Ciência eNatura, Santa Maria, v. 37 Part 1 2015, p. 199−206

ISSN impressa: 0100-8307 ISSN on-line: 2179-460X

Effect of nano particles on heat transfer in heat exchangers

Ali Mahrooghi[1],Mohammad Moghiman[2,*]

[1] Department of Mechanical Engineering , KhorasanRazavi Science and Reaserch Branch, Islamic Azad University, Neyshabur, Iran

[2] Department of Mechanical Engineering, Ferdowsi University of Mashhad, Mashhad, Iran [2] Department of Mechanical Engineering, Neyshabur Branch, Islamic Azad University, Neyshabur, Iran

[*]Corresponding author: (Professor Moghiman)+98 5138805091; fax: +98 5138763304

Email: moghiman@um.ac.ir

Abstract

In this paper, forced convection flow and heat transfer of a Al2O3/Water nanofluid have been investigated numerically by single and two phase (volume of fluid) models. Nanofluid flows inside the inner tube of the isothermally concentric circular and sinusoidal double tube heat exchangers while hot pure water flows in the outer tube. The single-phase and two-phase models is used to simulate the nanofluid forced convection of 2% and 3% volume concentrations. The renormalization group k-ε model is used to simulate turbulence in ANSYS FLUENT 15.0. Results show that the overall heat transfer coefficient increases with nanoparticle volume concentrations in the heat exchangers. The highest overall heat transfer coefficient rates are detected, for each concentration and shape, corresponding to the highest flow rate for the sinusoidal tube heat exchanger . The maximum overall heat transfer coefficient enhancement is ⊔ 220% for the particle volume concentration of 3% at the inner tube of concentric sinusoidal double tube heat exchanger corresponding to flow rate =10 LPM. The results reveal that the Al2O3/water pressure drop along the inner tube of circular and sinusoidal double tube heat exchanger increases by about 3% and 5% for volume concentrations of 2% and 3%, respectively, given flow rate compared to the base fluid.Comparison of these results with Rohit S. Khedkar's published experimental data, showed good agreement.

Keywords: *nanofluid,heat exchanger,heat transfer,two phases.*

Recebido: dia/mês/ano Aceito: dia/mês/ano

1 Introuction

Heat exchangers are widely used in many engineering applications, for example, applications in automotive industries ,chemical industry, environment engineering, waste heat recovery, air conditioning, refrigeration, power production and food industry. Application of additives to liquids is oneway of enhancing heat transfer. Augmenting of fluid thermal conductivity is the main purpose in improvement of the heat transfer characteristic of liquids. Their thermal conductivity plays an important role in the heat transfer between the working fluid and the heated surface. An innovative way to improve the thermal conductivity of a fluid is to suspend nano-sized particles with high thermal conductivity in the base fluid with low thermal conductivity. In recent years, extensive research has proven that nanofluids are superior as a heat transfer agent over conventional fluids.

Masuda et al. (1993) preseted the liquid suspension of nano-sized particles and then Choi (1995) for the first time proposed the name of nanofluid.

Pak and Cho(1998) [2] reported experimentally the turbulent forced convection heat transfer of Al_2O_3 /water and TiO_2 /water nanofluids inside a circular tube. It was presented that the heat transfer enhancement of Al_2O_3 particles is higher than TiO_2 particles.

Li and Xuan(2002) [3] concluded that in laminar and turbulent flow regime in forced convection, the heat transfer coefficient of Cu-water nanofluids flowing inside a uniformly heated tube remarkably increased compared to that of pure water. Moreover, it was discovered that the increase of nanoparticle concentration would also enhance the heat transfer coefficient.

Jang, et al(2007) [4] modeled the Brownian-motion induced nano convection as a significant agent nanoscale mechanism governing the thermal behavior of nanofluids. In their model effects of variety of parameters such as the ratio of the thermal conductivity of nanoparticles to that of a base fluid, volume fraction, nanoparticle size, and temperature on the effective thermal conductivity of nanofluids was included.

Chun, B. H. and et. al.(2008) [5] have investigated the convective heat transfer coefficient of nanofluids made of alumina nanoparticles and transformer oil which flow through a double pipe heat exchanger system in the laminar flow regime. They have presented an experimental correlation for an alumina-transformer oil nanofluid system to understand the enhancement of heat transfer of nanofluid.

Lotfi et al. [6] have compared the single-phase with the Mixture and Eulerian two-phase models for the forced convection flow of Al_2O_3/Water nanofluid with temperature independent properties. Also, they have compared the Nusselt number predictions for a 1% value concentration of nanoparticles with several correlations and one set of experimental values. They have also studied the effect of volume concentration on the wall temperature. Their results showed that the Mixture model is more precise than the other two models.

Bahiraei, M. and Hangi, M.(2013) [7] have studied the performance of water based Mn–Zn ferrite magnetic nanofluid in a counter-flow double-pipe heat exchanger under quadrupole magnetic field using the two-phase Euler–Lagrange method. They have examined the effects of different parameters including concentration, size of the particles, magnitude of the magnetic field and Reynolds number.

So, this paper aims to study, two-phase model of Volume of Fluid (VOF) has been used for considering flow and heat transfer characteristic of nanofluids. Al_2O_3 /water nanofluids are used in turbulent flow regime in the isothermally concentric circular and sinusoidal double tube heat exchangers. The analyze are conducted for varried flow rate and volume fractions ranging from 1 to 10 (LPM) and 2and 3 percent respectively. Comparison of these results with Rohit S. Khedkar 's published experimental data[1], showed good agreement.

2 Mathematical modeling

2.1 Problem geometry

The geometry of the present problem is shown in Fig 1. It consists of an isothermally concentric circular double tube heat exchanger with a length of 1000 mm and with the inner and the outer diameters of 6 mm and 16 mm respectively. Also, Fig 2 and Eqe (1) shows the schematic of isothermally concentric sinusoidal

Recebido: dia/mês/ano Aceito: dia/mês/ano

double tube heat exchanger and its surface equation. The minimum diameter and maximum diameter of inner tube is 4 mm and 8 mm and the diameter of outer tube is 16 mm and with a length of 1000 mm. Nanofluid flows inside the inner tube while hot pure water flows in outer tube. Nanofluids that enter the inner tubes composed of Al_2O_3/water with particle diameter of 20 nm. The computational model geometry and the grid were generated using GAMBIT 2.4.6 and the system of governing equations was solved by the control volume approach using the ANSYS FLUENT 15.0 software. Table.1 shows the thermophysical properties of base fluid and nanoparticles.

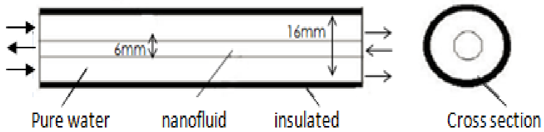

Fig1 1:Schematic of considering configuration of circular double tube heat exchanger.

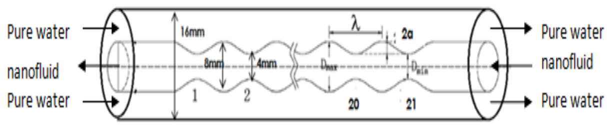

Fig 2:Schematic of considering configuration of circular double tube heat exchanger.

$$y = a\sin\left(\frac{2\pi}{\lambda}x - \frac{3\pi}{2}\right) + \frac{D_{\max} + D_{\min}}{4} \tag{1}$$

Table1 1:Thermophysical properties of materials under consideration.

Material	Density (Kg/m3)	Specific heat (J/Kg K)	Thermal conductivity (W/m k)
Al_2O_3	3880	773	36
water	998.2	4182	0.6

2.2 Numerical Method

The modeled cases were solved using ANSYS FLUENT software version 15.0. The renormalization group (RNG) k-ε model is used to simulate turbulence. In the numerical solution, finite volume method is utilized for solving the equations. PRESTO and QUICK Scheme is used for pressure correction and volume fraction respectively. A segregated, implicit solver option was used to solve the governing equations. The second order upwind discrimination scheme was used for the terms in energy, momentum, and turbulence parameters. A standard pressure interpolation scheme and SIMPLE pressure velocity coupling were implemented. Different non-uniform grids for heat exchanger inner and outer tubes are tested to insure independency of solution. 124000 and 166000 cells for the inner and the outer tube of concentric circular and sinusoidal double tube heat exchangers are sufficient for the present study respectively. Finer mesh is used near the wall because of higher velocity and temperature gradient.

2.3 Boundary Conditions

The fluid enters the inner tube with 1,2,4,6,8,9,10 (LPM) flow rates with uniform temperature of T = 28 ^{OC} and the flow rate of outer tube is 3.5 (LPM) with temperature of T = 80 ^{OC} .

2.4 Thermophysical Properties of the Nanofluids

Nanofluid thermophysical properties play a significant role in the accuracy of the results. The effect of nanoparticles can be taken into account by using the thermophysical properties of the nanofluid in the governing equations. In analyzing nanofluid as a two-phase flow, the interactions between nanoparticles and liquid are modelled too. The following formulas are used to compute the thermophysical properties of the nanofluids under consideration.Pak and Cho (1998) [2] equation is appropriate for using nanofluid density:

$$\rho_{nf} = (1-\phi)\rho_{bf} + \rho_{np} \tag{2}$$

Also the specific heat of nanofluid are achieved by the following equation:

$$C_{pnf} = \frac{(1-\phi)(\rho C_p)_{bf} + \phi(\rho C_p)_{np}}{(1-\phi)\rho_{bf} + \phi\rho_{np}} \tag{3}$$

Where subscripts of nf, bf and np indicate the nanofluid, base fluid (water in this case) and nanoparticles properties, respectively. Hamilton and Crosser [8] proposed a correlation for

Recebido: dia/mês/ano Aceito: dia/mês/ano

thermal conductivity. This correlation defined as:

$$K_{nf} = (4.97\phi^2 + 2.72\phi + 1)K_{bf} \quad (4)$$

As a general and accurate model for prediction of the viscosity of a nanofluid, μ, is not available at this moment. The effective dynamic viscosity of nanofluids is approximated by means reported in the literature. On one hand, it can be calculated using several existing formulas that have been derived for two-phase like one, proposed by Maiga et al [9]. The correlation proposed by Maiga, based on fitting curves through regressio analysis of experimental data, is used:

$$\mu_{nf} = (123\phi^2 + 7.3\phi + 1)\mu_{bf} \quad (5)$$

Where the viscosity of the base fluid which is defined as:

$$\mu_{bf} = AT^{\frac{B}{T-C}} \quad (6)$$

where, $A = 2.414*10^{(-5)}, B = 247, C = 140$

Also, Prandtl and Reynolds numbers presented as:

$$R_e = \frac{pvd}{\mu} \quad (7)$$

$$\mathrm{Pr} = \frac{Cp\mu}{k} \quad (8)$$

2.5 Governing equations

The VOF model solves a single set of momentum equations for all the phases and tracks their volume fraction all over the domain of study by solving a continuity equation for the secondary phases. The total summation of the volume fractions for all the phases is equal to unity. Therefore, the magnitude of the primary phase volume fraction will be calculated. In this method, all of the physical properties are calculated by taking a weighted average of different phases based on their volume concentrations throughout each control volume. The single set of momentum equation is solved to find the velocity components, which are shared by all the phases. In the same manner, a shared temperature is obtained from a single energy equation. Specifically, mass conservation is proposed as:

$$\nabla.(\varphi_q \rho_q \vec{V_q}) = 0 \quad (9)$$

where $\sum_{q=1}^{n} \phi_q = 1$ and all properties are calculated like $N = \sum_{q=1}^{n} \varphi_q N_q$. The conservation of momentum and energy equations are identical to Eqs. (10) and (11).

$$\rho\vec{V}.\nabla\vec{V} = -\nabla P + \nabla.(\mu\nabla\vec{V}) + \rho g \quad (10)$$

$$\nabla.(\vec{V}(\rho E + P)) = \nabla.(K\nabla T) \quad (11)$$

Also, the heat absorbed by the coolant in the inner tube is:

$$q = mc(T_{out} - T_{in}) \quad (12)$$

Where q is the heat transfer rate and U is overall heat transfer coefficient is expressed as follows:

$$U = \frac{q}{A_0\,(LMTD)} \quad (13)$$

Also Ao is the surface area; and LMTD is the log mean temperature difference, based on the inlet temperature difference, T1, and the outlet temperature difference, T2. LMTD is calculated as follows:

$$LMTD = \frac{\Delta T_2 - \Delta T_1}{Ln(\frac{\Delta T_2}{\Delta T_1})} \quad (14)$$

And the average Nusselt number of pure water is defined as:

$$\begin{aligned} &Nu = 0.024\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \\ &0.5 < \mathrm{Pr} < 120 \\ &6.0*10^3 < \mathrm{Re} < 1.0*10^7 \end{aligned} \quad (15)$$

And the average Nusselt number of nanofluid is calculated using

$$\begin{aligned} &Nu = 0.085\,\mathrm{Re}^{0.71}\,\mathrm{Pr}^{0.35} \\ &6.6 \le \mathrm{Pr} \le 13.9 \\ &10^4 \le \mathrm{Re} \le 5*10^5 \\ &0 < \varphi < 10\% \end{aligned} \quad (16)$$

3 Results and discussion

Recebido: dia/mês/ano Aceito: dia/mês/ano

In Figure 3, numerical simulation of overall heat transfer coefficient of base fluid in circular double tube heat exchanger has been carried out for the following values. First, we test the validity of the method used in this work. To do this, overall heat transfer coefficient of pure water flow through the inner and outer pipes of double tube heat exchanger has been numerically investigated and numerical results are compared with the experimental data of Rohit S. Khedkar et al 2013 [1]. The numerical results are in good agreement with experimental results. The difference between them almost is 2% .

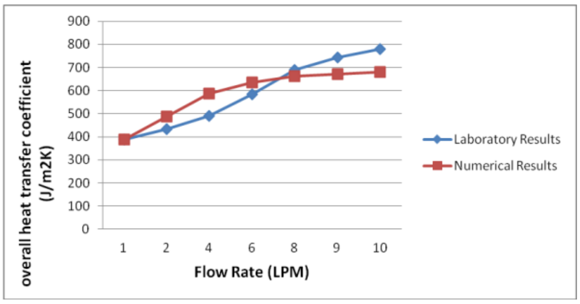

Fig3 3:Validation of numerical values with laboratory results of the overal heat transfer coefficient in the circular double tube heat exchanger.

Lack of experimental data, the solid viscosity for a liquid–solid two phase mixture is not available. So, for the first degree of approximation the following method is adopted in the present study. The corresponding overall heat transfer coefficient of a highly dilute nanofluid with volumetric concentration of 0.00001 (which is quite close to pure water), is compared with that of pure water, for flow rate of 10 LPM. Using the trial and error method, the value of viscosity in the solid phase of the highly dilute nanofluid is changed up to a point where the overall heat transfer coefficient of the highly dilute nanofluid and the pure water are matched. Fig 4 shows the viscosity of the solid phase reaches a value of 0.015 Pa s.

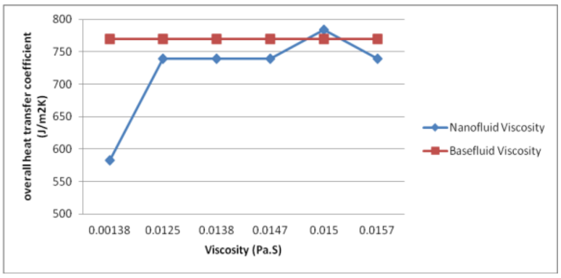

Fig4 4:Calculation of nano particle viscosity.

According to Figure 5, in the circular double tube heat exchanger, the results of overall heat transfer coefficient of nanofluids by two phase-VOF method is more accurate than single-phase model and the average error is dropped more than 5%.

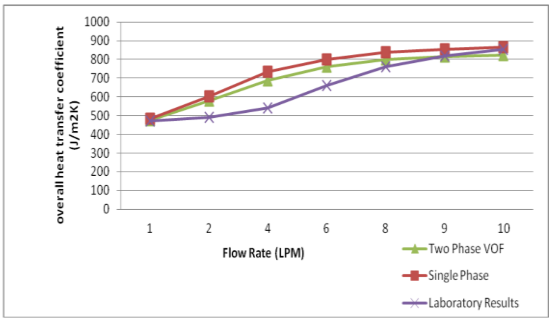

Fig5 5:Comparison between the laboratory and the numerical results of singel phase and two phase-(VOF) methods of the overal heat transfer coefficient in the circular double tube heat exchanger.

In Figure 6, the effects of particle volume concentration on the overall heat transfer coefficient of water–Al_2O_3 nanofluid at different flow rate are presented. The influence of the addition of nanoparticles to the base fluid on overall heat transfer coefficient enhancement is also visible in this figure. It can be seen that the nanofluid results show higher overall heat transfer coefficient enhancement in comparison to pure water results. Also, the overall heat transfer coefficient enhancement increases almost linearly with increase in nanoparticle volumetric concentration for two-phase modeling results. For instance, for 2% nanoparticle volume concentration, two-phase modeling shows 20% overall heat transfer coefficient enhancement in comparison to pure water results in flow rate=10 LPM, while it is 22% for 3% nanoparticle volumetric concentration results in circular heat exchanger.

Recebido: dia/mês/ano Aceito: dia/mês/ano

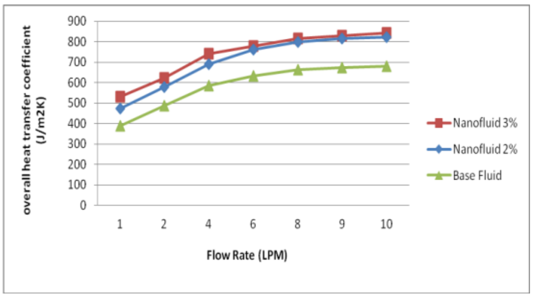

Fig6 6:Effects of nano particles volume concentration on the overal heat transfer coefficient ratio in the circular double tube heat exchanger.

Fig 7 shows the average Nusselt number for nanofluid in different flow rates and volume concentrations. According to Fig. 7, the average Nusselt number increases with an increase in nanoparticle volume concentration and flow rates. The increase in the average Nusselt number is due to higher convective effects and higher nanoparticle participation in the nanofluid effective thermal conductivity enhancement, respectively.

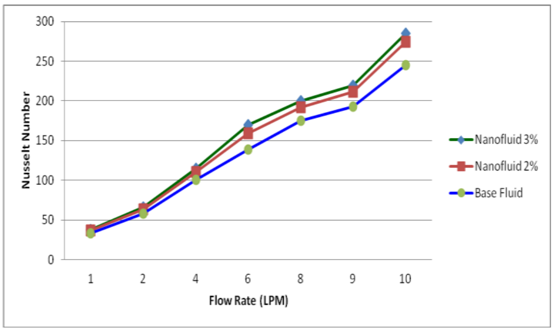

Fig7 7:Effects of nano particles volume concentration on the Nusselt number ratio in the circular double tube heat exchanger.

In practical application of nanofluids, it is essential to study the pressure drop, developed during the flow of such coolants. Therefore, the effect of Al2O3/water nanofluid on the friction factor, pressure drop was studied according to Figs. 8. As the nanoparticles loaded into the base fluid, increase the viscosity and density of the base fluid and also increase friction factor and pressure drop will be slightly increased. From the investigations, it can be inferred that, the pressure drop of nanofluids at low nanoparticle volume concentration range (2%-3.0%) is approximately the same as in the base fluid and that is negligible.

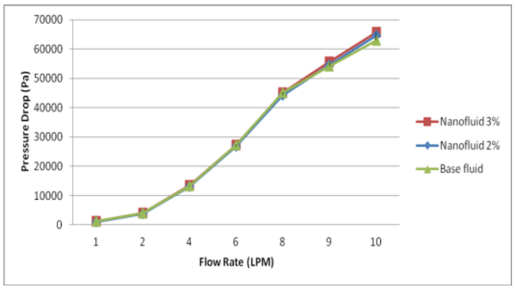

Fig8 8:Effects of nano particles volume concentration on the pressure drop ratio ratio in the circular double tube heat exchanger.

Figure 9, shows the ratio of wall shear stress average of nanofluid to base fluid at a typical value of flow rate = 10 LPM for two nanofluids. It is clear that addition of nanoparticles to the base fluid results in increasing the viscosity and the wall shear stresses. As can be seen from this Figure, for 2% nanoparticle volumetric concentration the wall shear stresses is smaller than one obtained by 3% nanoparticle volumetric concentration. Also, two curve approach approximately to the same asymptotic values. But the slope of variation in the 3% nanoparticle volumetric concentration curve is higher.

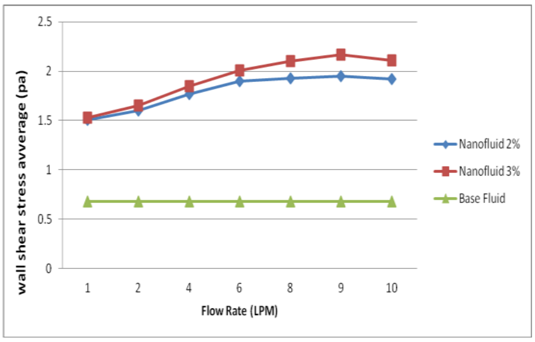

Fig9 9:Effects of nano particles volume concentration on the wall shear stress ratio ratio in the circular double tube heat exchanger.

As shown in Figure 10, addition of Al2O3 nanoparticles to pure water effectively enhances overall heat transfer coefficient in turbulent regime. The level of overall heat transfer coefficient increase depends on the shape of heat exchanger. As is seen, the effect of changing shape of heat exchanger on increasing overall heat transfer coefficient is less than adding nanoparticles to pure water. Although, addition of nanoparticles to pure water will more enhance overall heat transfer coefficient of sinusoidal double tube heat exchanger in

Recebido: dia/mês/ano Aceito: dia/mês/ano

comparison with circular one. For a fixed value of flow rate = 10 LPM, increase of the overall heat transfer coefficient of 2% nanofluid in the circular double tube heat exchanger is approximately 21 % while the level of the overall heat transfer coefficient reaches to about 220 % in sinusoidal one. In other words, computations with both methods show that the sensitivity of overall heat transfer coefficient enhancement to changing shape and adding nanoparticles to pure water for a fixed value of flow rate is very important.

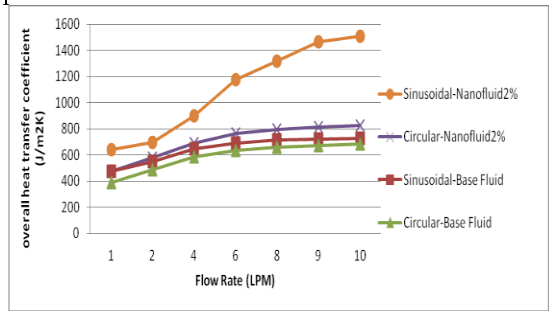

Fig10 10:The overal heat transfer coefficient nanofluid enhancement in comparison to pure water flow in circular and sinusoidal double tube heat exchanger.

According to Figure 11, in sinusoidal double tube heat exchanger, the overall heat transfer coefficient will be almost "three-fold and four-fold in comparison with pure water by increasing the concentration of nanoparticles to 2% and 3%.

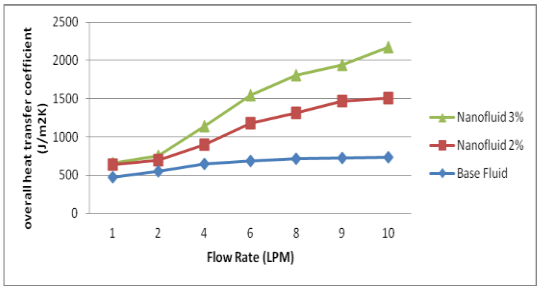

Fig11 11:Effects of nano particles volume concentration on the overal heat transfer coefficient ratio in sinusoidal heat exchanger.

The level of pressure drop increase depends on the volume concentration. For the value of φ = 2%, the pressure drop is approximately same as pure water while at the upper limit of φ (φ=3%), the level of pressure drop increases. Also, computations show that the sensitivity of the pressure drop enhancement to volume concentration for a fixed value of flow rate is negligible.

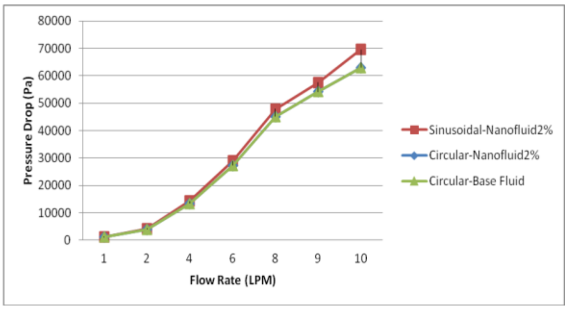

Fig12 12:Effects of nano particles volume concentration on the pressure drop ratio in sinusoidal heat exchanger.

4 CONCLUSIONS

In this study, we investigate numerically the heat transfer of base fluid and nanofluids with different volume concentration of nanoparticles in circular and sinusoidal double tubes heat exchangers. The results indicate an increase in overall heat transfer coefficient , Nusselt number, surface tension and low pressure drop nanofluids compared to the base fluid. Moreover, the results of two phase - VOF model compared to single-phase was much closer to the experimental results. Also the impact of geometry of heat exchanger on overall heat exchanger cofficient is less than the impact of nanoparticles. If you change the geometry of heat exchanger and use nanofluid instead of base fluid, the overall heat exchanger cofficient will be dramatically increased.

REFERENCES

Rohit S. Khedkar, Shriram S. Sonawane*, Kailas L Wasewar, Water to Nanofluids heat transfer in concentric tube heat exchanger: Experimental study,Procedia Engineering (Elsevier journal), Volume 51, 2013, Pages 318–323

B. C. Pak and Y. I. Cho, "Hydrodynamic and heat transfer study of dispersed fluids with submicron metallic oxide particles," Experimental Heat Transfer, vol. 11, no. 2, pp. 151–170, 1998.

Li, Q. and Xuan, Y.: Convective heat transfer and flow characteristics of Cu-water nanofluid, Science in China, Serie E: Tech. Science, Vol. 45, No. 4,pp. 408-416, 2002.

Jang, S.P. and Choi, S.U.S., "Effects of Various Parameters on Nanofluid Thermal

Recebido: dia/mês/ano Aceito: dia/mês/ano

Conductivity", ASME Journal of Heat Transfer, Vol. 129, (2007), 617-623.

Byung-Hee Chun, Hyun Uk Kang and Sung Hyun Kim, Effect of alumina nanoparticles in the fluid on heat transfer in double pipe heat exchanger system, Korean Journal of Chemical Engineering,25(5),2008, pp. 966-971.

R. Lotfi, Y. Saboohi, A.M. Rashidi, Numerical study of forced convective heat transfer of nanofluids: comparison of different approaches, Int. Commun. Heat Mass Transfer 37 (2010) 74e78.

Mehdi Bahiraei and Morteza Hangi, Investigating the efficacy of magnetic nanofluid as a coolant in double-pipe heat exchanger in the presence of magnetic field, Energy Conversion and Management, Volume 76, December 2013, Pages 1125-1133

Hamilton RL, Crosser OK. Thermal conductivity of heterogeneous two com ponent systems. I&EC Fundamentals 1962;1(3):187–91.

Maiga, S.-E.-B. N. Cong Tam, N. Galanis, G. Roy, T. Mare, & M. Coqueux . (2006). Heat transfer enhancement in turbulent tube flow using Al2O3 nanoparticle suspension. International Journal of Numerical Methods in Heat and Fluid Flow, 16 , 275-292.

Recebido: dia/mês/ano Aceito: dia/mês/ano