NUMERICAL AND EXPERIMENTAL INVESTIGATION OF HEAT TRANSFER OF ZnO/WATER NANOFLUID IN THE CONCENTRIC TUBE AND PLATE HEAT EXCHANGERS

by

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The plate and concentric tube heat exchangers are tested by using the water--water and nanofluid-water streams. The ZnO/water (0.5 v/v%) nanofluid has been used as the hot stream. The heat transfer rate omitted of hot stream and overall heat transfer coefficients in both heat exchangers are measured as a function of hot and cold streams mass flow rates. The experimental results show that the heat transfer rate and heat transfer coefficients of the nanofluid in both of the heat exchangers is higher than that of the base liquid (i. e. water) and the efficiency of plate heat exchange is higher than concentric tube heat exchanger. In the plate heat exchanger the heat transfer coefficient of nanofluid at $\dot{m}_{cold} = \dot{m}_{hot} = 10 g/s$ *is about 20% higher than the base fluid and under the same conditions in the concentric heat exchanger is 14% higher than the base fluid. The heat transfer rate and heat transfer coefficients increases with increase in mass flow rates of hot and cold streams. Also the computational fluid dynamics code is used to simulate the performance of the mentioned heat exchangers. The results are compared to the experimental data and showed good agreement. It is shown that the computational fluid dynamics is a reliable tool for investigation of heat transfer of nanofluids in the various heat exchangers.*

Key words: *heat transfer, plate heat exchanger, concentric tube heat exchanger, nanofluid, computational fluid dynamics*

Introduction

The various types of heat exchangers such as double pipe and plate heat exchangers (PHE) are widely used in food and chemical processing industries. The plate or corrugated plate heat exchangers are replacing conventional concentric or double pipe heat exchangers. A plate heat exchanger is a type of heat exchanger that uses metal plates to transfer heat between two fluids. This has a major advantage over a conventional heat exchanger in that

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the fluids are exposed to a much larger surface area because the fluids spread out over the plates [1, 2].

A plate heat exchanger consists of a series of thin and corrugated plates. These plates are gasketed, welded or brazed together depending on the application of the heat exchanger. The plates are compressed together in a rigid frame to form an arrangement of parallel flow channels with alternating hot and cold fluids.

To enhance the thermal efficiency of the heat exchangers, the thermal capability of the working fluid must be increase [3, 4].

Addition of small amount of high thermal conductivity solid nanoparticles in base fluid increases the thermal conductivity, thus increasing the heat transfer rate in the heat exchangers [5-8].

Nanofluid is the name conceived by Argonne National Laboratory (ANL), USA, to describe a fluid in which nanometer-sized particles are suspended. Nanofluids are a class of heat transfer fluids created by dispersing solid nanoparticles in traditional heat transfer fluids. Research results show that nanofluids have thermal properties that are very different from those of conventional heat transfer fluids such as water or ethylene glycol.

The ANL performed nanofluid experiment, where it was found a 20% increase in the thermal conductivity. Theoretically the thermal conductivity increases are based on the volume fraction and shape of the particles. The increase of the thermal conductivity leads to an increase in heat transfer performance. In fact the reduction of the thermal boundary layer thickness due to the presence of the nanoparticles and the random motion within the base fluid may have important contributions to such heat transfer improvement as well [9-11].

Nanofluids were first used by Choi [12]. They showed that addition of small amount of nanoparticles to the base fluids increased the thermal conductivity of the fluids up to approximately two times.

Many experimental works have been attempted in the nanofluids area. Some of these works are focused on use of nanofluids in the circular pipes or in the heat exchangers.

Yang *et al.* [13] reported experimental results which illustrated the convective heat transfer coefficient of graphite nanoparticles dispersed in liquid for laminar flow in a horizontal tube heat exchanger. Effects of the Reynolds number, volume fraction, temperature, nanoparticle source, and type of base fluid on the convective heat transfer coefficient have been investigated.

The experimental results showed that the heat transfer coefficient increased with the Reynolds number and the particle concentration. For example at 2.5 wt.% the heat transfer coefficient of the nanofluids was 22% higher than the base fluid at 50 °C fluid temperature and 15% higher at 70 °C.

Wen *et al.* [14] presented an experimental study which evaluated the convective heat transfer coefficient of Al_2O_3 nanoparticles suspended in deionized water for laminar flow in a copper tube, focusing in particular on the entrance region. The various concentrations of nanoparticles (0.6, 1.0, and 1.6%) were tested under a constant wall heat flux. The results showed that the use of nanofluids affected a pronounced increase in the heat transfer coefficient causing a decrease in the thermal boundary layer thickness and will decrease along the tube length.

Recently, Pantzali *et al.*, reported an experimental study that showed the role of nanofluids as coolant in the plate heat exchangers [15]. CuO/water nanofluid has been selected as a coolant in the plate heat exchanger. The experimental results confirmed that besides the physical properties, the type of flow inside the heat exchanging equipment also affects the efficacy of a nanofluid as coolant.

Experimental studies of pressure drop and convective heat transfer of TiO_2/water nanofluid in a double pipe heat exchanger are reported by Duangthongsuk *et al.* [16]. In this study effect of mass flow rate of hot and cold fluids, and nanofluid temperature on heat transfer coefficient have been studied. The results show that the convective heat transfer coefficient of nanofluid is slightly higher than that of the base liquid by about 6-11%.

Heat transfer in a mini heat exchanger using CuO/water nanofluids has been investigated by Mapa *et al.* [17]. In this study, theoretical heat transfer rates were calculated using existing relationships in the literature for conventional fluids and nanofluids. Also heat transfer enhancement determined compared to fluids without nanoparticles. At mass flow rates of 0.005 kg/s, a 5.5% increase in heat transfer rate of 0.01% CuO/water nanofluid is observed.

There are a few publications dealing with numerical studies of heat transfer in double-pipe or plate heat exchangers especially when a nanofluid has been used as a working fluid.

A double-pipe helical heat exchanger was numerically modeled for laminar fluid flow and heat transfer by Rennie *et al.* [18]. A computational fluid dynamics (CFD) package (PHOENICS 3.3) was used to numerically study the heat transfer. The results showed an increasing overall heat transfer coefficients as the inner Dean number is increased; however, flow conditions in the annulus had a stronger influence on the overall heat transfer coefficient.

Kanaris *et al*., reported a CFD model for investigation of flow and heat transfer in a corrugated plate heat exchanger [19]. In this study, the water has been used as a working fluid. Also CFD modeling has been used by Galeazzo *et al.* [20], Grijspeerdt *et al.* [21], and Fernandes *et al.* [22] for prediction of hydrodynamic and heat transfer performance in the plate heat exchangers. In these works also nanofluid has not been used.

Effect of nanofluids on the performance of a miniature plate heat exchanger, has been studied by Pantzali *et al.* [23]. In their work, the effect of the use of a nanofluid in a miniature plate heat exchanger has been investigated both experimentally and numerically. CuO/water 4% nanofluid has been used as a working fluid in the heat exchanger. In their work the effect of the nanofluid on the heat exchanger performance was studied using CFD simulation and the CFD predictions compared to the experimental data.

In the present work both experimental and computational works have been done for investigation of convective heat transfer of ZnO/water nanofluid in the double pipe and plate heat exchangers. Effects of hot and cold stream velocities on heat transfer rate and heat transfer coefficients have been investigated. The performance of two types of the heat exchangers are compared together and the CFD predictions are compared to the experimental results.

CFD simulation

The physical aspects of any fluid flow are governed by three fundamental principles: mass is conserved; Newton's second law and energy is conserved. These fundamental principles can be expressed in terms of mathematical equations, which in their most general form are usually partial differential equations. CFD is the science of determining a numerical solution to the governing equations of fluid flow whilst advancing the solution through space or time to obtain a numerical description of the complete flow field of interest [24, 25]. A commercial CFD package, CFX version 11 is used for the simulations.

In this work, incompressible flow without mass transfer and chemical reaction is assumed. Hence only the mass, momentum and energy equations are considered:

– continuity equation

$$\frac{\partial \rho}{\partial t} + \nabla(\rho u) = 0 \tag{1}$$

– momentum equation

$$\frac{\partial}{\partial t}(\rho u) + \nabla(\rho u u) = -\nabla \sigma + B \tag{2}$$

For the laminar Newtonian fluid flow, stress tensor can be expressed in the term of fluid properties as:

$$\sigma = -P\delta + \mu(\nabla u + (\nabla u)^T] + \left(k - \frac{2}{3}\mu\right)\nabla u \delta \tag{3}$$

– energy equation

$$\frac{\partial}{\partial t}(\rho h) + \nabla(\rho u C_\mathrm{P} T) = \nabla(k \nabla T) \tag{4}$$

In order to solve above-mentioned equations the thermophysical parameters of nanofluids such as density, viscosity, heat capacity, and thermal conductivity must be evaluated, and then these parameters must feed to CFX, for enabling the software to perform calculations of temperature distribution. These parameters are defined as:

– viscosity and density

The viscosity and density of the nanofluids can be calculated using the Drew and Passman relation [26]:

$$\mu_\mathrm{nf} = (1 + 2.5\varphi)\mu_\mathrm{f} \tag{5}$$

$$\rho_\mathrm{nf} = \varphi \rho_\mathrm{s} + (1 - \varphi)\rho_\mathrm{f} \tag{6}$$

– heat capacity

The heat capacity equation presented as:

$$C_{\mathrm{P}_\mathrm{nf}} = \frac{\varphi \rho_\mathrm{s} C_{\mathrm{P}_\mathrm{S}} + (1 - \varphi)\rho_\mathrm{f} C_{\mathrm{P}_\mathrm{f}}}{\rho_\mathrm{nf}} \tag{7}$$

– thermal conductivity

Yu and Choi correlation can be used for calculation of nanofluid effective thermal conductivity as [27]:

$$k_\mathrm{nf} = \left[\frac{k_\mathrm{s} + 2k_\mathrm{f} + 2(k_\mathrm{s} - k_\mathrm{f})(1 + \beta)^3 \varphi}{k_\mathrm{s} + 2k_\mathrm{f} - (k_\mathrm{s} - k_\mathrm{w})(1 + \beta)^3 \varphi}\right] k_\mathrm{f} \tag{8}$$

In this equation β is the ratio of the nanolayer thickness to the original particle radius and β = 0.1 was used [27].

Steady-state simulations of fluid flow in two types of heat exchangers have been carried out. The 3-D geometrical configurations of the problem under consideration for CFD simulation are shown in fig. 1.

The dimensions of these geometries are the same to the dimensions of experimental set-up. The geometries of the computational domains and the mesh preparation were generated with Gambit 2.2.30. Several different grid sizes have been used to solve the governing equations to check for grid incipiency. 700,000 and 1,450,000 unstructured tetrahedral grids were found to be suitable for calculations in the concentric tube and plate heat exchanger, respectively. The nodes were concentrated in the entrance region and near the walls.

The heat exchangers must have boundary conditions at the system boundaries for the mode1 to be closed. In the mode1 of the heat exchanger the boundary conditions must be specified at the side walls, the inlet sections and at the outlet sections (for both hot and cold fluids).

The side walls have a no slip condition for the hot and cold fluids, that means that the velocity of the fluid at the wall boundary is set to zero.

Figure 1. Computational domains (plate and double pipe heat exchangers)

At the inlet sections, "velocity inlet" boundary condition was used. The magnitude of the inlet velocity is specified and the direction is taken to be normal to the boundary. At this boundary, the appropriate values for the velocity components and inlet temperature for hot and cold fluids must be specified.

At the outlet section, "outlet flow" boundary condition was used. At the outlet section flow field is assumed fully developed and relative static pressure is specified over the outlet boundary. The flow regime is under laminar conditions.

In this study, the single phase approach applied to the nanofluid. Because the solid particles are very small and they are easily fluidized and can be approximately considered to behave as a fluid. In this model the fluid phase and the nanoparticles are in thermal equilibrium with zero relative velocity [28-30].

Experimental set-up

The experimental setup is shown in figure 2.

The calculations of heat transfer coefficients were conducted in a P. A. Hilton laboratory heat exchangers unit. The unit consists of a concentric tube, shell and tube, and plate heat exchangers. The concentric tube and plate heat exchangers have been used in this study. The main characteristics parameters of concentric tube and plate heat exchangers are shown in tab. 1. The test fluids were distilled water and ZnO/water nanofluid.

Figure 2. Experimental set-up

Table 1. Characteristics parameter of concentric tube and plate heat exchangers

Plate heat exchanger		Concentric tube heat exchanger	
Plate material	316 stainless steel	Inner tube material	316 stainless steel
Plate overall dimensions	0.072 m × 0.189 m	Outlet diameter (inner tube)	0.012 m
Total heat transfer area	0.024 m^2	Wall thickness	0.001 m
Number of plates	4	Outer tube material	Clear acrylic
Number of channels – Hot side	2	Inlet diameter (outer tube)	0.022 m
Number of channels – Cold side	1	Active heat transfer length	2 × 0.318 m

The heat removed from the hot fluid, Q_h, and the heat absorbed by the cooling liquid, Q_c, are calculated by eqs. (9) and (10). It is confirmed that they are practically equal:

$$Q = Q_c = m_c C_{pc}(\Delta T_c) \tag{9}$$

$$Q = Q_h = m_h C_{ph}(\Delta T_h) \tag{10}$$

The total heat transfer coefficient is defined as:

$$U = \frac{Q}{A \cdot LMTD} \tag{11}$$

$$\mathrm{LMTD} = \frac{T_{ho} - T_{ci} - T_{hi} - T_{co}}{\ln \dfrac{T_{ho} - T_{ci}}{T_{hi} - T_{ci}}} \tag{12}$$

During each experiment, the flow rates of the hot and cold fluids were set and after steady-state was achieved, the temperatures were recorded. Temperature data are acquired using high accuracy T-type thermocouples, located at the inlet and outlet of the hot and cold streams, respectively.

Then, the mass flow rate of hot fluid was increased and new data were obtained. The aforementioned procedure was also repeated for various cold fluid flow rates. Finally the overall heat transfer coefficients and heat transfer rates can be calculated from eqs. (11) and (12).

Results and discussions

Concentric tube heat exchanger

Concentric tube heat exchangers are the simplest type of heat exchanger. Essentially it is a tube inside of a tube. In one part a hot fluid flows and transfers its heat through the wall of a metal (stainless steel) to another cold fluid. The counter flow configuration has been used through all of the experimental works.

Nanofluid flows inside the inner tube as a hot stream while distilled water entered annular section as a cold stream. Effect of hot and cold fluids flow rates on the heat transfer rate and heat transfer coefficients of nanofluid has been studied.

Figures 3 and 4 show the heat transfer rate and total heat transfer coefficients *vs.* hot fluid mass flow rate for various cold fluid flow rates. Also the comparison between the heat transfer of nanofluid and base fluid is shown in these figures. It can be seen from fig. 3 that the heat transfer rates increases with increasing in hot and cold liquid flow rates. Also for a given hot or cold flow rate, the heat transfer rate and total heat transfer coefficient of the nanofluid is higher than the distilled water. Increasing the hot fluid flow rate leads to an increase in the heat transfer rate and results in an increase in the heat transfer coefficient.

Also as shown in fig. 5 for a given hot fluid flow rate, the increase in the overall heat transfer coefficient is more forcible at high cold flow rate. For example in the $\dot{m}_\mathrm{c} = 10$ g/s and $\dot{m}_\mathrm{h} = 40$ g/s the increase in U is 7.5% but in the $\dot{m}_\mathrm{c} = 40$ g/s and $\dot{m}_\mathrm{h} = 40$ g/s, the increase is 14.1%. This observation agrees with the results reported by Duangthongsuk, *et al.* [16].

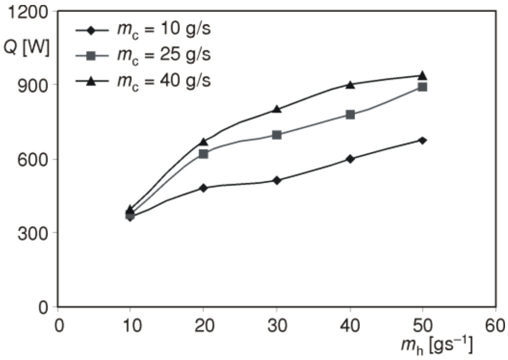

Figure 3. Effect of hot and cold flow rate on heat transfer rate of nanofluids in concentric tube heat exchanger

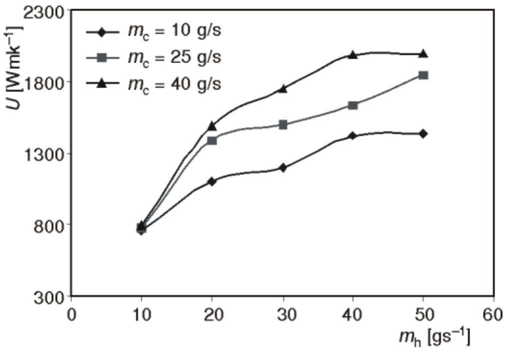

Figure 4. Effect of hot and cold flow rate on overall heat transfer coefficient of nanofluids in concentric tube heat exchanger

Plate heat exchanger

The plate heat exchangers are composed of multiple, thin, slightly-separated plates that have very large surface areas and fluid flow passages for heat transfer. This stacked-plate arrangement can be more effective, in a given space, than the concentric tube heat exchanger.

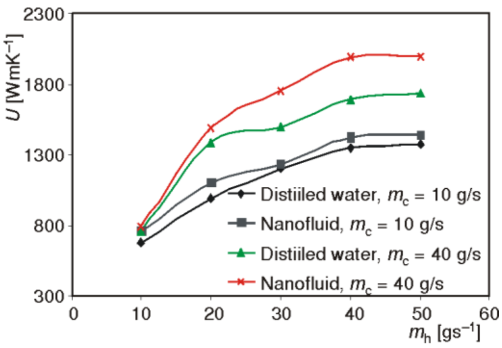

Figure 5. Comparison between heat transfer coefficient of distilled water and nanofluid in the concentric tube heat exchanger

The plates are often spaced by rubber sealing gaskets which are cemented into a section around the edge of the plates. The plates are pressed to form troughs at right angles to the direction of flow of the liquid which runs through the channels in the heat exchanger. These troughs are arranged so that they interlink with the other plates which forms the channel with gaps of 1.3-1.5 mm between the plates. Making each chamber thin ensures that the majority of the volume of the liquid contacts the plate, again aiding exchange. The troughs also create and maintain a turbulent flow in the liquid to maximize heat transfer in the exchanger. A high degree of turbulence can be obtained at low flow rates and high heat transfer coefficient can then be achieved [31, 32]. Figure 6 shows the structure of a typical plate heat exchanger.

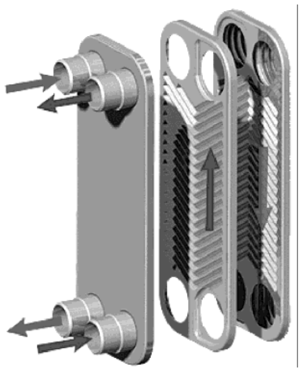

Figure 6. Structure of a typical plate heat exchanger

Figures 7 and 8 show the heat transfer rate and overall heat transfer coefficients *vs.* hot fluid mass flow rate for various cold fluid flow rates. It can be clearly seen that the heat transfer coefficient and heat transfer rate increases with increasing both hot and cold mass flow rates.

Figure 9 shows overall heat transfer coefficient of nanofluid and distilled water *vs.* hot fluid mass flow rate. From this figure it is clear that for a constant mass flow rate, heat transfer coefficients of nanofluid are much higher than distilled water. For example at the $m_h = m_c = 10$ g/s, the heat transfer coefficient of nanofluid is about 20% higher than distilled water.

The heat transfer area of the heat exchangers are almost same, so the heat transfer rate or heat transfer coefficient in

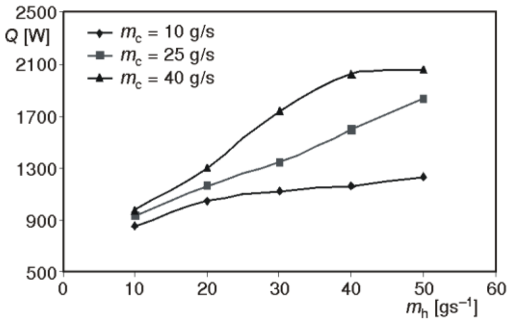

Figure 7. Effect of hot and cold flow rate on heat transfer rate of nanofluids in plate heat exchanger

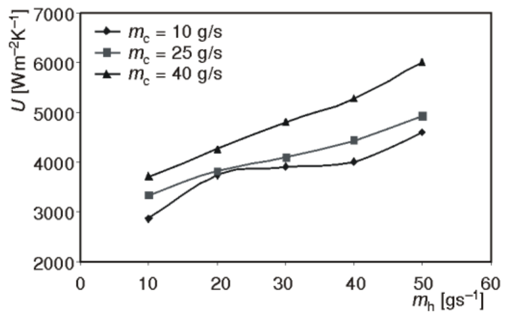

Figure 8. Effect of hot and cold flow rate on overall heat transfer coefficient of nanofluids in plate heat exchanger

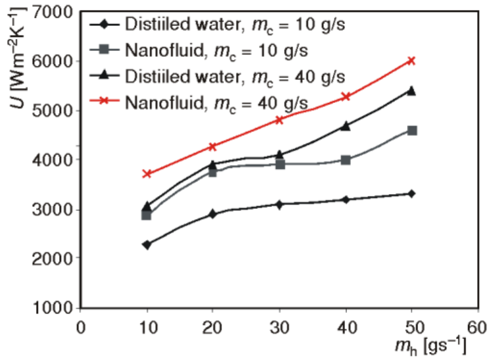

Figure 9. Comparison between heat transfer coefficient of distilled water and nanofluid in the plate heat exchanger

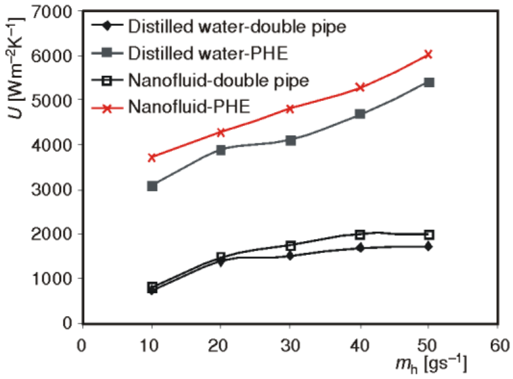

Figure 10. Comparison between heat transfer coefficients in the plate heat exchanger and concentric tube heat exchanger

the heat exchangers can be compare together. Figure 10 shows the comparison between heat transfer coefficients in the concentric tube and plate heat exchangers. It can be seen from this figure that the heat transfer coefficients for both distilled water and nanofluid in the plate heat exchanger is higher than double pipe heat exchanger. Because however the heat transfer areas are the same in the heat exchangers, but in the plate heat exchanger, the fluids spread out over the plates. This facilitates the transfer of heat, and greatly increases the speed of the temperature change.

CFD results

In order to verify the reliability and exactness of the model, the predicted heat transfer rate and overall heat transfer coefficients have been compared to the experimental data. The results presented here consist of 3-D simulations of the flow fields and heat transfer in the 3-D configuration

A typical sample of temperature distribution of ZnO/water (0.5%volume fraction) nanofluid through the concentric tube and plate heat exchangers are shown in figs. 11 and 12.

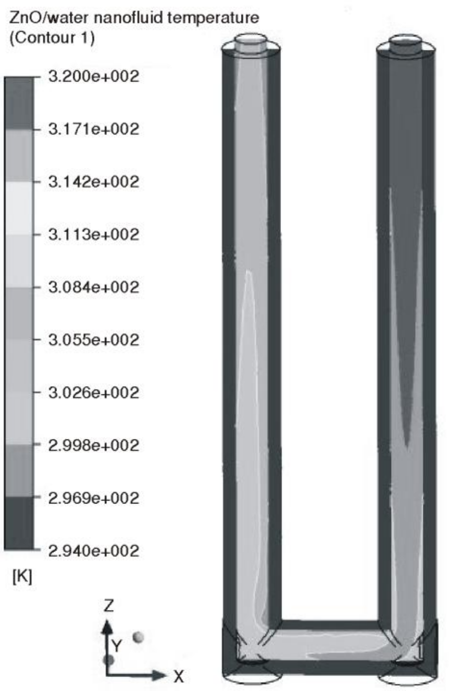

Figure 11. Temperature contours of nanofluids in a plane at the center of concentric tube heat exchanger

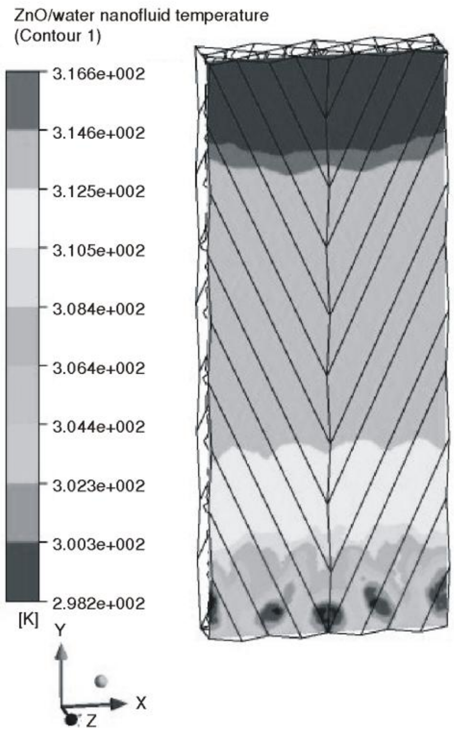

Figure 12. Temperature contours of nanofluids through the plate heat exchanger

The color map, from red to blue, ranges from 294 K and 320 K and 298 K to 316 K for the nanofluid temperature in the concentric tube and plate heat exchangers, respectively. The average temperature of ZnO/water nanofluid at the inlet and outlet of the heat exchangers are well-known. Now the heat transfer rate or heat transfer coefficient can be calculated using equations (9-11).

Figures 13 and 14 show the comparison between heat transfer rate predicted by CFD simulation and experimental data in the concentric tube and plate heat exchangers.

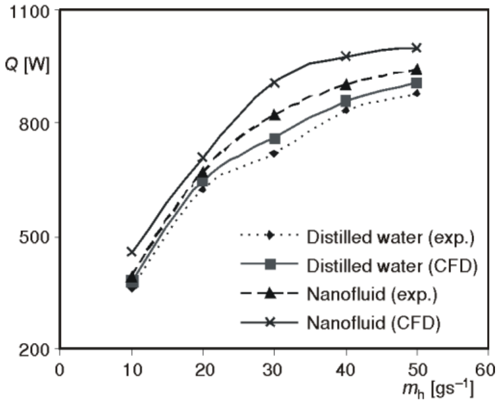

Figure 13. Comparison between CFD predictions and experimental data in concentric tube heat exchanger at m_c = 40 g/s

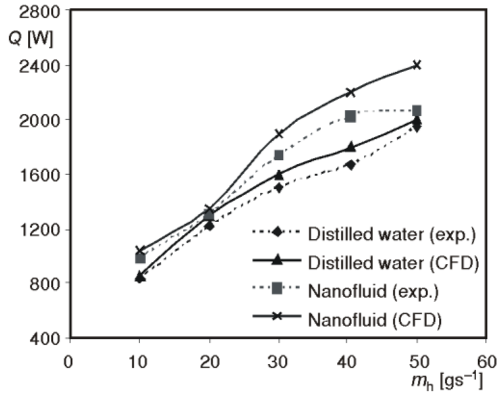

Figure 14. Comparison between CFD predictions and experimental data in plate heat exchanger at m_c = 40 g/s

It is clear that the heat transfer rate of nanofluid in both type of heat exchangers are higher than base fluid (distilled water). There is a good agreement between the experimental data and CFD results. The average relative error between the CFD predictions and experimental data in the concentric tube and plate heat exchangers are 8% and 7.5%, respectively.

As shown in these figures, the heat transfer rates predicted from the CFD simulation are higher than experimental data. Because in the CFD simulation some parameters that limit the heat transfer, are neglected. For example in the CFD simulation it is assumed that liquid distribution is uniform and all surface of the heat exchanger have been used by liquid, also the back-mixing and flow separation phenomena (especially in plate heat exchanger) is neglected, therefore the heat transfer rate in the CFD models is higher than experimental data.

Conclusions

In this study the CFD simulations have been developed to predict the overall heat transfer coefficient and heat transfer rate of ZnO/water nanofluid in a concentric tube and plate heat exchangers.

The volume averaged continuity, momentum, and energy equations were numerically solved using CFX version 11. Single-phase model has been used for prediction of temperature and fluid flow distribution and calculation of heat transfer coefficients and heat transfer rates. The effects of some important parameters such as hot an cold mass flow rate on heat transfer parameters have been investigated under laminar conditions. Overall heat transfer coefficient of nanofluid increases with increase in the hot and cold mass flow rates.

For a given hot fluid flow rate, the increase in the overall heat transfer coefficient is more forcible at high cold flow rate. Also for a constant mass flow rate, heat transfer coefficients of nanofluid are much higher than base fluid (distilled water). For example in the plate heat exchanger, at the $m_h = m_c$ = 10 g/s, the heat transfer coefficient of nanofluid is about 20% higher than distilled water.

The heat transfer area of the heat exchangers are almost same, so the heat transfer rate or heat transfer coefficient in the heat exchangers can be compare together.

It can be seen from the experimental results that the heat transfer coefficients for both distilled water and nanofluid in the plate heat exchanger is higher than double-pipe heat exchanger.

Computed heat transfer coefficients and heat transfer rate of nanofluid are in good agreement with the experimental data. The average relative error between the CFD predictions and experimental data in the concentric tube and plate heat exchangers are 8% and 7.5%, respectively.

Nomenclature

A – surface area, [m^2]
B – body force, [Nm^{-3}]
C_P – specific heat capacity, [$kJkg^{-1}K^{-1}$]
h – enthalpy, [$kJkg^{-1}$]
k – thermal conductivity, [$Wm^{-1}K^{-1}$]
m – mass flow rate, [kgs^{-1}]
P – pressure, [Pa]
Q – heat transfer rate, [W]
T – temperature, [°C]
t – time, [s]
U – overall heat transfer coefficient, [$Wm^{-2}K^{-1}$]
u – interstitial velocity vector, [ms^{-1}]

Greek letters

φ – volume fraction, [–]
μ – viscosity, [$kg^{-1}s^{-1}$]
ρ – density, [kgm^{-3}]
σ – surface tension, [Nm^{-1}]

Subscripts

c – cold
f – fluid
h – hot
i – inlet
nf – nanofluid
o – outlet
s – solid particle

Acronyms

ANL – Argonne National Laboratory, DuPage County, Ill., USA
CFD – computational fluid dynamics
LMTD – logarithmic mean temperature difference, [K]
PHE – plate heat exchanger

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