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STUDY AND THERMAL ANALYSIS ON TURBULENT NANOFLUID FLOW IN HELICAL TUBE HEAT EXCHANGERS

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Abstract:- In this study, the thermal characteristics of turbulent nanofluid flow in a helical tube in the tube heat exchanger were assessed numerically through computational fluid dynamics (CFD) simulations. The results of both of the turbulent models: i.e realizable k-epsion (k- ε) and re- normalization groups (RNG) k-epsilon were compared. The thermal distribution contour determines that realizable and RNG k-ε models, both together a swirl dominated flow are of more uniform distributions of temperatures. The proper prediction of the two layer theory will results to have a uniform thermal distribution and proper dimension less wall distance let it be (Y+). The turbulent flow and heat transfer of the two nano fluids i.e. SiO2, Al2O3 and a known base fluid with respect to swirl dominated flow was simulated through a RNG model. The effects of the concentration of nano particles on heat transfer characteristics in HTTHE and two turbulent models were analyzed into a comprehensive manner. It is concluded that upto 1% concentration of SiO2 and 1% concentration of Al2O3, similarly heat transfer characteristics will be observed. And the relative Comparison between the CFD simulations results with the predicted values for friction factor coefficient (f) and a Nusselt's number (Nu) will be calculated by an experimental correlations that will indicate the maximum errors of 6.56% and 0.27%, respectively.

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I INTRODUCTION

Numerous experimental and numerical assessment are being published to establish the better understanding of the nano fluids behavior [1], thermal properties in heat exchangers [2], their models graphs [3] and CFD simulation [4]. The related literature review indicates that there are few and cases thare are being run on HTTHEs [5]. These heat exchangers will create turbulence that will promote swirl in a convective heat transfer in the fluid respectively which eventually cause effective mixing of the fluid by providing a major surface area. Helical tube exchanger is superior because of its constructional configuration and generating secondary flow compared to the conventional tube heat exchangers. The secondary flow and increased heat transfer potential of nanofluids in the helically coiled tube heat exchangers is applied so as to increase the heat exchangers efficiency; thereby, a reduction of size in heat transfer of the two nanofluids i.e. SiO2, Al2O3 and a known base fluid with respect to swirl dominated flow was simulated through a RNG model. The effects of the concentration of nanoparticles on heat transfer characteristics in HTTHE and two turbulent models were analyzed into a comprehensive manner. It is concluded that up to 1% concentration of SiO2 and 1% concentration of Al2O3, similarly heat transfer characteristics will be observed. And the relative Comparison between the CFD simulations results with the predicted values for friction factor coefficient (f) and a Nusselt's number (Nu) will be calculated by an experimental correlations that will indicate the maximum errors of 6.56% and 0.27%, respectively.

Exchangers construction [6]. Due to the growth of industrial and commercial applications regarding heat transfer with respect to the energy and environmental concerns, these exchanger configurations have been improved to transfer heat with higher effective process. Applying the fin , tube [7]

[8] heat exchanger, these are well equipped with helical surfaced membrane coils [9] and by improving the thermal conductivity of base fluid will enhance the convective heat transfer of these setups. Examiners recommend that

improving the thermal conductivity of base fluid to enhance the performance of heat exchangers by applying nano fluids, which are a liquid and solid mixture of nano particles like metals, oxides, and some other vital compounds. Here, the base fluids usually consist of water, oils, and alcohols. Due to larger surface area of the nanoparticles, nano fluids will obtain superior properties, in high thermal conductivity, stability, and homogeny conditions will be achieved. [10].

The improvement and enhancement in thermal conductivity of nano fluids will merely depends on temperature, particles volume fraction, atomic shape, type, and some internal properties. Vajjoha and Das has [11] assessed the dependency of thermal conductivity on both the temperature rises from 298 to 363 K and particle concentration up to 10 %. The results indicate that an increase in temperature and concentration of nanoparticles will literally leads to an increase in the thermal conductivity and thermal effectiveness of fluids in comparison with the supplied base fluids.

An experimental investigation was carried out by Haeshemi and Akhiavan-Behabedi [12] to assess the pressure drop and heat transfer properties of nano fluid flow inside a horizontal constructed helical tube subjected to a laminar flow regime and constant heat flux. The CuO nanoparticles were dispersed in an industrial oil with a concentration of 0.56–2 %. The effect of some parameters like fluid temperature, Reynold number (Re) and nano fluid particle concentrations on pressure drop of the flow and heat transfer coefficient, were examined These results have suggested that by using the helical coil tube and the nano fluid instead of base fluid, the heat transfer performance improves. Naerrein and Muhamad [6] performed a numerical assessment on the effects of different nanoparticle, with different diameters and volumetric concentrations in base fluid types of water, engine oil, and ethylene glycol, which are influencing the thermal and hydraulic characters in helical coiled tube heat exchanger subjected to a laminar flow. They concluded that the Nu is at its highest when CuO has been taken into consideration for utilizing them.

The thermal performance of a single and hybrid type of nanofluid were assessed in a coiled heat exchanger in a laminar flow operating conditions respectively and constant wall temperature [13]. The maximum heat transfer rate was achieved using hybrid type at a concentration rate of 0.4 volume percentage that is 31.58 percent higher compared to water. Their results had indicated that nano fluids would improve the thermal properties and performance of the exchanger, while this fact is accompanied by an increase in the overall pressure drop.

The heat transfer characteristics of Al2O3-Cu/water hybrid nano fluid had examined in a permeable channel by Mollaah Ahmadi et al. [14]. The effects of the Re and the concentration of nanoparticles, on the heat transfer, were examined. The results shows that with the increase in the Re, increases the Nu. By using the hybrid nano fluid, instead of a pure nano fluid, the heat-transfer coefficient increased significantly.

The effect of using Al and Cu nano fluids in a conventional heat exchanger in a spiraled coil had been analyzed by Tajik Jamal-Abaad . [15]. The results showed that the nanofluid significantly increases the convective heat transfer coefficient, and Cu-water nanofluid has more effective thermal characteristics than Al- water nanofluid.

From reviewing the previous literature, it is clear that no sufficient attempts were made to study the effects of different turbulent models and different nanoparticles in helical coiled heat exchangers under turbulent flow regime in both inner and annulus tube sides. In order to evaluate at high rate on this issue, an attempt was made to assess and examined a 3D turbulent flow with respect to the effects of different turbulent models. The main objective of this article is to study the heat transfer behavior and pressure drop of water-based SiO2 and Al2O3 that are subjected to the different turbulent models and analysis. It is expected that the research’s here could fulfill the evaluation gap regarding HTT heat exchangers operating with nanofluids up to a certain range.

II.METHODOLOGY

a. Numerical model

A commercially CFD simulation code will be applied to run the numerical simulations and calculations of a subjected 3D geometry. The computational geometry simulation was developed in Gambit, and heat transfer and pressure drop analysis was done using Fluent. The considered heat exchanger geometry withdi,i, di,o, R, and H dimension’s will be recorded in the given Fig. 1.Inlet conditions and geometrical parameters of HTT heat exchanger are presented in the below shown Table A. The cold nanofluid flow in the annulus side of the tube, while the hot nanofluid flow inside the inner coiled tube. The material of the heat exchanger component was made up of copper. Their physical properties are tabulated in Table B. As per the obtained result by Rea et al. [16] , the single-phase model will be applied frequently for nano fluids thermal behavior prediction.

b. Governing equations

Setting the governing equations to complete the CFD analysis of the HTT heat exchanger is of a major importance. In this study, the RNG k-ε turbulent model with a enhanced wall treatment and also swirl dominated flow, the RNG turbulent model with a enhanced treatment of wall conditions without swirl dominated flow and the realizable k-ε model have been selected. The mentioned equations have been written as:

Table A. Geometry and inlet conditions of CTITHE

	Inner tube	**Outer tube**
Outer diameter, m	0.00635	0.01587
Inner diameter, m	0.00475	0.01407
Coil diameter, m	0.3	0.3
Pitch, m	0.03174	0.03174
Number of turns	1.5	1.5
Tube material	Copper	Copper
Flow rate, LPM	2	10
Inlet temperature, °C	50	20

C.Grid testing and model validation

Grid testing

The 3D geometry of this heat exchanger with a meshing would be displayed in Fig. 2. A grid independence analysis will be run to analyze the effects of different grid sizes on the obtained results. The grid independency will be checked through sets of 4 hexahedral meshes i.e. 325200, 461000, 688000, & 753200 cells, respectively. The last two types of mesh yielded similar results with respect to the heat transfer coefficients and evaluated Y+. Here in this study, by considering the precision of obtained outcomes (temperature distribution and Y+) and in order to decrease the computational time, the overall set of 688000 cells have been selected.

Validation

The validations will be made according to the boundary conditions and geometrical conditions respectively, that are being introduced by Aly [5]. For the purpose of computation and recording purpose, the realizable k-ε turbulent model is taken into consider. The code was validated by comparing between the obtained flow results with the experimental relations that are introduced by Genielinski [17], Mishra [18] and Ito [19].

The obtained results will then compared with the constant experimental correlations presented in Table 3. It has been observed that results show the agreement with Eqations.(5) through(7)

Numerical procedures

The numerically computed simulations have been run by solving the Equations. (1-4), thereby applying the finite volume formulation. The numerical solution procedure will adopt the SIMPLEC modelled coded algorithm for the pressure velocity couplings. The second-order upwind phenomenon was applied to the momentum, turbulent kinetic energy- dissipation value, and third-order QUICK discretization scheme was applied for energy

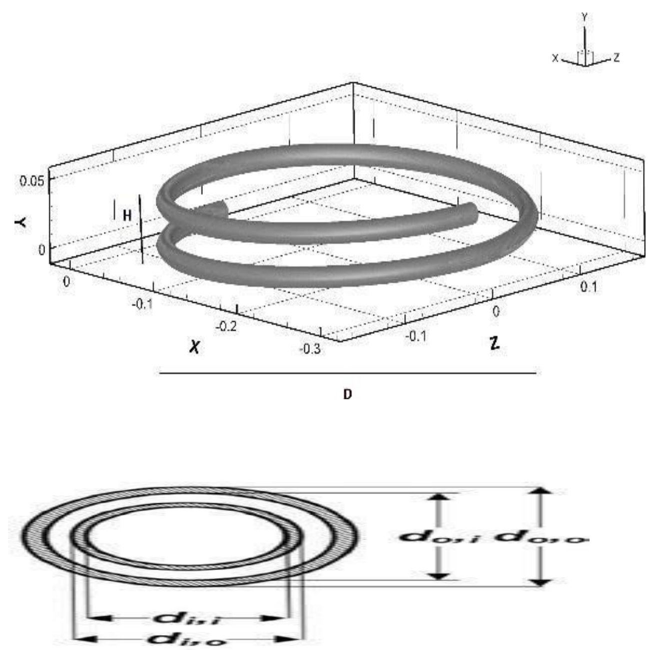

Figure 1. Schematic diagram and cross section of CTITHE

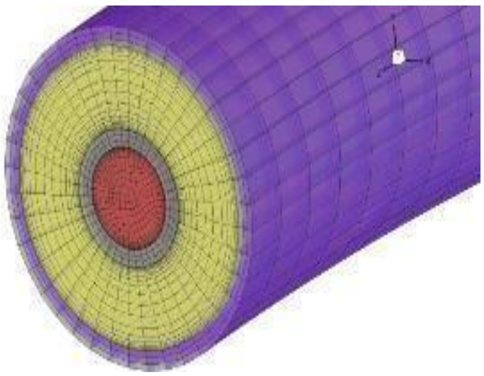

Figure 2. Grid system for CTITHE

Table 3. Friction factor and Nusselt number correlations

Author	Correlation	Condition	Eq.
Gnielinski[17]	$Nu = \frac{Pr.Re(f/8)}{1.0 + 12.7 f / 8(Pr^{2/3} - 1)}$ $f=0.3164Re^{-0.25}+0.03\delta^{0.5}$	$Re > 2.2\times10^{4}$	(5)
Mishra Gupta [18]	$f=f_s+0.03\delta^{0.5}$ $f_s=0.3164Re^{-0.25}$	$4500<Re<10^{5}$	(6)
Ito [19]	$f=0.304Re^{-0.25}+0.029\delta^{0.5}$	$300>Re\delta^{2}>0.034$	(7)

Table 4. Thermal and physical properties of the base fluid and the nanoparticles [20, 21].

Property	Water	Al_2O_3	SiO_2
Density (kg/m^3)	998.2	3300	2200
Specific heat (J/kg.K)	4182	880	703
Thermal conductivity(W/m.K)	0.6	42.34	1.2

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III RESULTS AND DISCUSSION

In this study, the chosen nanoparticles were effective for the transfer of heat and its application that are owed to their superior specific heat and thermal conductivities as well as lower viscosities even at a high concentration rate, good dispensability [22], higher efficiency in the case of considering nanoparticle economy [23].

The relative function of the Re denotes that the improvement cum enhancing of the convection heat transfer.similarly, the nanofluids will obtain the higher Nu than water with an equal Re. The known relation of Dittus-Boelter suggests that the increase in Re leads to increase in Nu and hence, the heat transfer coefficient of nanofluid [24]. The comparison of the Nu of SiO2 nanofluid with that of Al2O3 nanofluid results obtained by Aly [5] in the inner and annulus tube sides of the HTT heat exchangers which are presented in Fig. 3 (a) and (b), respectively. The Nu from Fig. 3 was compared when subjected to the same Re in 11600 - 28120 range. The numerical results indicated that due to the low thermal conductivity of SiO2 nanofluid, Nu is higher than that of other nano fluids. An gradual increase in Re leads to increment in Nu. The calculated AARE percentage of Nui and Nuo in this study in relation to Gnielinski's correlation are as 1.4 % and 3.6 %, respectively. The obtained results revealed the good agreement in between the computation analysis model and the experiment correlation, that will confirm the accuracy of this model.

The frictional factors calculated from the CFD simulation results with respect to pressure drop and the variant frictional factors with respect to Re for SiO2 nanofluid and their comparison with water and the results reported by Mishra and Gapta and Ito are also represented in Fig. 4. According to Eq. (6), an increase in Re will results to a reduction in the obtained f [25]. This fact is confirmed through this study, thereby which f decreases when Re increases (see Fig. 4). It is also expected from the particles Brownian movement by adding nano particle to the water, thereby causing an increase in the momentum transfer, the obtained f for nanofluid will be more than that of water. It is said to be noted that according to Fig. 4, the Ito's model depicts less f than Mishra and Gupta's model subjected to the as usual Re. this leads to , Ito's equation is not a proper correlation for predicting the nanofluid pressure reduction.

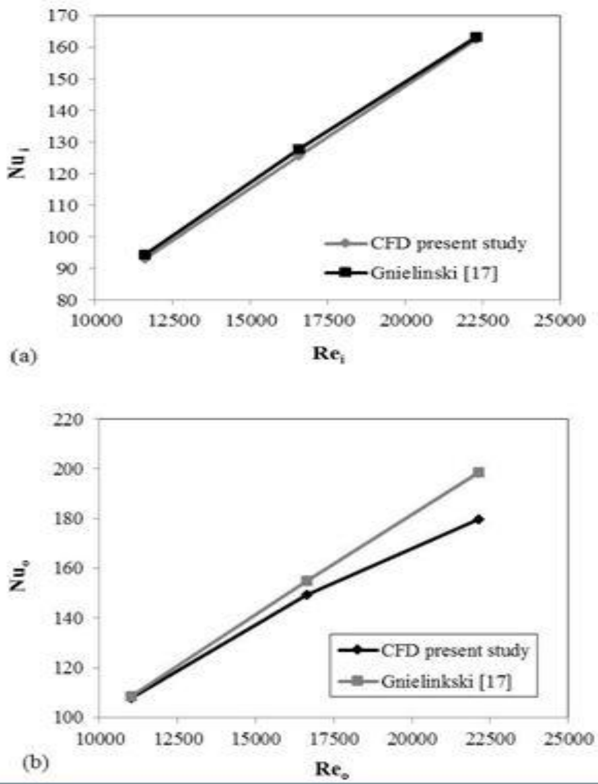

Figure 3. Comparison of Nu between the present study and Gnielinski [17] based on Re(a) inner tube side, (b) annulus tube side.

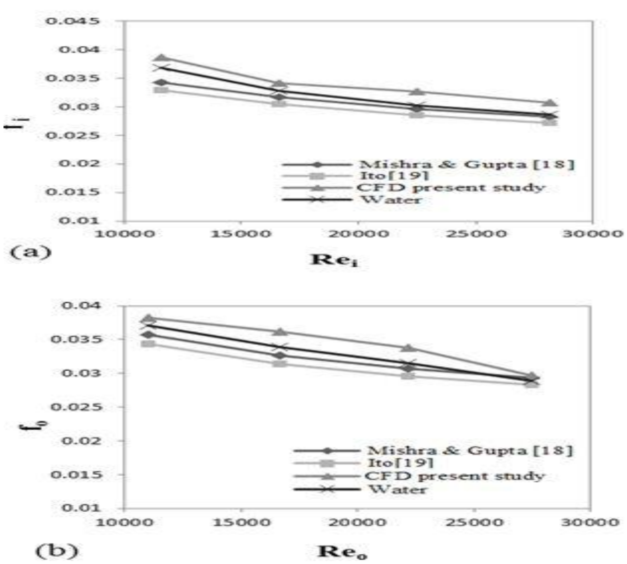

Figure 4. f versus Re(a) inner tube side, (b) annulus tube side

The heat transfer coefficient and Re for HTT heat exchangers have been shown in Fig. 5 (a) and (b) for the inner cum annulus tube sides. As the increase in Re, the significance of thermal conductivity in heat transfer enhancement becomes less considerable [26]. It is observed from Fig. 5 that the highest heat transfer coefficient is obtained for SiO2 and Al2O3 nanofluids. At 1% volume concentration of SiO2 nanofluid and Al2O3 nanofluid, the heat transfer coefficient of considered the nanofluids is 8.60 % and 8.2 % greater than the basefluid in annulus tube side, respectively; while this value is 4.30 % and 1.5 % in inner tube side. This behavior is due to the wall effect, that leads to the turbulent flow. Due to an increase in heat transfer coefficient by using SiO2 and Al2O3 and therefore an improvement and enhancement in the heat transfer characteristics of both the nanofluids, and these nanofluids were recommended to be used in heat exchangers. The possible explanation behind this phenomenon is the fact that at 1% concentration of SiO2 and 1% concentration of Al2O3, these two nano fluids show same behavior.

The CFD simulated Nu versus the predicted Nu of Eq.

(5) and f of Eq. (6) are presented in Figs. 6 (a) and (b), respectively. In this figure, the values of predicted Nu and f are in accordance with the simulated ones in this study. Here, the maximum error of Nu is 6.56 %, and the same amount for f is 0.27 %. It is deduced that Eq. (6) is valid for the tested nanofluids in the turbulent flow regime. In this study, it is found that nanofluids act as a homogeneous fluid.

Thermal distribution contours for all the 3 various simulations for inner and annulus tube sides have been shown in Figs. 7 and 8, respectively. We can observe that the RNG model will not predict that the nano fluid behavior in thermal distribution contours in a effective way, by providing that the swirl dominate flow will considered in the equations. We can see that in Figs. 7 and 8 that RNG model shows the same result with the k-ε model when the swirl dominated flow has been applied; thus, the RNG model is adopted in evaluating the heat transfer characteristics of nano fluid in HTT heat exchangers. These contours show that realizable k-ε and RNG k-ε models, together with the swirl dominated flow are of more uniform thermal distribution. Temperature differences are shown in Figs. 7 (a), (b) and (c) are 3.9, 3.7 and 12.6 for a inner tube side and the same at Figs. 8 (a) , (b) & (c) are 5, 3.7

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and 9.3 for annulus tube side,. Thus the RNG model, together with the swirl dominated flow shows more uniform thermal distribution at any cross sectional than remaining other two models. In a previous study [9], it has been reported that realizable k-ε model was accurate and accuracy was at high range than the RNG turbulent simulation model, while, according to the obtained results in this study (Figs. 7 and 8), the RNG model could depict the nano fluid behavior in thermal distribution contours better to realizable k-ε model, by considering the swirl computational simulation model.

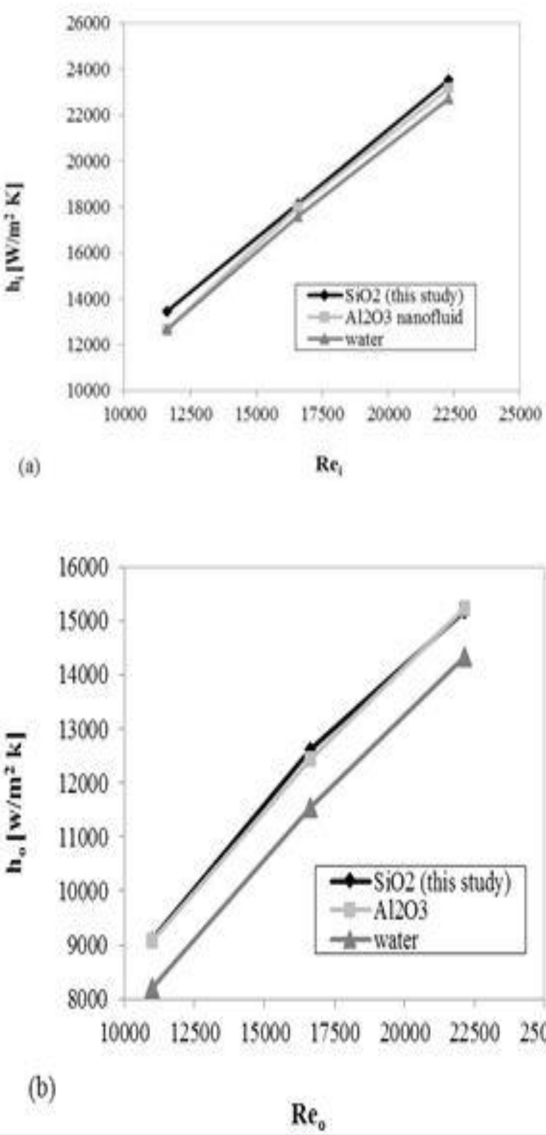

Figure 5. Heat transfer coefficient versus Re(a) innertube area , (b) annulus tube side

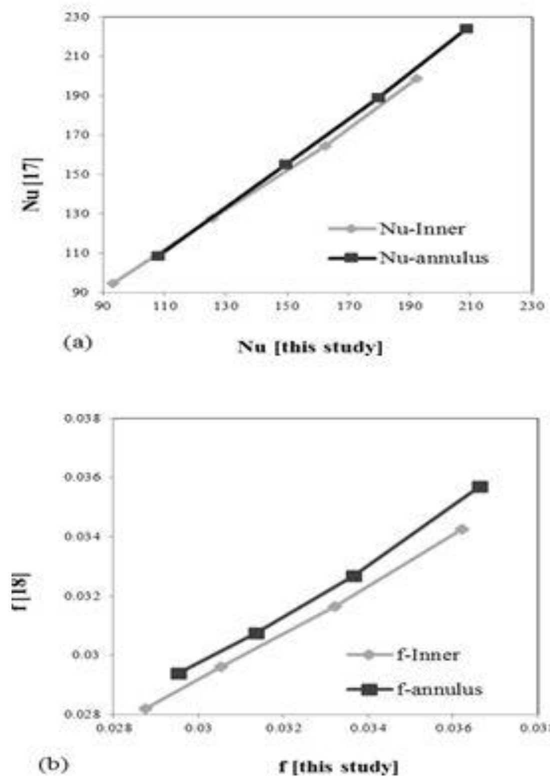

Figure 6. Comparison between the simulated values of (a) Nu, f, with those predicted by Gnielinski [17] and Mishra and Gupta [18], respectively

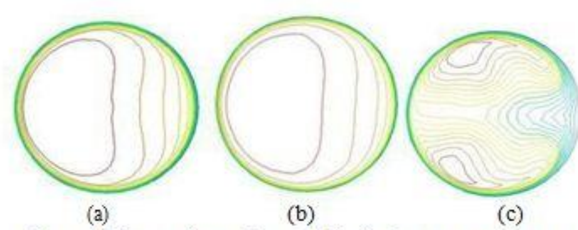

Figure 7. Comparison of thermal distribution contours among (a) realizable, (b) RNG, (c) RNG where no dominated flow model for inner side of the tube is of discussed

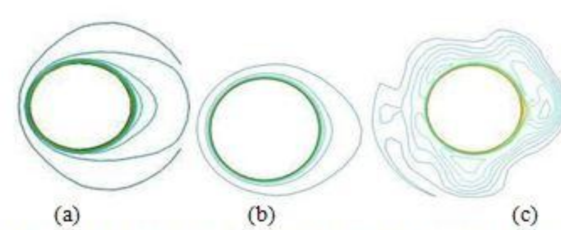

Figure 8. Comparison of thermal distribution contours among (a) realizable, (b) RNG, (c) RNG where nodominated flow model for outer side of tube is discussed.

The temperature vector in HTT heat exchangers for RNG model where the swirl dominated flow is of concern and the three different 325200, 461000, and 688000 cells are shown in Fig. 9. As we observe here, when the mesh size increases, the temperature profile for nanofluid is higher. As for the vectors in this figure, the laminar flow is evident near to wall, and turbulent flow is evident in the central parts of the tube. The phenomenal behavior is clearly observed at a range of 688000 cells, where small temperature range due to the systematic establishment of the boundary layers is observed as well. This simulation experiment analysis will helps in the overall improvement of the convective heat transfer coefficients at the boundaries, by which the considerable improvement in the efficiency of heat transfer is obtained. It is said that when the mesh size decreases, the value of Y+ decreases; as for 688000 cells, the value of Y+is almost 1. According to the result obtained by Aly [5], the exact range for Y+ is 1. Thereby it is concluded that the value obtained here by 688000 cells is an appropriate value for simulation due to the proper value of Y+ and clear thermal profile.

IV CONCLUSION

In this article, the properties of pressure drop and convective heat transfer of SiO2 nanofluid flow in HTT heat exchangers have been assessed. The following conclusions can be defined based on the obtained results:

• The 3D k-ε RNG model with & without swirl

dominated flow, and realizable k-ε turbulence model will be assessed. Due to constant rate temperature distribution, the two layers theory and proper Y+, the RNG k-ε model with swirl dominate flow is proper to simulation of the turbulent flow in HTT heat exchangers.

• The changes in f and Nu are against Re for SiO2 and Al2O3 nanofluids at a value of 1 % . and it was also assessed. The results obtained from the CFD model analysis of Al2O3 denotes a good agreement with that of reported in the related literature. The heat transfer coefficient for SiO2 and Al2O3 nanofluids at 1 % concentration in a separate manner demonstrated a better outcome than that of water.

• Comparison between the computational fluid analysis study against some known values for f and Nu through experimental correlations indicate that experimental correlations are established based on single phase fluid data which holds true for multiphase flow with maximum errors approximately < 6.56%.

Nomenclature

Cp Specific heat at a constant pressure, C Model parameter

D Coil diameter, m

Dn Deans Number

d Tube diameter, m

f Darcy–Weisbach friction factor

h Heat transfer coefficient

H Coil pitch, m

k Turbulence K.E, m2/s2

L Length of the tube, m

m Mass flow rate, kg/s

K Thermal conductivity, W/m.K Nu Nusselts number

P Pressure, Pa

Pr Prandtl number

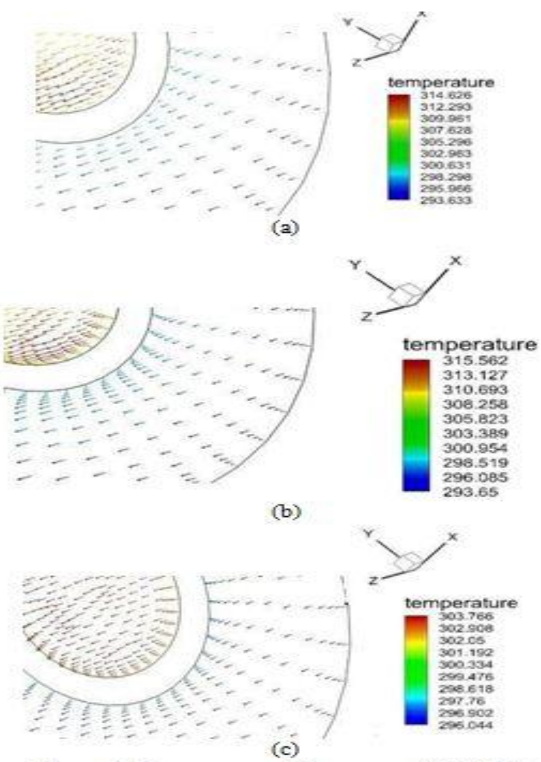

Figure 9. Temperature profile vectors in CTITHE (a) 3252000, (b) 461000 and (c) 688000 grid cells

Greek symbols

B Curve-fit relations

δ Curvature ratio

δij Dirac delta function

Turbulent dissipation rate, m2/s3ρ Density of test fluid, kg/m3

µ Dynamic viscosity, kg/m.s

στ Turbulent Prandtl number in the energyequation

σk Diffusion Prandtl number for kΔ Difference operator

ϕ Nanoparticle volume concentration

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