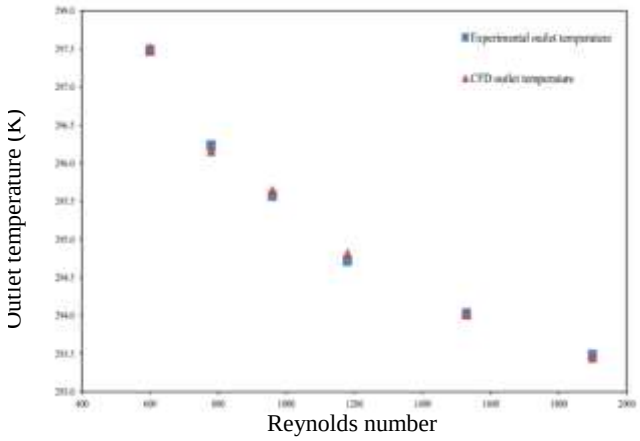

Figure 2. The comparison of the change in the out temperature of the cold fluid depending on the Reynolds number for water

As it is seen in Fig. 2, there is less 0.2 % amount of error on average in the out temperatures of water obtained numerical analysis and experimentally. At the end of the heat transfer to nanofluids, there is approximately 0.18 % error between the out temperatures obtained with numerical analysis and out temperatures obtained experimentally for nanofluids. The results similar to the experimental study have been obtained in the numerical study too. The change in the heat transfer coefficient obtained through numerical analysis and experimental results in according with the concentrations prepared depending on to the Reynolds number, gives in Fig. 3. With numerical analysis, an increase in the heat transfer coefficient of approximately 23% were obtained. For example, the increase rate in the heat transfer coefficient of the nanofluids ranges between 1.52 and 2.07 while the Reynolds number varies between 600 and 1900 at 1% volumetric concentration. Also, an excellent consistency all of volumetric concentration between heat transfer coefficients experimentally obtained with numerical analysis was obtained.

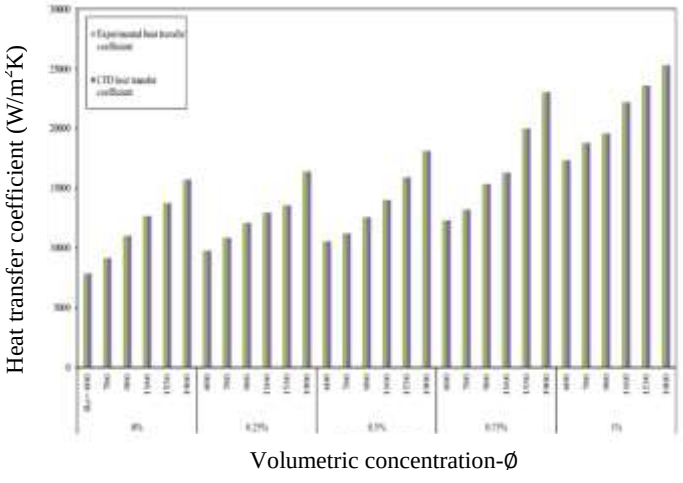

Figure 3. The comparison of the numerical heat transfer coefficient according to volumetric concentration with the experimental heat transfer coefficient depending on the Reynolds number

Very high heat transfer coefficients are achieved due to the breakup and reattachment of boundary layers, swirl or vortex flow generation. The turbulence is one of the main flow properties on heat transfer enhancement. Finally, high thermal performance can be achieved in plate heat exchanger channels using Al_2O_3/water nanofluids.

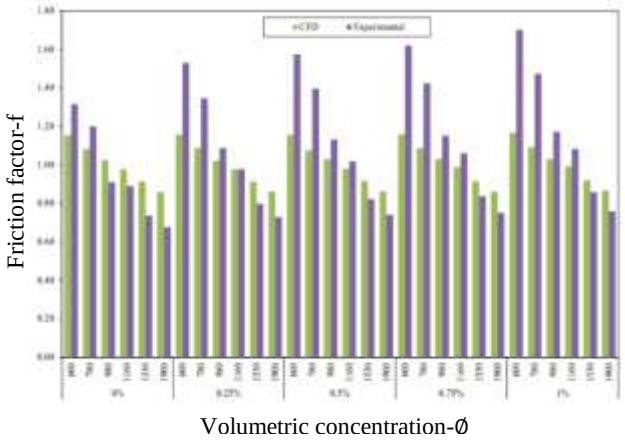

Figure 4. The experimental and numerical comparison of friction factor according to volumetric concentration

The change depending on the volumetric concentration changing in accordance with Reynolds number in the friction factor obtained with experimental and numerical analysis can be seen in Fig. 4. The numerically obtained friction factor showed a similar trend with the experimental results. The corresponding pressure-drop penalty, however is slightly higher depending on the Reynolds number, from 0.8 to 1.46 times higher friction factors between experimental and numerical analysis. In response to the increasing heat transfer coefficient, the difference in the pressure drop obtained by the comparison of the numerical analysis and experimental study is at a negligible level. Among the experimental and numerical studies in Fig. 3, error results obtained for the heat transfer coefficient are satisfactory in terms of the calculation of the heat transfer coefficient. It is concluded that 2D model set up for the numerical calculation of the heat transfer coefficient of the nanofluids and single phase approach are suitable for the plate heat exchanger with nanofluids. On the account of this, employment of the plate heat exchangers that are frequently used in industry for different purposes and as for the fluids to carry out the tasks expected from them can be reasonable.

VII. Conclusion

In this study, numerical analysis of the cases in the channel of counter flow the plate heat exchanger with nanofluid, with different volumetric concentrations (0.25%, 0.5%, 0.75%, 0.1%) and the cases that Reynolds number varies from 600 to 1900 by using 2D model was carried out. While the boundary conditions necessary for the numerical analysis are the same with those in the experimental study, the results obtained with the numerical analysis were compared with the results in the experimental study. The single phase fluid approach was taken as a basis for the thermophysical properties of the nanofluids in the numerical analysis of the flow in the channel. The expressions used for the calculation of the chosen concentration and thermophysical properties of the nanofluid were observed to be proper for this approach and it was obtained that the numerical and experimental results showed a good match for all of volumetric concentration. Another result is that heat transfer

coefficient increased in accordance with the increasing Reynolds number and volumetric concentration as in the experimental study. As a result of the numerical studies done, it is concluded that heat transfer coefficient obtained for different Reynolds number and different volumetric concentrations is in a good consistency with the heat transfer coefficient in the experimental study, however, in term of pressure drop is slightly penalty. The single-phase approach is easy to implement and requires less computational time; the predicted results concerning convective heat transfer characteristics of nanofluids agree well with experiments. But the results depend strongly on the selected thermophysical property models, especially those for thermal conductivity and viscosity. Despite the nanofluid is actually a two-phase fluid in nature, the results show that the nanofluid behave more like a different fluid than a liquid-solid mixture.

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