Astray State-Laminar Forced Convective Heat Transfer of Al_2O_3 – H_2O Nanofluid through 3D-Rectangular Cross- Sectional Duct

G. Shiva Guru Prakash, Srinivaas Ashok Kumar

ABSTRACT--- A Steady state-laminar forced convective heat transfer has been simulated by Computational Fluid Dynamics (CFD) with a Single Phase Model (SPM), Multi Phase model & Diameter effects and also determined the effects of nanoparticles concentration and nanofluid flow rate through 3D rectangular duct under certain boundary condition (constant heat flux). The nanofluid contains Alumina nanoparticles of size 60nm diameter used for MPM which is mixed with base fluid (water) with volume fraction of 0% ≤ ¢ ≤ 5% and Reynolds number (Re) ranges from 250 ≤ Re ≤ 1000. ANSYS 18.0 has been used for simulation. Three cases of analysis have been carried out in which the thermal conductivity (k) and dynamic viscosity (μ) of nanofluids are determined using two sets of theoretical models and one set of experimental k & μ data from literature respectively. The nanoparticles which stay more dispersed in the base fluid due to increase in Reynolds number which improves HTC and also decreases the friction factor accordingly. Particular attention has been paid to the variation of heat transfer characteristics when the modeling approach is switched from SPM to MPM. It is revealed that higher heat transfer rates are observed in MPM. The results shows that the friction factor decreases and Nusselt number (Nu) increases when there is an increase in the flow rate and also increase in the volume concentration of the nanofluid, while the pressure drop increases only slightly. The increase in HTC is one of the most important aims for industry and researchers.

Keywords — Heat transfer; Al_2O_3 Nanoparticles, Laminar flow; Nanofluid; Nusselt number.

Nomenclature

Cp	Specific heat [J / kg / K]	**Greek Letters**	
C_B	Boltzmann's constant [1.38066×10^{-23} J / K]	¢	Particle concentration
		μ	Dynamic viscosity [N s / m^2]
dp	Particle diameter [m]	**ρ**	Density [kg / m^3]
k	Thermal conductivity [W m^{-1} K^{-1}]		
p	Pressure [Pa]	**Subscripts**	
Pr	Prandtl number	eff	effective
Q	Heat flux [W m^{-2}]	bf	Base fluid
Re	Reynolds number	f	frictional
T	Temperature [K]	fr	freezing
vb	Brownian velocity [m s^{-1}]	in	Inlet
V	Velocity [m s^{-1}]	out	Outlet
N	Avogadro number	nf	Nanofluid
M	Molecular weight	p	particle
df	Diameter of base fluid		
Nu	Nusselt number		

I. INTRODUCTION

Nowadays to enhance the heat transfer rate of fluid by eliminating the milli & micro sized particles which has been used in fluid and by introducing nano-sized particles to a fluid which has colloidal suspensions of nanoparticles in a base fluid is called a Nanofluid. The various types of nanoparticles used in nanofluid are typically made of metals, oxides, carbides or a carbon nanotube which help to improve the heat transfer rate, thermal conductivity of the fluid and also reduces the friction factor of the fluid by increasing the concentration of nanoparticles. In a present era the nanoparticles are taken as major role in technologies and economic challenge. There are some areas which are innovated like health, energy and industry.

In 1995 to increase thermal conductivity of fluid the nanoparticles as suspensions are introduced and conventionally to transfer the heat from fluids by "Choi", as suspended the coarse solid particles (milli & micro sized particles) to improve the fluid properties like heat transfer of thermal fluids. In the past, milli & micro sized particles are used in fluid which causes many problems and very difficult to overcome from excessive pressure drop, clogging of small channels, rapid sedimentation and the abrasive actions of the particles causes erosion of components, pipelines & working systems, to avoid this problems a new heat transfer fluid is required to maintain the system without any side effects. There are some important applications where Nano fluids are mainly utilized to reduce the emissions, global warming potential and greenhouse gas effect.

Nanofluids were proposed during 90's which consists of water and metal oxide nanoparticles like Al_2O_3, TiO_2, and CuO etc., of nano-sized dimensions. The mixture at certain

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proportions which promises better performance and improves the Thermal conductivity, Heat transfer rate, Nusselt number and small change in Pressure drop of the fluid with mixture of limited volume fraction of nanoparticles. The heat transfer performance of the fluids is increased by suspending nano- sized particles to the fluid. This is due to heat capacity, effective thermal conductivity and increase in surface area of the fluid.

Recently some studies were reported about using the nanofluid in solar collectors. "Toorajyousefi" et al., [4] has investigated the enhancement of heat transfer by using Al_2O_3 – water nanofluid and also increase in efficiency by varying the nanoparticles weight fraction of 0.2% & 0.4%, the size of nanoparticle was 15nm. "Yiminxuan&Qiong li" [5] has studied the thermal conductivity of nanofluid is measured by hot wire apparatus with suspended Cu nanophase powders. There are some factors such as volume fraction, dimensions, shapes and properties of the nanoparticles are discussed.

"Rong-yuanjou" [7] has investigated that the enhancement of heat transfers in a 2D rectangular enclosure by varying buoyancy parameter and volume fraction of nanofluid. As the result shows that by increasing these parameters which causes an increase in the average heat transfer co-efficient. "Gianpiero Colangelo" et al., [10] have investigated that shape and size of the flat solar thermal collector to avoid the sedimentation of nanofluid. According to that the modified flat plate solar collector has been considered due to proper flow of velocity and avoids the issues by nanofluid. After considered Al_2O_3 – water was chosen as heat transfer fluid which increases the heat transfer coefficient and thermal conductivity with increase in volume fraction. "Jung–yeuljung " [11] has investigated that the effect of volume fraction of nanoparticles to the convective heat transfer coefficient of Al_2O_3 nanofluid and the results are compared with distilled water which shows that there is increase in Nusselt number with increasing the Reynolds number in laminar flow regime. "Omidmahian" et al., [14] has studied that performance of different nanofluid on the flat plate solar collector which shows that Al_2O_3 nanofluid are highest heat transfer coefficient when compared to Cu-water, TiO_2-watre and SiO_2-water at volume fraction of 4% and nanoparticle size of 25nm. "Sandesh S .chaegule" [20] has investigated the performance of the wickless heated pipe solar collector using carbon nano tube (CNT) as nanofluid, and tested for different concentration and tilt angle. The optimized value of concentrated CNT nanofluid shows better performance is obtained and discussed. "Stephen U.S. choi" [26] has investigated the water based Al_2O_3 nanofluid over a uniformly heated circular tubes in the fully developed laminar flow regime and measured the pressure drop, convective heat transfer coefficient and thermal conductivity are discussed. "Pritamkumar das" [8] has been investigated that the effects of Cu nanoparticle by considering 100nm of particle size is mixed with base fluid (water) according to the volume fraction considered and simulated on CFD with a SPM on Laminar Forced Convection heat transfer in vertical tube. The result obtained that increase in average Nusselt number and heat transfer when compared to base fluid (water) & also increase in wall stress with increase in volume fraction and Reynolds number. "Mohammed benkhedda" [3] has investigated that laminar mixed convection flow of Ag-water nanoparticles in 3D horizontal concentric annulus on SPM approach. By varying Grashof number and volume fraction, the result obtained are Nusselt number, bulk temperature and wall temperature increases with increase in Grashof number and volume fraction.

"Mohammad kalteh" et al., [9] has studied that laminar forced convection flow in isothermally heated micro channel by using Cu-water nanofluid on Single phase mode (SPM) and multi phase model (MPM) approach. The results obtained are MPM shows high heat transfer than SPM with increases in Reynolds number and volume fraction as well as with decrease in nanoparticle diameter. "Yu feng" [17] has investigated that flow of convective nanofluid in parallel disks and studied the rate of heat transfer in terms of velocity profile, friction factor, temperature distribution and Nusselt number by employing the new K_{nf} model for nanofluid. The result shows that increase in heat transfer rate by using Al_2O_3-water nanofluid than the base fluid and also generation of entropy rate is lower than base fluid. "S. Zeinaliheris" [6] has studied that flow of Al_2O_3-water nanofluid in a square cross sectional duct under constant heat flux condition. As a results obtained that increase in heat transfer rate and Nusselt Number in nanofluid than the base fluid with increase in Reynolds number and volume fraction. "R. Deepak selvakumar" [25] has investigated that flow of forced convection nanofluid around circular cylinder on SPM and MPM approach, which show that increase in heat transfer rate, Nusselt number with increase in Reynolds number and volume fraction in MPM than the SPM approach. "Ghofranesekrani" [19] has investigated that flow of laminar and turbulent forced convection Al_2O_3-water nanofluid on SPM and MPM approach, which results shows that MPM has better heat transfer coefficient (HTC) with increase in volume fraction and Reynolds number than SPM approach which in turn the temperature dependent fluid properties result in a better prediction of the thermal field under the effect of a constant heat flux. "Vincenzo bianco" [16] has studied that flow of forced laminar convection of Al_2O_3-water nanofluid in an asymmetric heated channel of flat plate PV/T collector, which observed that increase in heat transfer and Nusselt number of nanofluid with increase in Reynolds number and volume fraction which also increase in change in pressure drop as well as decrease in entropy. "W.Y.Lai" [27] has investigated that the flow of Alumina nanofluid in a single 1.02mm stainless steel tube and studied the thermal performance under convective heat transfer conditions and also HTC in developing and fully developed regions by using water based Alumina nanofluid which observed that increase in HTC, change in pressure drop and Nusselt number.

IoNanofluids (INF) are basically a particular type of nanofluids (i.e., Suspensions of nanoparticles in conventional heat transfer fluids). The term INF is defined as the suspensions of nanoparticles (particles, tubes and rods) in ionic liquids and it is a new in multidisciplinary field such as Nano-science, nanotechnology, thermo fluid, chemical and mechanical engineering.INF is a new and innovative class of heat transfer fluid which exhibits fascinating thermo physical properties compared to their base ionic liquids. INF consists in suspending little amount of high conductivity nanoparticle in ionic liquids (IL's). By using INF in flat plate solar collector to access their feasibility and performance in heat transfer devices. "Carlos A. Nieto de castro" et al., [12, 13] has investigated that the INF and IL's thermo physical properties and estimating heat transfer by comparing the both. By using INF the results are better than the IL's. Thus INF is used as innovative heat transfer fluid. "Wael M. wel-maghlany" [18] has studied that by introducing new co-relation for heat transfer and friction factor in pipes subjected to constant heat flux considering [C4 mim] [NTf2] IL-based nanofluid. The performance and parameters are studied according to the Reynolds number and volume fractions are discussed.

In this project, A Steady state-laminar forced convective heat transfer has been simulated by CFD with a Single Phase Model (SPM), Multi Phase model & Diameter effects and also determined the effects of nanoparticles concentration and nanofluid flow rate through 3D rectangular duct under certain boundary condition (constant heat flux). The nanofluid contains Alumina nanoparticles of size 60nm diameter used for MPM which is mixed with base fluid (water) with volume fraction of $0\% \leq \phi \leq 5\%$ and Reynolds number (Re) ranges from $250 \leq Re \leq 1000$. In this study, steady state-laminar forced convective heat transfer and Friction factor of Al_2O_3 nanofluid in 3D rectangular duct flat solar plate collector under constant heat flux boundary condition was measured in laminar flow regime, which has not been studied yet.

II. MATHEMATICAL MODEL:

This study, the analysis of the steady state forced convection of laminar flow by using Al_2O_3-water based nanofluid within a 3D rectangular cross-section of dimensions (2m x 16cm x 1cm) 2m length, 16cm width and 1cm height are investigated. This project is about to study the effects of nanoparticles size and also rate of heat transfer in Alumina Nanofluid. The 3D rectangular channel is placed under an array of photovoltaic cells which helps to reduce the temperature and improves their performances. A schematic of the system is shown in **Figure1**.

The flow within the 3D rectangular channel is laminar and Reynolds number ranges from $250 \leq Re \leq 1000$, whereas the particles concentration varies from $0\% \leq \phi \leq 5\%$. In this project to study Single-Phase Model (SPM), Multi-Phase Model (MPM) and Diameter effects of nanoparticles. For MPM diameter of 60nm has been considered and for diameter effects, the particle diameter varies from $10nm \leq dp \leq 90nm$.

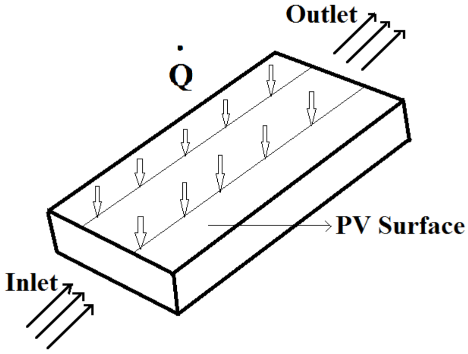

Figure1: A schematic of 3D rectangular duct system

There are certain boundary conditions are considered according to the heat transfer. The numerical problem consist the Conservation of mass (Eq. (1)), Conservation of momentum (Eqs. (2) - (3)) and Conservation of energy (Eq. (4)).

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

$$\boldsymbol{\rho}_m\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = -\frac{\partial p}{\partial x} + \left[\frac{\partial}{\partial x}\left(\boldsymbol{\mu}_m\frac{\partial u}{\partial x}\right) + \frac{\partial}{\partial y}\left(\boldsymbol{\mu}_m\frac{\partial u}{\partial y}\right)\right] \quad (2)$$

$$\boldsymbol{\rho}_m\left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right) = -\frac{\partial p}{\partial y} + \left[\frac{\partial}{\partial x}\left(\boldsymbol{\mu}_m\frac{\partial v}{\partial x}\right) + \frac{\partial}{\partial y}\left(\boldsymbol{\mu}_m\frac{\partial v}{\partial y}\right)\right] \quad (3)$$

$$\boldsymbol{\rho}_m C_p,\ m\left(u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y}\right) = \left[\frac{\partial}{\partial x}\left(k_m\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k_m\frac{\partial T}{\partial y}\right)\right] \quad (4)$$

To solve Eqs. (1) – (4), the following boundary conditions are imposed:

For $y = D$ and $0 \leq x \leq L$: $-k_m\frac{\partial T}{\partial y} = q$, $\frac{\partial T}{\partial x} = 0$, $u = 0$, $v = 0$

For $y = 0$ and $0 \leq x \leq L$: $\frac{\partial T}{\partial y} = 0$, $\frac{\partial T}{\partial x} = 0$, $u = 0$, $v = 0$

For $0 \leq y \leq D$ and $x = 0$: $u = u_{in}$, $v = 0$, $T = T_{in}$

For $0 \leq y \leq D$ and $x = L$: $\frac{\partial T}{\partial x} = 0$, $\frac{\partial T}{\partial y} = 0$, $\frac{\partial u}{\partial x} = 0$, $\frac{\partial v}{\partial y} = 0$

The Al_2O_3-water based nanofluid is considered as a homogeneous fluid with modified thermophysical properties for SPM, MPM and Diameter effects of nanoparticles. As suggested in Ref. [28]. Moraveji and Esmaeili compared the Nusselt number results of Multi-Phase Model (MPM) with the Single-Phase Model (SPM) which shows better performance in MPM approach than SPM.

The solutions obtained from SPM, MPM and Diameter effects by using different Thermophysical property equations for each model according to the study and some of the investigated literature review to get better results.

Thermophysical properties

The thermophysical properties of nanoparticle and the base fluid are listed in **Table 1**. It is one of the key steps related to nanofluids problems according to the boundary conditions of the different models. As demonstrated by Das et al. [30], nanofluids do not obey the rule of mixtures.

In the present analysis, the following basic equations are considered:

$$\boldsymbol{\rho}_{nf} = (1 - \phi)\,\boldsymbol{\rho}_{bf} + \phi\boldsymbol{\rho}_p \quad (5)$$

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$(\rho C_p)_{nf} = (1 - ¢)\,(C_p\rho)_{bf} + ¢\,(C_p\rho)_p$ (6)

$K_{nf} = (1 - ¢)\,k_{bf} + ¢k_p$ (7)

$\mu_{nf} = (1 - ¢)\,\mu_{bf} + ¢\mu_p$ (8)

Where,

- (C_p, ρ, μ, k) are Specific heat, Density, Dynamic viscosity and Thermal conductivity respectively.
- nf is nanofluid, bf is base fluid (water) and p is Alumina particle (Al_2O_3).

The equations from (5) – (8) represents Density (ρ), Specific heat (C_p), Thermal conductivity (k) and Dynamic viscosity (μ) of the nanofluid respectively. It is estimated on the basis of the mixture rule and by taking some of the literatures on experimental data and assumptions about thermal equilibrium between the nanoparticles and the base fluid, as suggested in Ref. [28-33].

i). For SPM approach:

In this study, we adopt a Single-Phase approach in which the nanofluid is assumed to be a homogeneous liquid with effective physical properties are considered for this approach for better results and performance.

$K_{eff} = k_{bf} * [(k_p + 2k_{bf}) + 2¢ * (k_p - k_{bf})/(k_p + 2k_{bf}) - ¢ * (k_p - k_{bf})]$ (9)

$\mu_{eff} = \mu_{bf}/(1 - ¢)^{2.5}$ (10)

$\rho_{eff} = (1 - ¢)\,\rho_{bf} + ¢\rho_p$ (11)

$(C_p)_{eff} = [(1-¢)(C_p\rho)_{bf} + ¢\,(C_p\rho)_p] / (\rho_{eff})$ (12)

The equations from (9) – (12) represents effective physical properties of thermal conductivity (k_{eff}), Dynamic viscosity (μ_{eff}), density (ρ) and Specific heat (C_p) of the nanofluid according to the assumptions and by taking some literature reviews [32-33].

ii).For MPM approach and Diameter effects:

In this study, we adopt a Multi-Phase approach and also for Diameter effects according to the mixture rule and feasibility, the different Dynamic viscosity (μ) and Thermal conductivity (k) of nanofluid are used.

$\frac{knf}{kbf} = 1 + 4.4 * Re_p^{0.4} * Pr^{0.66} * \left(\frac{T}{Tfr}\right)^{10} * \left(\frac{kp}{kbf}\right)^{0.03} * ¢^{0.66}$ (13)

$Re_p = (\rho_{bf} * v_b * d)/\mu_{bf}$ (14)

$v_b = [(2 * C_b * T)/(\pi * \mu_{bf} * d^2)]$ (15)

By substituting Eqn. (15) in Eqn. (14), we obtain the equation for Re_p

$Re_p = [(2*\rho_{bf} * Cb * T)/(\pi * \mu^2_{bf} * d^2)]$ (16)

Where,

- C_b is Boltzmann constant (1.38×10^{-23})
- T_{fr} is the freezing point of pure water (base fluid)
- Pr is the Prandtl number
- Re_p is the Reynolds number of the particle and rest of the parameters are defined above.

$\frac{\mu bf}{\mu nf} = 1 - 34.87\left(\frac{dp}{df}\right)^{-0.3} * ¢^{1.03}$ (17)

$d_f = 0.1 * (6M/N\pi\rho_f)^{1/3}$ (18)

Where,

- dp is diameter of the particles
- d_f is diameter of the base fluid
- M is the molecular weight of the base fluid
- N is Avogadro number and rest of the parameters are defined above.

The equation (13) and (17) represents the Thermal conductivity (k) and Dynamic viscosity (μ) of the nanofluid respectively, which are considered for this approach according to the reviews &assumptions

Table1: Thermo-physical properties of the base fluid (water) and nanoparticles (Al_2O_3).

Sl. No	Material	Thermal conductivity (k) [W/m-K]	Dynamic viscosity (μ) [kg/m-s]	Density (ρ) [kg/m3]	Heat capacitance (C_p) [J/kg-K]
01	Al_2O_3	40	-----	3970	765
02	H_2O	0.6	0.001003	997	4179

III. GRID INDEPENDENT AND CODE VALIDATION:

In order to obtain optimized numerical solution, Grid Independency check and comparison with related verified researches are necessary. The problem is solved for all grids which are shown in **Table2**, for the Nusselt number. It reveals that augmentation the optimized Grid number has selected according to the percentage difference of Nusselt number and also time taken for simulation. The chosen mesh is (15x300x30) for this problem and discussed further.

Table2: Grid Independency

Re	Grid	Nusselt Number	%difference
250	15x300x30	2.649965	3.1404
	25x400x45	2.735883	3.164
	35x500x60	2.838465	----
500	15x300x30	3.533462	3.032
	25x400x45	3.643949	5.5663
	35x500x60	3.858738	----
1000	15x300x30	4.612096	2.7636
	25x400x45	4.743181	0.6085
	35x500x60	4.77222	----

In this Simulation, three grids are analyzed inorder to ensure the accuracy as well as the consistency of the results. The analyzed grids are (15x300x30), (25x400x45) and (35x500x60) nodes respectively. To get better accuracy and solution time Grid (15x300x30) has been considered for further process. The Grid indicates (height,

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length, width) respectively. Furthermore, validation as to be carried out by using the Shah's correlation, as suggested by Utomo et al. [34]. The local Nusselt number can be calculated according to Shah's correlation,

Where, x* is equal to:

$$x^* = \frac{x}{Di*Re*Pr}$$

$$Nu_x = \begin{cases} 1.302 * x^{*(-1/3)} - 1 & x^* \leq 0.00005 \\ 1.302 * x^{*(-1/3)} - 0.5 & 0.00005 \leq x^* \leq 0.0015 \\ 4.364 + 8.68 * (1000x^*)^{-0.506} & \exp(-41x^*) * x^* > 0.0015 \end{cases}$$

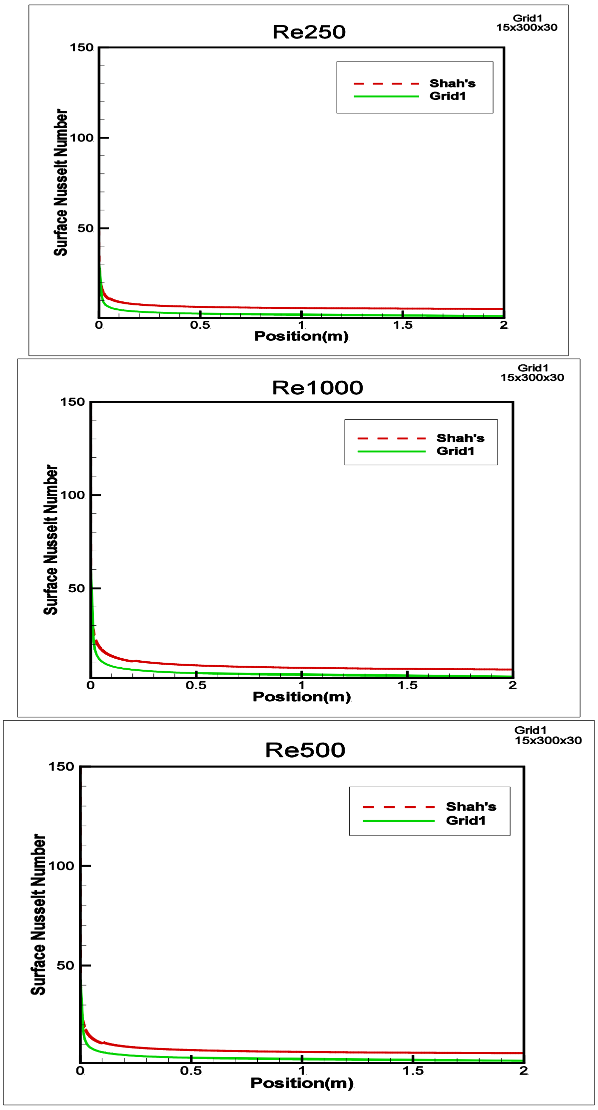

Figure2: Model validation for different Re values: 250; 500; 1000

IV. RESULTS & DISCUSSION:

After considered the Grid size for problem according to the Grid Independency and Code validation the problem has further solved by applied all boundary conditions for different approaches and obtained the results in the following graphs which as shown below:

i). Heat Transfer coefficient (h) vs. Length (L):

- The results obtained for different Reynolds number by varying the volume fraction and plotted the graph.
- As the volume fraction increases from 0% to 5%, the HTC increases along the length when compared to low volume fraction.

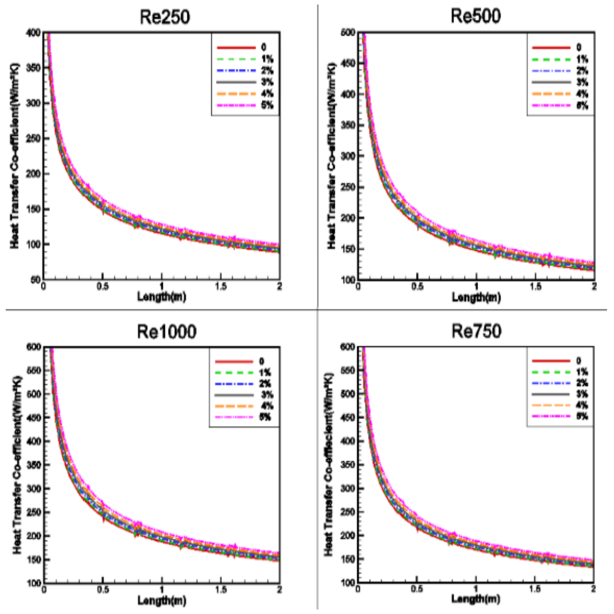

Figure3: HTC vs. Length for 250 ≤ Re ≤ 1000iii).

ii). HTC (h) vs. Volume fraction (Ø):

As the volume fraction increases from 0% to 5%, the HTC increases with increase in Reynolds number. The obtained results in terms of graphs are plotted and shown below.

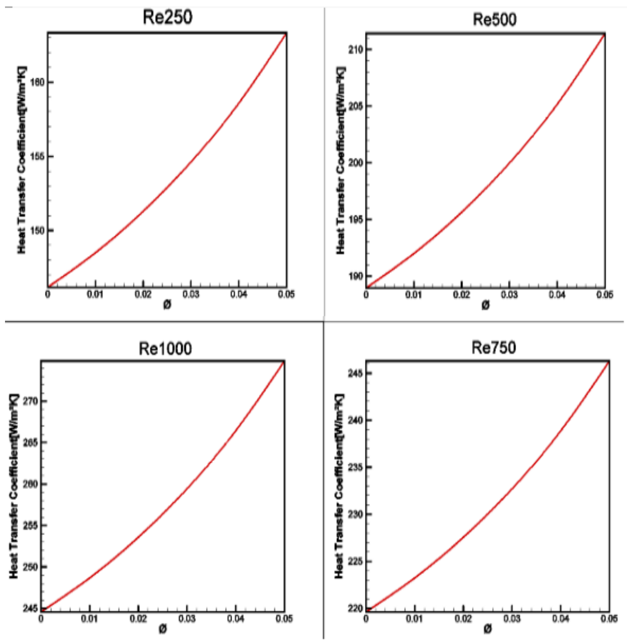

Figure4: HTC vs. volume fraction for 250 ≤ Re ≤ 1000

iii) Bulk Temperature (T_{bulk}) vs. Length (L):

The graph shows that, at 0% of volume fraction the change in temperature increases and at 5% the change in temperature over the length is decreases. Also we can see that the change in temperature is decreases with increases in Reynolds number.

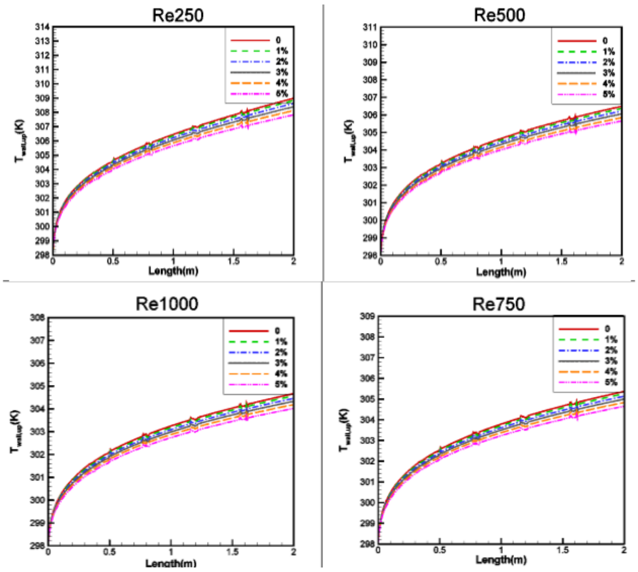

Figure5: Bulk temperature vs. Length for 250 ≤ Re ≤ 1000

iv). Upper wall temperature ($T_{wall,\,up}$) vs. Length (L):

- The upper wall of the rectangular duct as heat flux of 1000W/m2.
- The results show that the temperature increases gradually over the length for all volume fraction and also shows that the difference in temperature decreases with increase in Reynolds number at higher volume fraction.

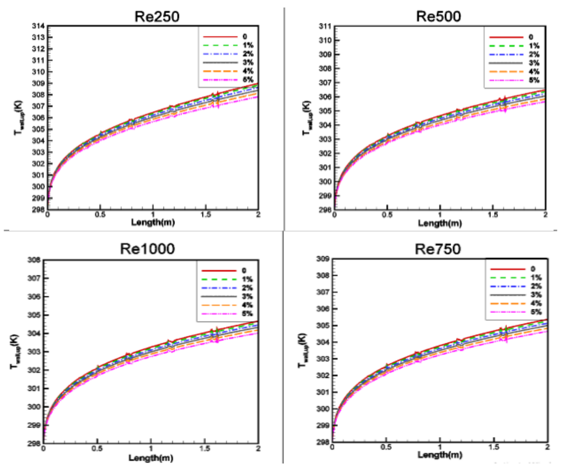

Fig6: Upper wall temp vs. Length for 250 ≤ Re ≤ 1000v). Lower wall temperature ($T_{wall,\,down}$) vs. Length (L):

- The lower wall and side walls are considered to be adiabatic condition where there is no transfer of heat (i.e., heat flux is 0 W/m2).
- The results show that there is a constant temperature for certain length and increases. It shows that at higher temperature and volume fraction the variation of temperature is less.
-

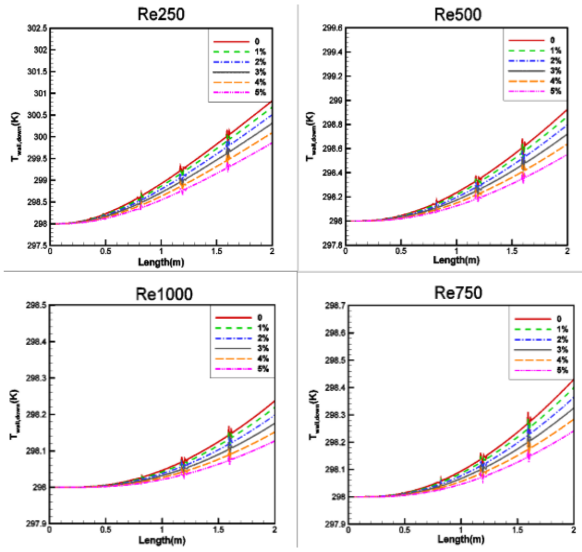

Figure7: Lower wall temp vs. Length for 250 ≤ Re ≤ 1000

vi). Mean Nusselt no. (Nu) vs. Particle diameter (d_p):

The results shows that, as the particle diameter increases the mean Nusselt number decreases and also indicates that the increase in Nusselt number at higher Reynolds number and decreased with lower Reynolds number.

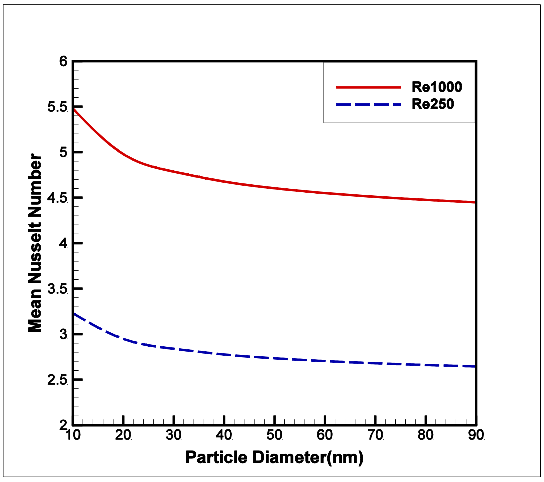

Figure8: Mean Nusselt number vs. Particle diameter for Re value: 250; 1000

Vii). Mean Nusselt Number (Nu) vs. volume fraction (Ø):

The result shows that increase in Nusselt number with increase of volume fraction and Reynolds number. At higher volume fraction (5%) and Reynolds number (1000) the mean Nusselt number increased.

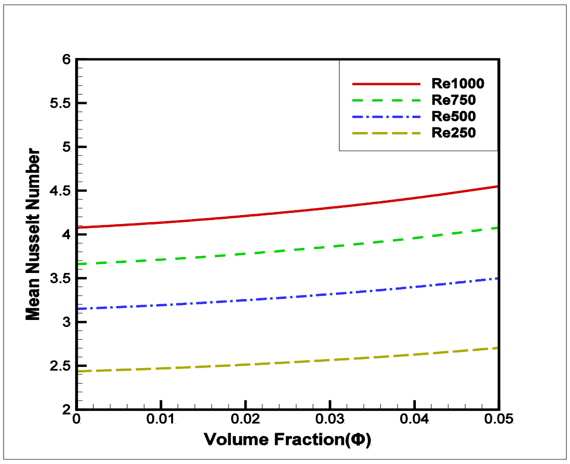

Fig9: Mean Nusselt number vs. Volume fraction for $250 \leq Re \leq 1000$

Viii). Mean Nusselt no. (Nu) vs. Volume fraction (Ø):

The result obtained for Single Phase approach (SPM) which shows that decrease in Mean Nusselt number with increase in volume fraction from 0% to 5%. Thus it shows that the mean Nusselt number is inversely proportional to the Reynolds number and Volume fraction.

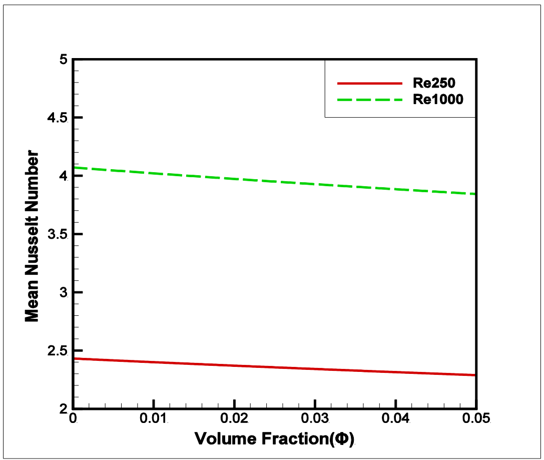

Figure10: Mean Nusselt number vs. Volume fraction for Re value: 250; 1000

ix). Pressure drop (Pr) vs. Volume fraction (Ø):

The result obtained for Multi-Phase approach (MPM) which shows that the change in pressure drop increases with increase in volume fraction and Reynolds number. Thus it shows that change in pressure drop is directly proportional to the volume fraction and Reynolds number.

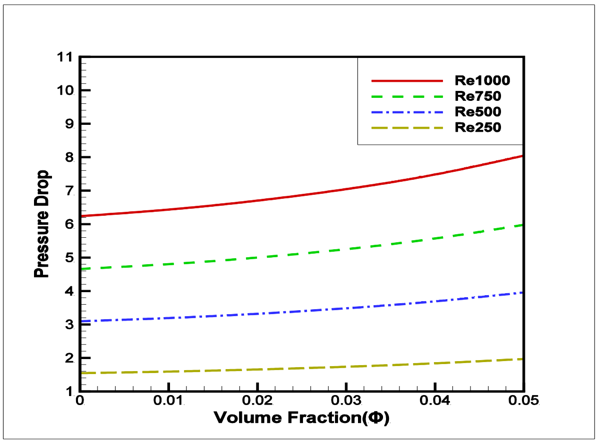

Fig11: Pressure Drop vs. Volume fraction for $250 \leq Re \leq 1000$

V. CONCLUSION

Forced convective heat transfer on a 3-DRectangular duct during a steady, laminar flow of Al_2O_3–H_2O Nanofluid has been numerically studied using a mixture model based Multi-Phase Modeling (MPM) approach. Effects of Reynolds number, particle volume fraction and diameter of nanoparticles on the heat transfer characteristics have been investigated under CWT and UHF. Salient feature of this work is that, apart from analyzing the effects of particle volume fraction and Reynolds number; a comparative study using three cases in which different models for effective viscosity and thermal conductivity and experimental data available in literature were used. Special attention has been given to the change in heat transfer enhancement when the modeling approach is changed from SPM to MPM. Findings of the numerical investigation can be summarized as follows:

- GhofraneSekrani [19] obtained the result by investigated Single-phase approach (SPM) and Multi-phase approach (MPM) by using Alumina Nanofluid and showed that MPM has better heat transfer coefficient with increase in volume fraction and Reynolds number than SPM under uniform heat flux (UHF).
- The increase in heat transfer and Nusselt number of Alumina Nanofluid with increase in Reynolds number and volume fraction which also increase in pressure drop as well as decrease in entropy in flat plate solar collector [16].
- It was observed that increase in heat transfer, pressure drop and Nusselt number of Alumina Nanofluid in a single stainless steel tube [27].
- The increase in heat transfer and Nusselt number of Alumina Nanofluid with increase in Reynolds number and volume fraction in a square cross section duct under UHF [6].

- By using Alumina Nanofluid and showed that MPM has better heat transfer coefficient with increase in volume fraction and Reynolds number than SPM in a circular cylinder under uniform heat flux (UHF) [25].
- By using new thermal conductivity for Alumina Nanofluid in a parallel disks which shows that increase in heat transfer rate as well as decrease in entropy rate than base fluid [17].
- The performance of different Nanofluid are investigated on a flat plate solar collector and obtained results shows that Alumina Nanofluid has better heat transfer coefficient than Cu – H_2O, TiO_2 – H_2O, and SiO_2 – H_2O [14].

Thus, we can conclude that the choice of modeling approach is very important while numerically analyzing Nanofluid flow and heat transfer. From the above graphs which shows that performance of HTC and Nusselt number on SPM & MPM approach for different Reynolds number, volume fraction and diameter of nanoparticles. Thus the results obtained have better performance in MPM than SPM by using Alumina Nanofluid on a 3D rectangular solar collector under CWT and UHF as boundary conditions.

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