STUDY OF MULTISPECIES NANO-PARTICLES TO ENHANCE THERMO-HYDRAULIC PERFORMANCE IN MICROCHANNELS

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STUDY OF MULTISPECIES NANO-PARTICLES TO ENHANCE THERMO-HYDRAULIC PERFORMANCE IN MICROCHANNELS

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I do hereby declare that this thesis is my original work and it has been written by me. I have duly acknowledged all the sources of information which have been used in the thesis. This thesis has not been submitted previously to any university for any degree.

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ABSTRACT

Nanofluid is the colloidal suspension of nano-sized solid particles of metals or metal oxides in base fluids such as water, ethylene glycol etc. When liquid is mixed with nanoparticles, it exhibits substantially higher thermal conductivity than those of the corresponding base fluids. The augmented thermal conductivity of nanofluids over the base fluids is considered one of the driving factors for enhanced heat transfer performance of nano-fluids. The forced convection heat transfer of nanofluid is investigated by numerous researchers over the last few years. Recently, multispecies nanofluids have been defined as a new class of nanofluids with possible applications in almost all fields of heat transfer. The idea of using multispecies nano-fluids is expected to improve the heat transfer characteristics of individual nanofluids and to beneficially combine different properties from metal oxides, metals etc.

The present research work is undertaken using the Computational Fluid Dynamics (CFD) to analysis and assess the high performing nanofluid for micro-channel applications. The study considers three metal oxide, two metal nano particles and their combinations in the base fluid i.e. desalinated water. The study is conducted for different Reynolds numbers and heat capacity. The performance of the nano-fluids is assessed based on the convective heat transfer coefficient, Nusselt number and pumping power requirement based on total pressure loss.

The extensive numerical analysis suggests that MgO-Water nano-fluid possesses excellent heat transfer performance over other combinations considered. Study also reveals that the metal oxides possess better cooling performance in terms of convective heat transfer coefficient as compared to metal nanofluids. Among multispecies nanofluids of Ag-MgO-Water, Al_2O_3-Cu-Water and CuO-Cu-Water, the Al_2O_3-Cu-Water nano fluid performed better providing highest Nusselt number which is approximately 6% over and above that provided by pure water.

Utilizing the data generated by parametric study for different nano-fluids, two combinations nanofluids are utilized for design of a compact heat exchanger with three different heat capacities i.e. 1 kW, 50kW and 100 kW. The hydraulic performance of this heat exchanger was compared in terms of pumping power requirements and it revealed that pumping power requirement increases nearly exponentially for higher Reynolds numbers.

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LIST OF SYMBOLS

C_P	Specific heat at constant pressure
k	Thermal conductivity
Q	Rate of heat transfer
$\dot{m}$	Mass flow rate
T_i	Inlet temperature
T_e	Exit temperature
q	Heat flux
h_x	Local heat transfer coefficient
T_s	Surface Temperature
T_b	Bulk temperature of a fluid
A_s	Surface area
R_e	Reynolds number
ρ	Density
v	Mean fluid velocity
μ	Dynamic viscosity
D_h	Hydraulic diameter
A_c	Cross-sectional area
N_u	Nusselt number
h	Convective heat transfer coefficient
P_r	Prandtl number
T_w	Wall temperature
Φ	Volume concentration
W	Pumping power per unit length of the pipe

Subscripts

i	Inlet value
o	Outlet value
p	nanoparticle
bf	Base fluid
nf	Nanofluid

CHAPTER 1

INTRODUCTION

1.1 Overview

The present study reports an investigation of the enhanced thermo-hydraulic performance in micro sized channels. In particular, a simulation model of micro-sized rectangular channel has been developed where nanofluids have been used as coolants. The combinations of several nanofluids and their volume concentrations are varied to observe bulk temperature, convective heat transfer coefficient, Nusselt number and total pressure drop.

1.2 Preface

With the demand for energy continues to grow globally, there is a need to make heat transfer equipment more energy efficient. Over the last decade, micromachining technology has been increasingly used for the development of highly efficient cooling devices [1] because of its undeniable advantages such as less coolant demands and small dimensions. Hence, the study of fluid flow and heat transfer in micro channels has attracted more attentions with broad applications in both engineering and medical fields. When convective heat transfer is considered, one of the most important parameters that affect the heat transfer is the thermal conductivity of the working fluid. Therefore, it is possible to develop heat transfer enhancement by using a working fluid with higher thermal conductivity. Thus, an innovative idea is to use suspended nano-sized solid particles in the fluid to improve heat transfer characteristics of the fluid [2].

1.2.1 Microchannel

Microchannel, as the name suggests, is a channel with a hydraulic diameter less than 1 millimeter. The concept of the microchannel was first proposed by Tuckerman and Pease [3] as early as 1981. Their publication titled “High Performance Heat Sinking for VLSI” is credited as the first study on microchannel heat transfer and since then the microchannel flow has been recognized as a high-performance heat removal tool.

Before proceeding with microchannel flow and heat transfer, it is appropriate to introduce a definition for the term “microchannel”. The scope of the term is among the topics of debate between researchers in the field. The following classification was used based on manufacturing techniques required to obtain various ranges of channel dimensions, “D”, being the smallest channel dimension [4]:

$1\ \mu m < D < 100\ \mu m$: Microchannels
$100\ \mu m < D < 1\ mm$: Minichannels
$1\ mm < D < 6\ mm$: Compact Passages
$6\ mm < D$: Conventional Passages

Subsequently, a different classification was adopted by Kandlikar and Grande [4] based on the rarefaction effect of gases in various ranges of channel dimensions, “D” being the smallest channel dimension:

$1\ \mu m < D < 10\ \mu m$: Transitional Microchannels
$10\ \mu m < D < 200\ \mu m$: Microchannels
$200\ \mu m < D < 3\ mm$: Minichannels
$3\ mm < D$: Conventional Passages

A simpler classification was proposed by Obot [5] based on the hydraulic diameter rather than the smallest channel dimension. Obot classified channels of hydraulic diameter below 1 mm ($D < 1$ mm) as microchannels, which was also adopted by many other researchers later on.

Microchannels offer advantages due to their high surface-to-volume ratio and their small volumes. The large surface-to-volume ratio leads to high rate of heat and mass transfer, making microdevices as more efficient tools for compact heat exchangers. Compared with the conventional heat exchanger, micro-channel heat exchanger reveals distinctive flow and heat transfer characteristics due to its structural and physical effects. It also enjoys advantages in the weight reduction and the improvement of the compactness of equipment. Thus, micro-channel heat exchanger plays an important role in heat transfer augmentation in microelectronics and micro-electromechanical systems (MEMS), miniaturized chemical reactors, combustors, aerospace, and biomedical systems.

1.2.2 Nanofluids

Despite considerable efforts are made to improve the rate of heat transfer by usage of extended surfaces, mini-channels and micro-channels, further enhancement in heating and cooling rate is always in demand. Since solid materials possess higher thermal conductivities, many studies have been carried out on thermal properties of conventional heat transfer fluids with suspension of solid particles [2]. As a result, nanofluid, a new class of heat transfer fluid with nanometer sized particles suspended in the base fluid has been first coined by Choi in 1995 at Argonne National Laboratory of USA [6]. Suspensions of nanoparticles are possible because the interaction of the particle surface with the solvent is strong enough to overcome differences in density, which usually result in a material either sinking or floating in a liquid. They have a very high surface area to volume ratio which provides a tremendous driving force for diffusion, especially at elevated temperatures. Due to their large surface area to volume ratio, the incipient melting temperature of nanoparticles is also reduced. The physical properties of some nanoparticles are mentioned below:

Table 1.1: Properties of nanoparticles [7].

Nanoparticles	**Density (kg/m^3)**	**Specific Heat, C_p (J/Kg k)**	**Thermal Conductivity, k (W/m k)**
Al_2O_3	3890	880	35
SiC	3216	610	15
Fe_3O_4	5810	670	161
Ag	10490	235	429
MgO	3580	874	55
CuO	6500	533	17.65
Cu	8940	385	401
SiO_2	2200	740	1.38
TiO_2	4175	692	8.4

Nanofluid as a new, innovative class of heat transfer fluid represents a rapidly emerging field where nanoscale science and thermal engineering meet. Nanofluids, engineered by dispersing nanometer-sized solid particles in conventional heat transfer fluids have been found to possess superior thermal performance compared to their base fluids. Hence, there is a need for fundamental understanding of the heat transfer behavior of nanofluids in order to explore their potential benefits and applications. Various potential benefits of the application of nanofluids include: improved heat transfer, heat transfer system size reduction, minimal clogging and micro channel cooling.

1.2.3 Thermo-hydraulic performance analysis

In the absence of any work interactions (such as electric resistance heating), the conservation of energy equation for the steady flow of a fluid in a tube can be expressed as [8]

$$Q = \dot{m}C_p(T_e - T_i) \quad 1.1$$

Where, T_i and T_e are the mean fluid temperatures at the inlet and exit of the tube, respectively, and Q is the rate of heat transfer to or from the fluid. The temperature of a fluid flowing in a tube remains constant in the absence of any energy interactions through the wall of the tube. For the present study, the thermal conditions at the surface have been approximated with reasonable accuracy to be constant surface heat flux. Surface heat flux is expressed as

$$q = h_x(T_s - T_b) \quad 1.2$$

Where, h_x is the local heat transfer coefficient and T_s and T_b are the surface and the mean fluid temperature at that location. The mean fluid temperature or bulk temperature T_b of a fluid flowing in a tube changes during heating or cooling. In the case of $q = constant$, the rate of heat transfer can also be expressed as

$$Q = qA_s = \dot{m}C_p(T_e - T_i) \quad 1.3$$

Then the mean fluid temperature at the tube exit becomes

$$T_e = T_i + \frac{qA_s}{\dot{m}C_p} \quad 1.4$$

Thus, the mean fluid temperature at the exit increases linearly in the flow direction in the case of constant surface heat flux, since the surface area increases linearly in the flow direction (A_s is equal to the perimeter, which is constant, times the tube length).

For the determination of convection heat transfer coefficient h, some dimensionless numbers will be introduced. The Reynolds number, Re, is interpreted as a flow characteristic proportional to the ratio of flow momentum rate to viscous force for a specified geometry. Re is also called as flow modulus and defined for internal flow as

$$R_e = \frac{\rho v D_h}{\mu} \quad 1.5$$

Where, is $D_h$the characteristic length of the channel, ρ is the density, μ is the viscosity of fluid and v is the mean fluid velocity. D_h is defined for noncircular ducts and is expressed as [8]:

$$D_h = \frac{4A_c}{p} = \frac{4 \times net\ free\ flow\ area}{wetted\ perimeter} \qquad 1.6$$

At large Reynolds numbers, the inertia forces, which are proportional to the density and the velocity of the fluid, are large relative to the viscous forces, and thus the viscous forces cannot prevent the random and rapid fluctuations of the fluid. At small Reynolds numbers, however, the viscous forces are large enough to overcome the inertia forces and to keep the fluid streamlined. Thus, the flow is turbulent in the first case and laminar in the second.

The Nusselt number, N_u is one of the dimensionless representations of the heat transfer coefficient. Nu is a ratio of the convective temperature gradient to the conductive temperature gradient. It represents the enhancement of heat transfer through a fluid layer as a result of convection relative to conduction across the same fluid layer [8]. The larger the Nusselt number, the more effective the convective heat transfer. It is defined as

$$N_u = \frac{hD_h}{k} \qquad 1.7$$

Here, k is the thermal conductivity of the fluid and h is the convective heat transfer coefficient. A Nusselt number, $N_u = 1$ for a fluid layer represents heat transfer across the layer by pure conduction.

The Prandtl number, P_r, is the fluid property modulus representing the ratio of momentum diffusivity to thermal diffusivity of the fluid. Physically, it describes the relative thickness of the velocity and the thermal boundary layer. P_r is expressed as:

$$P_r = \frac{\mu C_p}{k} \qquad 1.8$$

Pressure drop is an important part of hydraulic performance analysis as it indicates the required pumping power. It is difficult to predict the pressure drop in a non-standard heat exchanger. Therefore, in this work, pressure drop will be determined using the measured difference in pressure at the inlet and outlet of the heat exchanger.

1.2 Applications

Microchannel heat exchangers have applications in several important and diverse fields including: nuclear reactors; aerospace; automotive; bioengineering; cooling of gas turbine blades, power and process industries; refrigeration and air conditioning; infrared detectors and powerful laser mirrors and superconductors; microelectronics; and thermal control of film deposition. The advantages of microchannel heat exchangers include high volumetric heat flux, compactness for space-critical applications, robust design, effective flow distribution, and modest pressure drops. In particular, these can be extensively used in aviation- during regenerative cooling of semi- cryogenic rocket engine thrust chambers, in utility system due to higher heat transfer coefficient, higher heat recovery from exhaust gas of boiler, heat exchanger and reactor, in oil-gas industry during gas processing and refining, in petrochemicals during distillation, fractionating of component into variety of products where materials with variable ranges of temperature are necessary.

1.3 Objectives of the Present Work

As it is evident from the diversity of application areas, the study heat transfer in microchannels is very important for the technology of today and the near future, as developments are following the trend of miniaturization and energy efficient systems in all fields. Although various techniques have been applied so far, research is still going on to enhance the heat transfer rate. Literature shows a good number of studies on microchannels and microchannels heat sinks, but there is limited research related to the performance study of microchannel heat exchangers using nanofluids and their various combinations.

Considering these aspects, the present work is undertaken to study the following:

- To design a compact narrow microchannel heat exchanger proposed to be used in aircraft cooling requirements.

- To undertake a CFD analysis of microchannel heat exchanger with base fluid to understand its hydrodynamic and thermal performance.

- The effects of nanoparticle sizes (20 to 200nm) and its different compositions (Al_2O_3, SiO_2, MgO, CuO) as additive to base fluid will be analyzed using CFD technique for thermal (Bulk temperature, Heat transfer coefficient, Nusselt number) and hydraulic performance (Total pressure drop) for different Reynold's number (200 to 20000).

1.4 Outline of the Report

Chapter 1 represents introduction of project work including definition of micro channel and its application as heat exchanger, definition of nanofluids and its use as additive to coolant, objectives and applications of the present study and lastly the outline of the report.

Chapter 2 is devoted on the literature survey on experimental study of fluid flow and heat transfer in micro channels, numerical study of fluid flow and heat transfer in micro channels and nanofluids as heat enhancer.

Chapter 3 represents the validation and grid independence test details.

Chapter 4 deals with the computational modeling and its associated correlations and calculations of the physical properties due to suspension of nanoparticles in the base fluid. Also, the specification of the problem, simulation of single-phase fluid flow in a rectangular micro channel mentioning different flow parameters are included in this chapter. It also represents the effects of multispecies nanofluids on thermo-hydraulic performance of heat exchanger.

Chapter 5 deals with overall conclusion and future scope of the work.

CHAPTER 2

LITERATURE REVIEW

Some experimental and computational work on micro channel heat exchanger and effects of nanofluids have been done in the last decades. Due to a sustainable imbalance between the demand and reserves for energy, both the industrial and academic people have taken interest in this area. The following is a review arranged according to similarity to the work done in this thesis. In this literature review emphasis is directed on:

- ✓ Experimental study of fluid flow and heat transfer in micro channels
- ✓ Numerical study of fluid flow and heat transfer in micro channels
- ✓ Experimental study of nanofluids
- ✓ Numerical study of nanofluids

2.1 Experimental Study of Fluid Flow and Heat Transfer in Microchannels

With the development of micro fabrication technology, microfluidic systems (MFD) have been increasingly used in different scientific and industrial applications. Microchannels are one of the essential devices for microfluidic systems. Therefore, the importance of convective transport phenomena in microchannels and microchannel structures has increased dramatically. In recent years, a number of researchers have reported the heat transfer and pressure drop data for laminar and turbulent liquid or gas flow in microchannels.

The concept of micro channel heat sink at first was proposed by Tuckerman and Pease [3]. They demonstrated that the micro channel heat sinks, consisting of micro rectangular flow passages, have a higher heat transfer coefficient in laminar flow regime than that in turbulent flow through conventional-sized devices. They concluded that the heat transfer can be enhanced by reducing the channel height down to micro scale. This pioneering work initiated other studies and many researchers compared their numerical or analytical studies with Tuckerman and Pease.

Zhou et al. [9] investigated experimentally and numerically the flow and heat transfer characteristic of liquid laminar flow in microtube. They used the smooth fused silica and rough stainless steel microtubes with the hydraulic diameters of 50–100 mm and 373–1570 mm, respectively. The Reynolds numbers was ranged from 20 to 2400. The results indicated that the conventional friction prediction is valid for water flow through microtube (MT) with a relative surface roughness less than1.5%. The experimental results showed that the Nusselt number along the axial direction does not accord with the conventional results especially when the Reynolds number is low and the relative tube wall thickness is high.

Yang and Lin [10] investigated experimentally the forced convective heat transfer performance of water flow through six micro stainless-steel tubes with inner diameters ranging from 123 to 962 μm. The results indicated that the conventional heat transfer correlations for laminar and turbulent flows can be well applied for predicting the fully developed heat transfer performance in the micro tube. They observed that the heat loss is less than 4% and 1% of the total heat input for 123 μm and 962 μm tubes, respectively. At the lowest Reynolds number in the 123 μm tube, a 4% heat loss cause 25% heat transfer coefficient overestimate.

Sara et al. [1] investigated experimentally the laminar convective mass transfer and friction factor in MT with diameter of 0.20 mm and L/D in the range of 100 to 500. The results reveal that the friction factor was in a very good agreement with the Hagen–Poiseuille theory in the laminar regime and the conventional correlations can be used to determine the friction factor for microtube with ID $\geq$ 0.20 mm. For mass transfer, the Sherwood number for MT was smaller than those obtained from the correlations developed for macrotubes.

Sharma et al. [11] analyzed experimentally for the fluid flow and heat transfer with Reynolds number varying from 350 to 2200 for micro-channels having hydraulic diameter 0.61 mm and depth 2.5 mm. When mass flow rate decreases, friction factor also decreases due to decrease in velocity of flowing fluid. The pressure drop increases with increase in velocity of fluid, thereby increases heat flux removal rate. Initially fanning friction factor decreases as there is increase in mass flow rate. But after certain value of mass flow rate it starts increasing. The heat transfer rate increases with increase in mass flow rate. And in turn pressure loss is converted into heat which is carried by the water which was used as cooling agent.

2.2 Numerical Study of Fluid Flow and Heat Transfer in Microchannels

To design an effective microchannel heat sink, fundamental understanding of the characteristics of the heat transfer and fluid flow in microchannel are necessary. At the early stages the designs and relations of macroscale fluid flow and heat transfer were employed. The strength of numerical simulations is the possibility to investigate small details that are impossible to observe in experiments.

Xiong and Chung [12] proposed a novel bottom-up approach to generate a three-dimensional microtube surface with random roughness. A computational fluid dynamics solver is used to isolate the roughness effect and solve the three-dimensional N–S equations for the water flow through the generated rough microtubes with diameter D = 50 μm and length L = 100 μm. No-slip, constant heat flux and periodic boundary conditions are applied to achieve the characteristics of fully developed flow. It is found that the wall roughness strongly affects the velocity near the wall and it almost has no effect on the flow at the center. The temperature has large values at the peaks and small values at the valleys of the microtube surface. However, the roughness has almost no effect on the averaged Nusselt number, Nu because the effects of peaks and valleys counteract each other for the whole microtube. The local Nusselt numbers along the microtube randomly scatter from the theoretical value with a deviation less than 2%. The model has a potential to be used for direct simulations of three-dimensional surface roughness effect on the slip flow.

Liang Xia, Yue Chan [13] reports an investigation of the enhanced heat transfer effect in micro sized channels. In particular, a simulation model has been established using CFD to investigate such micro effect. Here, hot fluid flow is squeezed into the micro channel and the heat is gradually transferred into the cold surface boundaries of the channel having rectangular cross-sections. The area of inlet is varied to investigate and identify the enhanced heat absorption arising from the size effect and it is found that heat transfer rate per unit effective heat transfer area is greater if the inlet area becomes smaller. Lower pressure drop can also be deduced if smaller radii channels are considered. Moreover, a new design of heat exchanger is proposed based on these extra micro-effects.

Emran and Islam [14] performed a numerical simulation to investigate the flow dynamics and heat transfer characteristics in a microchannel heat sink. A unit cell containing a single microchannel of a width of 231 μm and a depth of 713 μm with surrounding solid is used for simulation. Water at 15 °C is used as the cooling liquid with Reynolds number ranging from 225 to 1450 and a constant heat flux is applied at the bottom of the heat sink. A commercial CFD code employing finite element method is used for the numerical simulation. The pressure drop in the microchannel obtained from the simulation agrees well with experimental results. The highest temperature is found at the bottom of the heat sink immediately below the channel outlet and the lowest is at the channel inlet. The heat flux is high near the channel inlet due to thin thermal boundary layer in the developing region and varies around the channel periphery with lower values at the corners. These findings demonstrate that the conventional Navier–Stokes and energy equations can adequately predict the fluid flow and heat transfer characteristics of microchannel heat sinks.

Hetsroni et al. [15] has verified the capacity of conventional theory to predict the hydrodynamic characteristics of laminar Newtonian incompressible flows in micro channels in the range of hydraulic diameter from 60 to 2000 μm . They have compared their results with the data available in open literature. The theoretical models were subdivided in two groups depending on the degree of correctness of the assumptions. The first group includes the simplest one-dimensional models assuming uniform heat flux, constant heat transfer coefficient, etc. The comparison of these models with experiment shows significant discrepancy between the measurements and the theoretical predictions The second group is based on numerical solution of full Navier–Stokes and energy equations, which account the real geometry of the micro-channel, presence of axial conduction in the fluid and wall, energy dissipation, non-adiabatic thermal boundary condition at the inlet and outlet of the heat sink, dependence of physical properties of fluid on temperature, etc. These models demonstrate a fairly well correlation with the available experimental data.

Khanafer et al. [16] has investigated heat transfer enhancement in a two-dimensional rectangular enclosure utilizing nanofluids. The material used is water/ copper. The developed transport equations were solved numerically using the finite-volume approach along with the alternating direct implicit procedure. The effect of suspended ultrafine metallic nanoparticles on the fluid flow and heat transfer processes within the enclosure was analyzed. The heat transfer correlation of the average Nusselt number for various Grashof numbers and volume fractions was also presented.

2.3 Experimental Study of Nanofluids

Asirvatham et al. [17] conducted an experimental study of steady state convective heat transfer of de-ionized water with a low volume fraction (0.003% by volume) of copper oxide nanoparticles dispersed to form a nanofluid that flows through a copper tube. The effect of mass flow rate ranging from (0.0113 kg/s to 0.0139 kg/s) and the effect of inlet temperatures at 10º C and 17º C on the heat transfer coefficient are studied on the entry region under laminar flow condition. The results have shown 8% enhancement of the convective heat transfer coefficient of the nanofluid even with a low volume concentration of CuO nanoparticles. Likewise, the improvement in convective heat transfer is also observed at higher temperature and high heat flux.

Salman et al. [18] numerically investigated laminar convective heat transfer in a two-dimensional microtube with 50 μm diameter and 250 μm length with constant heat flux. Different types of nanofluids Al_2O_3, CuO, SiO_2 and ZnO, with different nanoparticle size 25, 45, 65 and 80 nm, and different volume fractions ranged from 1% to 4% using ethylene glycol as a base fluid were used. The results have shown that SiO_2–EG nanofluid has the highest Nusselt number, followed by ZnO–EG, CuO–EG, Al_2O_3–EG, and lastly pure EG. The Nusselt number for all cases increases with the volume fraction but it decreases with the rise in the diameter of nanoparticles.

Salman et al. [19] considered forced convective laminar flow of different types of nanofluids such as Al_2O_3 and SiO_2, with nanoparticle size 30 nm, and different volume fractions ranged from 0.5% to 1% along with water as base fluids for experimental investigation. The Microtube with 0.01 cm diameter and 20 cm length was used in this investigation. This investigation covered Reynolds number in the range of 90 to 800. The results concluded that SiO_2-water nanofluid has the highest Nusselt number, followed by Al_2O_3-water, and lastly pure water. The maximum heat transfer enhancement was about 22% when using the nanofluids and the experimental results agreed well with the conventional theory.

Hee Chon et al. [20] reported an experimental correlation for the thermal conductivity of Al_2O_3 nanofluids as a function of nanoparticle size ranging from 11 nm to 150 nm nominal diameters over a wide range of temperature from 21 to 71 °C. It was experimentally validated that the Brownian motion of nanoparticles constitutes a key mechanism of the thermal conductivity enhancement with increasing temperature and decreasing nanoparticle sizes.

Calvin H. Li and G. P. Peterson [21] conducted an experimental investigation to examine the effects of variations in the temperature and volume fraction on the steady-state effective thermal conductivity of two different nanoparticle suspensions. Copper and aluminum oxide nanoparticles with area weighted diameters of 29 and 36 nm, respectively, were blended with distilled water at 2%, 4%, 6%, and 10% volume fractions and the resulting suspensions were evaluated at temperatures ranging from 27.5 to 34.7 °C. The results indicate that the nanoparticle material, diameter, volume fraction, and bulk temperature, all have a significant impact on the effective thermal conductivity of these suspensions. The 6% volume fraction of CuO nanoparticle/distilled water suspension resulted in an increase in the effective thermal conductivity of 1.52 times that of pure distilled water and the 10% Al_2O_3 nanoparticle/distilled water suspension increased the effective thermal conductivity by a factor of 1.3, at a temperature of 34 °C.

Divya and Kumar [22] conducted an experimental study of performance of Al_2O_3 Nano fluid in a heat exchanger in his work. Pure water was used in a heat exchanger and its performance was studied. Al_2O_3 nano particles were mixed in water by 0.1 % & 0.2 % volume concentration and the performance was studied. The result said that nano fluids can be used to improve heat transfer and energy efficiency in many thermal control systems and heat exchangers.

Tijerina et al. [23] investigated the laminar forced convection for the flow of nanofluids in conventional straight tube (L = 5.34 m, dt = 10 mm) and straight microtube (L = 0.3 m, dt = 0.5 mm) under the constant temperature and constant heat flux conditions, separately. A wide range of the process parameters has been studied by varying three different type of base fluids including water, ethylene glycol and turbine oil with five different type of nanoparticles viz. Al_2O_3, TiO_2, CuO, SiO_2 and ZnO. Six different combinations of geometries, base fluids and nanoparticle concentrations are considered in the present study. In addition to the single-phase model (SPH), the single-phase dispersion model (SPD) has also been used for effectiveness of the computed results. The results showed that Nusselt number considerably enhanced up to 16% at volume fraction 4% and Re = 950 with the increase in nanoparticle concentrations.

Sharma et al. [24] examined the viscometric properties of various hybrid nanofluids of different concentrations (0.5~ 3 vol. %) at a range of temperature (25º C ~ 50º C) and different shear rates. All the way through this study it is established that temperature and volume concentration have substantial upshot over viscosity of hybrid nanofluids. Shear-thinning activities is observed in all the test samples at different temperatures. It is implicit from the experimental results that at lesser volume concentration to upto 3% the shear thinning of hybrid nanofluid is due to Newtonian character. It is also observed that the viscosity ratio of hybrid nanofluids decline with rise in temperature. This turns out due to the deterioration of intermolecular communications and adhesion forces among the molecules. The experimental outcomes have been compared with predictions by the existing theoretical models. It has been found that the experimental results are notably greater than those predicted by different models given by different investigators. Drop off in viscosity with rise in temperature is shown potential for using nanofluids as superior coolants in thermal managing applications.

Abbasian Arani [25] experimentally investigated the convection heat transfer characteristics in fully developed turbulent flow of T_iO_2–water nanofluid. The effect of mean diameter of nanoparticles on the convective heat transfer and pressure drop studied at nanoparticle volume concentration from 0.01 to 0.02 by volume. The experimental apparatus is a horizontal double tube counter-flow heat exchanger. The nanoparticles of T_iO_2 with diameters of 10, 20, 30 and 50 nm dispersed in distilled water as base fluid. The results indicated higher Nusselt number for all nanofluids compared to the base fluid. It is seen that the Nusselt number does not increase by decreasing the diameter of nanoparticles. In this study both Nusselt number and pressure drop were considered in definition of thermal performance factor. The results show that nanofluid with 20 nm particle size diameter has the highest thermal performance factor in the range of Reynolds number and volume concentrations.

Suresh et al. [26] synthesized Al_2O_3–Cu hybrid particles by hydrogen reduction technique from the powder mixture of Al_2O_3 and CuO in 90:10 weight proportions obtained from a chemical route synthesis. Al_2O_3–Cu/water hybrid nanofluids with volume concentrations from 0.1% to 2% were then prepared by dispersing the synthesized nanocomposites powder in de-ionized water. The experimental results have shown that both thermal conductivity and viscosity of the prepared hybrid nanofluids increase with the nanoparticles volume concentration. The thermal conductivity and viscosity of nanofluids have been measured and it has been found that the viscosity increase is substantially higher than the increase in thermal conductivity. The experimental measurement of thermal conductivity showed a maximum enhancement of 12.11% for a volume concentration of 2%. The experimental results have been compared with the classical theoretical models available in literature.

Takabi and Shokouhmand [27] numerically analyzed forced convection of a turbulent flow of pure water, Al_2O_3/water nanofluid and Al_2O_3–Cu/water hybrid through a uniformly heated circular tube. They also examined the effects of these three fluids as the working fluids, with wide range of Reynolds number $10000 \leq Re \leq 100000$ and also the volume concentration varying between 0 to 2% on heat transfer and hydrodynamic performance. The finite volume discretization method is employed to solve the set of the governing equations. The results indicated that employing hybrid nanofluid improves the heat transfer rate with respect to pure water and nanofluid, yet it reveals an adverse effect on friction factor and appears severely outweighed by pressure drop penalty. However, the average increase of the Nusselt number (when compared to pure water) in Al_2O_3–Cu/ water hybrid nanofluid is 32.07% and the amount for the average increase of friction factor is 13.76%.

2.4 Numerical Study of Nanofluids

Bianco et al. [28] investigated a model for developing laminar forced convection flow of a water– Al_2O_3 nanofluid in a circular tube, provided with a constant and uniform heat flux at the wall, is numerically investigated. A single and two-phase model (discrete particles model) is employed with either constant or temperature dependent properties. The investigation is accomplished for size particles equal to 100 nm. The maximum difference in the average heat transfer coefficient between single and two phase models is about 11%. Convective heat transfer coefficient for nanofluids is greater than that of the base liquid. Heat transfer enhancement increases with the particle volume concentration, but it is accompanied by increasing wall shear stress values. Higher heat transfer coefficients and lower shear stresses are detected in the case of temperature dependent models. The heat transfer always improves, as Reynolds number increases, but it is accompanied by an increase of shear stress too.

Ghatage et al. [2] have developed the CFD models to predict the convective heat transfer coefficient of Al_2O_3/ water nanofluid with different volume fraction in circular tube by using experimental data. The volume- averaged continuity, momentum and energy equations were numerically solved by using ANSYS Fluent version 13.0. They suggested that at fixed Reynolds number 6000, the heat transfer coefficient for 1% Al_2O_3/ water nanofluid increases by 1.55 times over the base fluid which indicates that the use of nanofluid can significantly increase the heat transfer capabilities of cooling systems.

Akbarinia et al. [29] studied a forced convection Al_2O_3-water nanofluid flows in two-dimensional rectangular microchannels to investigate heat transfer enhancement due to addition of the nanoparticles to the base fluid especially in microchannels at low Reynolds number. Three different cases are examined to evaluate proportion impact of increasing nanoparticles volume fraction and the inlet velocity on heat transfer enhancement. Two-dimensional Navier-Stokes and energy equations accompanied with the slip velocity and the jump temperature boundary conditions expressions have been discretized using the Finite Volume Method. The calculated results show good agreement with the previous numerical and analytical data. It is found that at a given Reynolds number, the major enhancement in the Nusselt number is not due to increasing the nanoparticle concentrations but it is due to the increasing the inlet velocity to reach a constant Re. Constant Reynolds number studies of nanofluids are not sufficient approach to evaluate the heat transfer and the skin friction factor due to the nanofluids usage.

Zainal et al. [30] performed a CFD simulation analysis about enhancement of turbulent flow heat transfer in a horizontal circular pipe by convenient software where FLUENT was used to predict the heat transfer coefficient and Nusselt number for forced convection heat transfer of Ag/HEG - water nanofluid. The range of Reynolds number selected were 20000 and 40000 in a horizontal straight tube of diameter 0.01m with heat flux of 1000 W/m^2. The volume fraction of nanoparticle considered were 0.1%, 0.2%, 0.3%, 0.5%, 0.7% and 0.9%. This problem also considered 100x30 meshing in order for y^+ to approach 1 to get more accurate results. The results show that the heat transfer coefficient and Nusselt number were decreasing with increasing of volume fraction. Finally, the results were compared with the theoretical values obtained from Dittus-Boelter Equation by using ANSYS FLUENT which have shown similar results.

Salman and Abd Al Saheb [31] studied two-dimensional turbulent flow of different nanofluids and ribs configuration in a circular tube numerically using FLUENT. Two samples of CuO and ZnO nanoparticles with 2% volume concentration and 40 nm as nanoparticle diameter combined with trapezoidal ribs with aspect ratio of p/d=5.72 in a constant tube surface heat flux was conducted for simulation. The results showed that Nusselt number for all cases rises with Reynolds number and volume fraction of nanofluid, likewise the results also reveal that ZnO with volume fractions of 2% in trapezoidal ribs offered highest Nusselt number at Reynolds number of Re= 30000.

Nimmagadda and Venkatasubbaiah [32] studied laminar forced convection flow of nanofluids in a wide rectangular micro-channel. The flow and heat transfer characteristics of Aluminum oxide, silver and hybrid (Al_2O_3 + Ag) nanofluids have been investigated in a micro-channel. A two-dimensional conjugate heat transfer homogeneous phase model has been developed and results are reported for different Reynolds numbers. The governing equations are solved by Simplified Marker and Cell (SMAC) algorithm on non-staggered grid using finite volume method. The effects of Reynolds number, pure nanoparticles volume concentration, hybrid nanoparticles volume concentrations and nanoparticles size on the flow and heat transfer characteristics are reported. The results show that the average convective heat transfer coefficient increases with increase in nanoparticles volume concentration and Reynolds number. The nanofluids obtained by dispersing nanoparticles such as Al_2O_3, Ag and hybrid (Al_2O_3 + Ag) in the base fluid shows a significant enhancement of average convective heat transfer coefficient in comparison with pure water. It is also observed that that 3% volume of hybrid nanofluid shows higher average convective heat transfer coefficient than that of pure water, pure oxide (Al_2O_3) and pure metallic (Ag) nanofluids.

CHAPTER 3

NUMERICAL INVESTIGATION

The Navier-Stokes equation (Continuity, momentum and energy equations) are solved to study the hydraulic and thermal behavior of a typical micro channel heat transfer. The chapter presents the numerical models for CFD analysis and validation of CFD results with experimental studies available in open literatures [33].

3.1 Governing Equations

The N-S equations are solved in the CFD study which includes continuity, momentum and energy equations. Since the fluid in the micro-channel is expected to be fully developed, turbulent equations are also needed to be solved. The turbulence modelling utilized in present study is RANS based two equation model i.e. RNG k-ε model with standard wall functions. However, in low Reynolds number cases (i.e. Re< 2000), the flow is expected to be laminar. In laminar flow cases, turbulent equations are not solved. The mathematical modelling equation [8] used in the CFD are as presented in subsequent paragraphs.

3.1.1 Mass conservation equation

The equation for conservation of mass, or continuity equation, can be written as follows:

$$\frac{\partial \rho}{\partial t} + \nabla(\rho V) = 0 \qquad 3.1$$

Equation (3.1) is in the form of a partial differential equation which relates the flow field variable at a point in the flow and valid for both incompressible and compressible flows.

3.1.2 Momentum conservation equation

Conservation of momentum is described by

$$\frac{\partial(\rho u)}{\partial t} + \nabla(\rho u V) = -\frac{\partial P}{\partial x} + \rho f_x + F_x \qquad 3.2a$$

$$\frac{\partial(\rho v)}{\partial t} + \nabla(\rho v V) = -\frac{\partial P}{\partial y} + \rho f_y + F_y \qquad 3.2b$$

$$\frac{\partial(\rho w)}{\partial t} + \nabla(\rho w V) = -\frac{\partial P}{\partial z} + \rho f_z + F_z \qquad 3.2c$$

Where equation (3.2a), (3.2b), (3.2c) represents the x-momentum, y-momentum and z-momentum respectively and the subscripts x, y, x, f and F denote the x, y, z components of the body and viscous forces respectively. Moreover, all the equations are applicable to the unsteady, compressible or incompressible, viscous or inviscid three-dimensional flow.

3.1.3 Energy equation

The energy equation is shown in the partial differential form in the following:

$$\frac{\partial}{\partial t}\left[\rho\left(e+\frac{V^2}{2}\right)\right]+\nabla\left[\rho\left(e+\frac{V^2}{2}\right)V\right]=\rho q'-\nabla(\rho V)+\rho(fV)+Q'_{viscous}+W'_{viscous} \quad 3.3$$

Where, e is the internal energy and $Q'_{viscous}, W'_{viscous}$ represent the proper form of the viscous terms.

3.2 Problem Specification for Validation

The first step in numerical analysis is undertaking of validation study to ascertain the correctness of CFD simulated results. For the purpose of validation, the paper presented by Sudo and Miyata et al. [33] is considered. In the experimental study mentioned above, the differences in the single-phase forced-convection heat transfer characteristics between upflow and downflow for a vertical rectangular channel simulating a subchannel in the fuel element of the research nuclear reactor, JRR-3 at the Japan Atomic Energy Research Institute. Each fuel element of the JRR-3 has twenty flat fuel plates arranged in parallel with 2.25 mm gap. The configuration of a subchannel is considered consisting of two adjacent rectangular fuel plates with 750mm in length, 66.6mm in width and 2.25mm in gap. For the condition of normal operation at 20 MW, fuel plates are cooled by downward flow at the velocity of about 6.2 m/s without allowing the nucleate boiling anywhere in the core. It is considered that there are no differences in the single-phase forced-convection heat transfer characteristics between upflow and downflow for such a high velocity at the normal operation condition of the JRR-3.

3.2.1 Experimental setup

The test loop is composed of a coolant storage tank, a recirculation line and a test section simulating a subchannel of a JRR-3 standard fuel element. As a coolant, water is used in this experiment. The coolant storage tank is 0.2 m^3 in volume and has a cooling line of copper coil in it to keep constant the coolant temperature at the inlet of test section during a test. The tank is open to the atmosphere & the recirculation line has a pump, a bypass line, a rotor flow meter, an electromagnetic flow meter, stop valves and regulating valves. Both upflow and downflow can be selected for the test section. The flow paths are shown with dotted lines for downflow and with solid lines for upflow in Fig. 3.1.

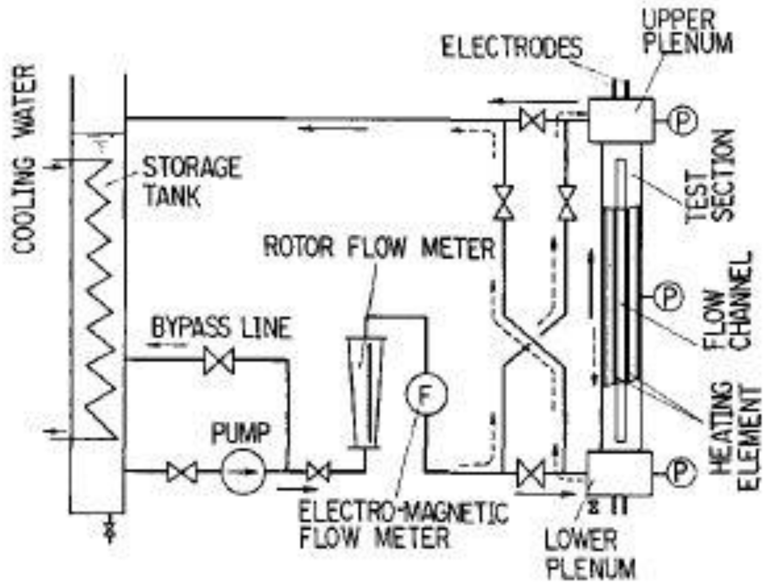


Figure 3.1: Schematic diagram of the experimental setup for validation [33].

The test section has a lower plenum and an upper plenum at the lower end and the upper end, respectively. The configuration of flow channel is rectangular with 50 mm in width, 2.25 mm in gap and 750 mm in length, and the flow channel is composed of adjacent two heating plates apart from each other at 2.25 mm. The heating plates are made of Inconel 600 with 1.0 mm in thickness and 40 mm in width. The lengths of heating plates are 750 mm for channel heated from both sides and 370 mm for channel heated from one side.

3.2.2 Geometry in ANSYS workbench

The Computational domain of the narrow rectangular micro channel is represented in two-dimensional (2D) form in ZX- plane as shown in Fig. 3.2. The geometry consists of a wall, a heated wall, an inlet and outlet boundaries.

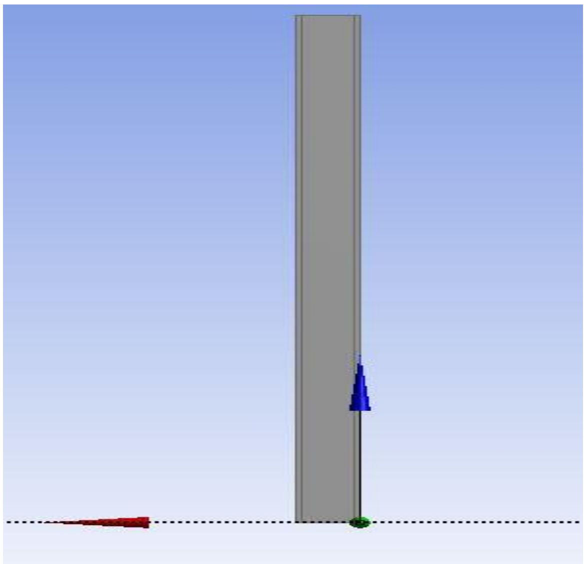
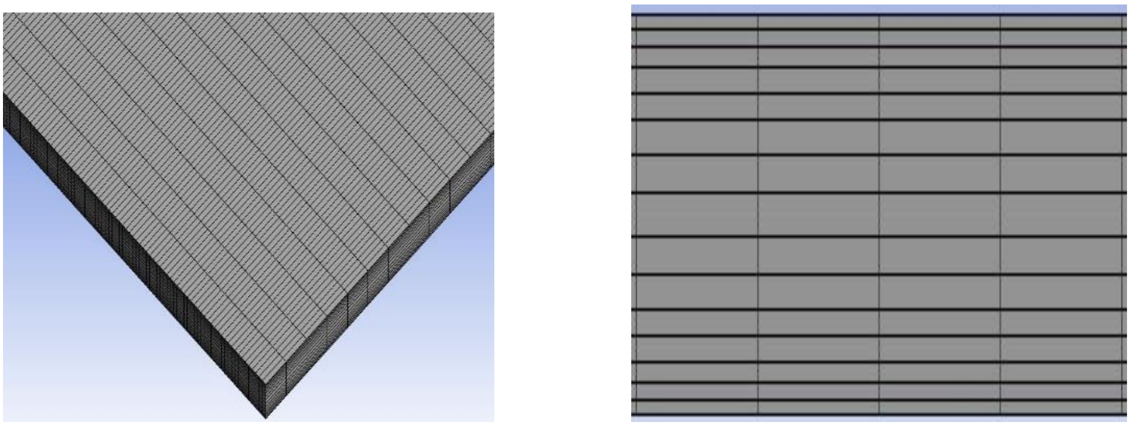


Figure 3.2: Geometry of narrow rectangular Micro channel in ANSYS 16.0.

3.2.3 Meshing of geometry

Structured meshing method done in ANSYS Workbench 16.0 was used for meshing the 3D geometry where, 0.675 million cells were created. The 2D view of geometry of narrow rectangular micro channel with structured mesh is shown in Fig. 3.3



(a) **(b)**

Figure 3.3: (a) A 3D geometry and (b) cross sectional view of microchannel.

3.2.4 Physical models

Since the experiment is focused on understanding the differences in single phase forced convection heat transfer characteristics between upflow and downflow for a high velocity at the normal operating condition of the research reactor, standard k-ε (RNG) model is used for turbulent flow.

3.2.5 Boundary conditions

A constant heat flux is applied on the channel wall. Axis symmetry was assigned at centerline. A uniform mass flow inlet and a constant inlet temperature were assigned at the channel inlet. At the exit, pressure was specified. For validation, particular test conditions of upward flow and downward flow was maintained. The inlet mass flow rate was 0.050278 kg/s, inlet temperature was 286 k, a constant heat flux of 32500 W/m^2 was applied for both upward and downward flow.

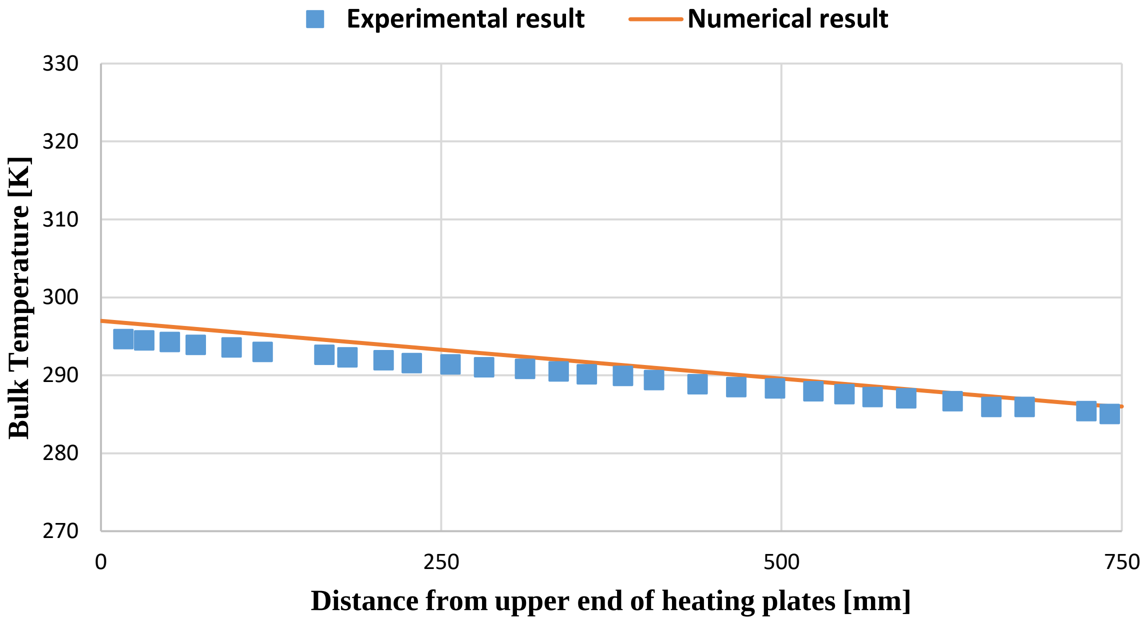
3.2.6 Method of solutions

Sudo Jo et al. 1984 have solved the problem experimentally and compared their solution with the existing correlations for turbulent forced-convection heat transfer. The existing correlations compared with are those proposed by Dittus-Boelter, Sieder-Tate and Colburn and among these, Dittus- Boelter correlation shows good prediction with the experimental results.

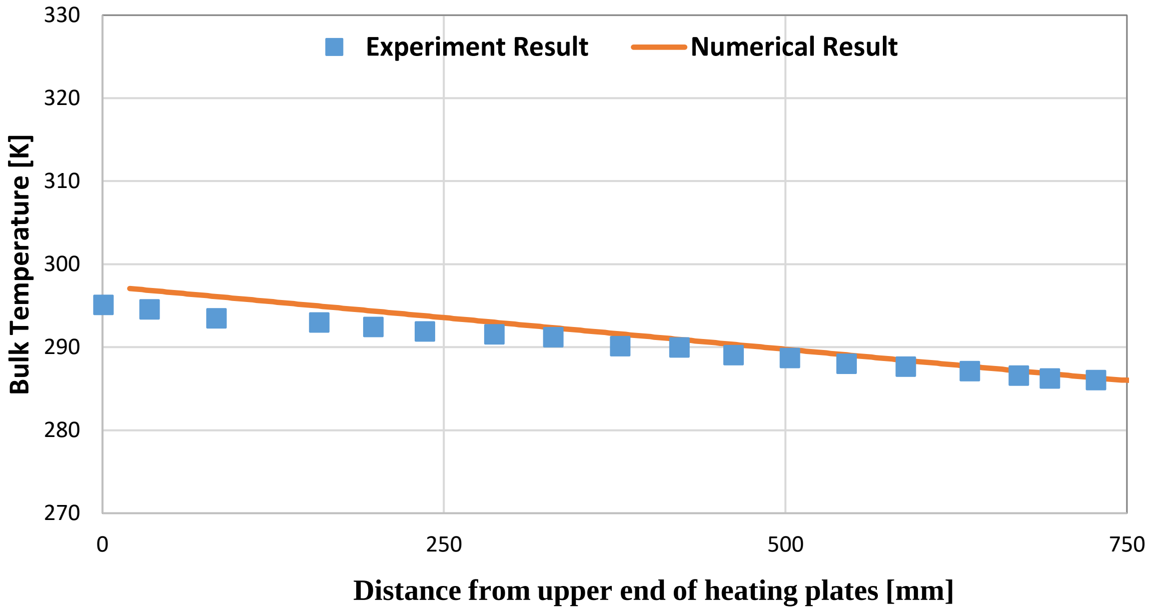
In the present validation study, the bulk temperature and heat transfer coefficients are obtained using CFD methods and then both the values are compared with the experimental results by Daeseong Jo et al. The CFD method follows the use of commercial software ANSYS Fluent 16.0 to solve the problem. The specified solver in Fluent uses a pressure correction based iterative SIMPLE algorithm with 2nd order power law scheme for discretizing the convective transport terms. The convergence criteria for all the dependent variables are specified as 0.000001. The default values of under-relaxation factor are same as the previous validation work.

3.2.7 Result and discussions

The thermal behavior of the channel can be studied in terms of temperature distribution within the channel. The variation of centerline temperature along axial position (X) with water for upward and downward flow are displayed in Fig. 3.4



(a)



(b)

Figure 3.4: Bulk temperature for (a) upward flow; (b) downward flow.

From figure: 3.4, it is observed that the bulk temperature increases linearly from inlet to outlet for both the flow and there is no significant difference in flow characteristics of upflow and downflow. The differences between the experimental and numerical result are 0.494% and 0.4703% for upflow and downflow respectively.

3.3 Mesh Independence Test

The simulation on study of forced convective heat transfer was performed using different grid number (mesh 1= 450000, mesh 2= 525000, mesh 3= 675000, mesh 4= 1050000 and mesh 5= 1350000) which is shown in Figure 3.5.

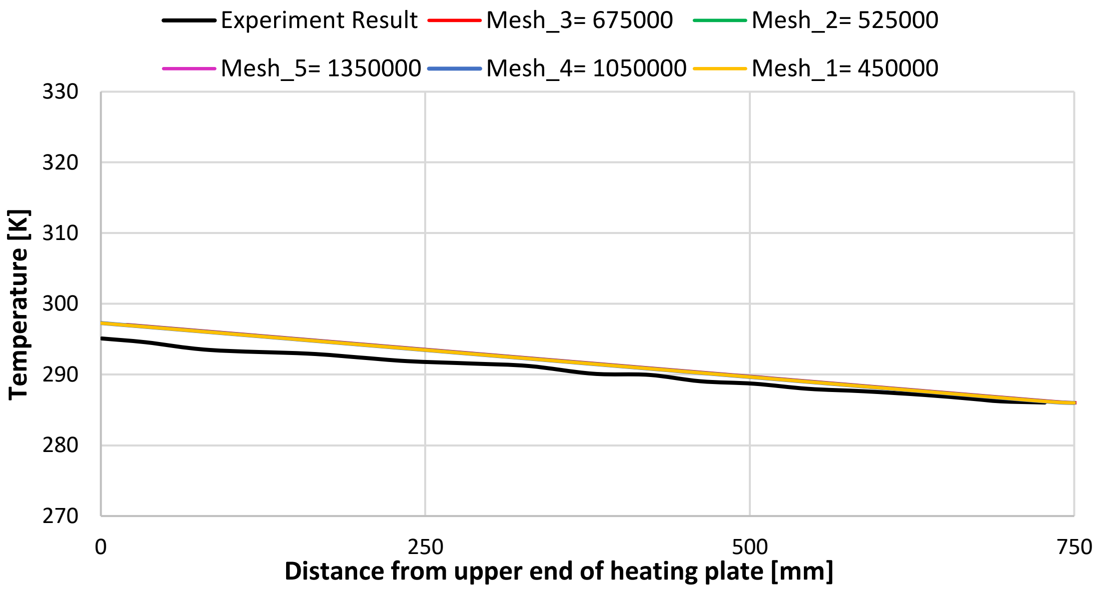


Figure 3.5: Grid Independence Test**.**

Comparing the result with the experimental values, an optimization is made between the degree of accuracy and the time for iteration and thus a grid of 675000 was chosen.

CHAPTER 4

COMPACT DESIGN AND MODELING OF FLUID FLOW

In the previous chapter, the experimental work performed by Sudo et al. was simulated where water was used as the only working fluid. This chapter deals with the similar simulation with the participation of different types of nanofluids is presented. The heat transfer enhancement using different combinations of nanofluids is observed. Moreover, a new generation heat exchanger is designed suitable for aviation use and its performance in terms of pumping power for different heat capacity and Reynolds number is presented.

4.1 Thermo-physical Properties of Nanofluid

Preparation of nanofluids is an important stage and nanofluids are prepared in a systematic and careful manner. A stable nanofluid with uniform particle dispersion is required and the same is used for measuring the thermo physical properties of nanofluids. In the present work, water is taken as the base fluid for preparation MgO, Al_2O_3, CuO nanofluids. Basically, three different methods are available for preparation of stable nanofluids, such as- i) By mixing of nano powder in the base liquid, ii) By acid treatment of base fluids, iii) By adding surfactants to the base fluid [34]. As the present work is only simulation based, preparation by mixing is considered for measuring the properties of nanofluids.

The most important properties needed for estimation of convective heat transfer coefficient of nanofluids are its density; thermal conductivity, viscosity, and specific heat. The thermo-properties of nanofluids are estimated using existing correlations for concentrations 1% to 5% and the results obtained are applied to understand their thermal and hydraulic behavior.

4.1.1 Estimation of nanoparticle volume concentration

The amount of nanoparticles required for preparation of nanofluids is calculated using the law of mixture formula. The weight of the nanoparticles required for preparation of 100 ml nanofluid of a particular volume concentration, using water base fluid is calculated by using the following relation as shown in equation 4.1.

$$\% \, Volume \, Concentration = \frac{\frac{W_p}{\rho_p}}{\frac{W_p}{\rho_p}+\frac{W_{bf}}{\rho_{bf}}} \qquad 4.1$$

4.1.2 Density of nanofluids

The base fluid consists of water and the density of nanofluids for different volume concentrations are measured using the density correlation equation (equation 4.2) developed by Pak and Cho for nanofluids, which is stated as follows

$$\rho_{nf} = \Phi\rho_p + (1-\Phi)\rho_{bf} \qquad 4.2$$

Where, ρ_{nf} = Density of nanofluids, kg/m^3

Φ = Nanoparticle volume concentration,

ρ_p = Density of nanoparticles, kg/m^3

ρ_{bf} = Density of base fluid, kg/m^3

4.1.3 Specific heat of nanofluids

The specific heat is one of the important properties and plays an important role in influencing heat transfer rate of nanofluids. Specific heat is the amount of heat required to raise the temperature of one gram of nanofluids by one degree centigrade. For a given volume concentration of nanoparticles in the base liquid, the specific heat can be calculated using the mixture formula. This formula is valid for homogeneous mixtures and is given by the Eq. (4.3).

$$C_{pnf} = \frac{(1-\Phi)\rho_{bf}C_{p\,bf}+\Phi\rho_{np}C_{p\,np}}{\mu_{np}} \qquad 4.3$$

4.1.4 Thermal conductivity of nanofluids

The nanofluids possess unique features with regard to their thermal performances. The properties of nanofluids are different from the properties of conventional heat transfer fluids. The nanoparticles offer large total surface area as a result of which higher thermal conductivities are expected in nanofluids. Many research findings reveal that traditional thermo fluids in the presence of nanoparticles exhibit better thermo physical properties.

Several classical models proposed by Maxwell, Hamilton and Crosser, Wasp are available in the literature [40] to predict the effective thermal conductivities of liquid-solid suspension. Maxwell developed a model (equation 4.4) to predict the effective thermal conductivity of solid-liquid suspension for low volume concentration of uniform sized spherical microparticles suspensions.

$$K_{nf} = K_{bf} \frac{3(\frac{K_{np}}{K_{bf}}-1)\Phi}{\left(\frac{K_{np}}{K_{bf}}+2\right)-(\frac{K_{np}}{K_{bf}}-1)\Phi} \quad 4.4$$

Thermal conductivity studies taken up on nanofluids by different research groups have reported a considerable enhancement in the thermal conductivity over the base fluids. To account for the shape of the particles, Maxwell's model was modified by Hamilton and Crosser (equation 4.5) by introducing a shape factor.

$$K_{nf} = K_{bf} \frac{K_{np}+(n-1)K_{bf}+(n-1)(K_{np}-K_{bf})\Phi}{K_{np}+(n-1)K_{bf}-(K_{np}-K_{bf})\Phi} \quad 4.5$$

Where n is the shape factor. The value of n is given by 3/ψ where ψ is the particle sphericity, defined as surface area of the sphere of same volume as that of given particle to the actual surface area of the particle. The value of ψ is 3 for sphere.

In the present study, Wasp model (equation 4.6) is used to predict effective thermal conductivity of nanofluids since it is considered as a homogeneous mixture:

$$K_{nf} = K_{bf} \frac{K_{np}+2K_{bf}-2\Phi(K_{bf}-K_{np})}{K_{np}+2K_{bf}+2K_{np}(K_{bf}-K_{np})} \quad 4.6$$

The viscosity of the nanofluid is determined from the well-known Einstein equation (Equation 4.7) for estimating viscosity in the present case, which is validated for spherical particles and volume concentrations not greater than 5.0 vol. %

$$\mu_{nf} = \mu_{bf}(1 + 2.5\Phi) \quad 4.7$$

Brinkman extended the Einstein's model to be used for moderate particle concentration. He assumed the mixture of solid particles and base fluid as continuum and formulated the theoretical correlation given as equation 4.8:

$$\mu_{nf} = \mu_{bf}(\frac{1}{((1-\Phi)^{2.5}}) \quad 4.8$$

Batchelor recognize the effect of Brownian motion of small particles in suspension. He modified Einstein's model by introducing the effect of Brownian motion. The model is applicable to isotropic suspension of rigid and spherical particles. The equation is given as:

$$\mu_{nf} = \mu_{bf}(1 + 2.5\Phi + 6.2\Phi^2) \qquad 4.9$$

Table 4.1: Calculation of thermo-physical properties of nanofluids using different models.

Volume Fraction of nanoparticles	Base Fluid	Density of MgO/Water	Specific heat of MgO/Water		Thermal Conductivity of MgO/Water			Viscosity of MgO/water		
		Pak & Cho	Mixture Formula	Pak & Cho	Wasp	Hamilton & Crosser	Maxwell	Einstein	Brinkman	Batchelor
1% MgO	Water	1013.92	4065.199	4148.92	0.620062	0.608418	0.017662	0.001028	0.001029	0.001029
2% MgO		1039.84	3954.222	4115.84	0.638072	0.614557	0.035672	0.001053	0.001055	0.001056
3% MgO		1065.76	3848.642	4082.76	0.656442	0.620821	0.054042	0.001078	0.001082	0.001084
4% MgO		1091.68	3748.077	4049.68	0.675181	0.627214	0.072781	0.001103	0.001111	0.001113
5% MgO		1117.6	3652.175	4016.6	0.694302	0.63374	0.091902	0.001128	0.001140	0.001144

Before applying the specific models in the present study, the effects of different models have been considered. For density calculation, Pak and Cho equation is the most widely used one and for viscosity measurement, the three available models show almost similar values. Therefore, only the models for specific heat and thermal conductivity have been compared.

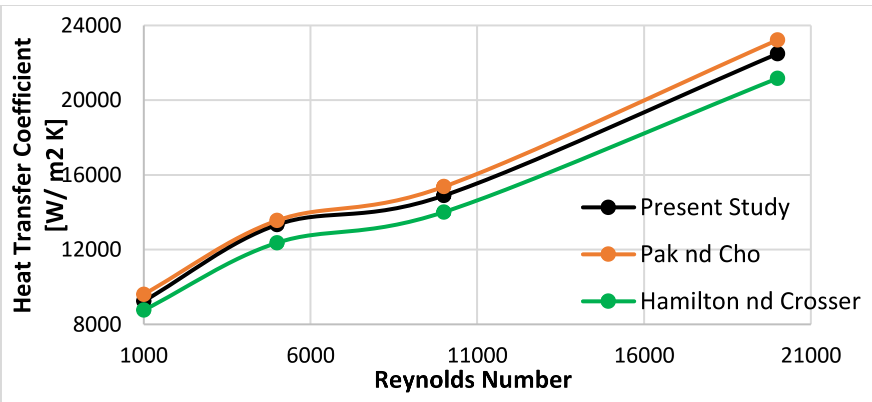


Figure 4.1: Comparison of various models.

In the present study, Pak and Cho equation, mixture formula, Wasp model and Einstein equation have been used for calculation of density, specific heat, thermal conductivity and viscosity respectively. For the second case, Pak and Cho correlation is used for the specific heat calculation and for the third case, Hamilton and Crosser model have been used which indicate a difference of 3% and 6% with the present case respectively. Therefore, any of the models can be used for the prediction of thermo-hydraulic performance of nanofluids.

Table 4.2: Thermo-physical properties of nanofluids at 1% volume concentration.

Nanofluids	Density [kg/ m^3]	Specific Heat [J/ Kg K]	Thermal Conductivity [W/ mK]	Viscosity [Kg/ ms]
Al_2O_3/ Water	1017.02	4055.702	0.619735	0.001028
MgO/ Water	1013.92	4065.199	0.620062	0.001028
CuO/ Water	1043.12	3954.62	0.618889	0.001028
Al_2O_3-Cu/ Water	1042.47	3956.76	0.620286	0.001028
Ag-MgO/ Water	1048.713	3927.174	0.620454	0.001028
CuO-Cu/ Water	1055.519	3908.042	0.619858	0.001028

Table 4.3: Thermo-physical properties of nanofluid at 5% volume concentration.

Nanofluids	Density [kg/ m^3]	Specific Heat [J/ Kg K]	Thermal Conductivity [W/ mK]	Viscosity [Kg/ ms]
Al_2O_3/ Water	1133.1	3615.202	0.692531	0.001128
MgO/ Water	1117.6	3652.175	0.694302	0.001128
CuO/ Water	1263.6	3243.471	0.687967	0.001128
Al_2O_3-Cu/ Water	1264.4	3239.82	0.698392	0.001136
Ag-MgO/ Water	1296.527	3135.609	0.699344	0.001136
CuO-Cu/ Water	1329.607	3081.703	0.695984	0.001136

4.2 Identification of Efficient Nanofluid

4.2.1 Heat transfer coefficient for different volume fraction of nanofluid

The figure 4.2, 4.3 and 4.4 represents the effect of volume fraction of nanofluids on heat transfer coefficient at different Reynolds number. From the figures, it is observed that the value of convective heat transfer coefficient increases with the increase of volume concentration and Reynolds number. As the thermal conductivity increases and specific heat decreases with the addition of nanofluids, the difference between wall temperature and mean temperature decreases and this phenomenon increases the heat transfer coefficient gradually.

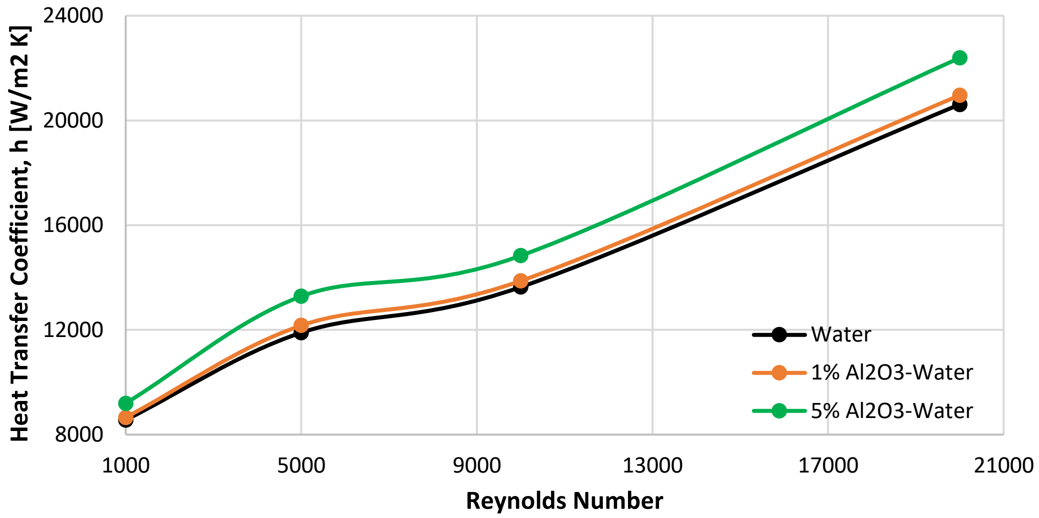


Figure 4.2: Comparison of heat transfer coefficient for different volume fraction of Al_2O_3-Water nanofluids.

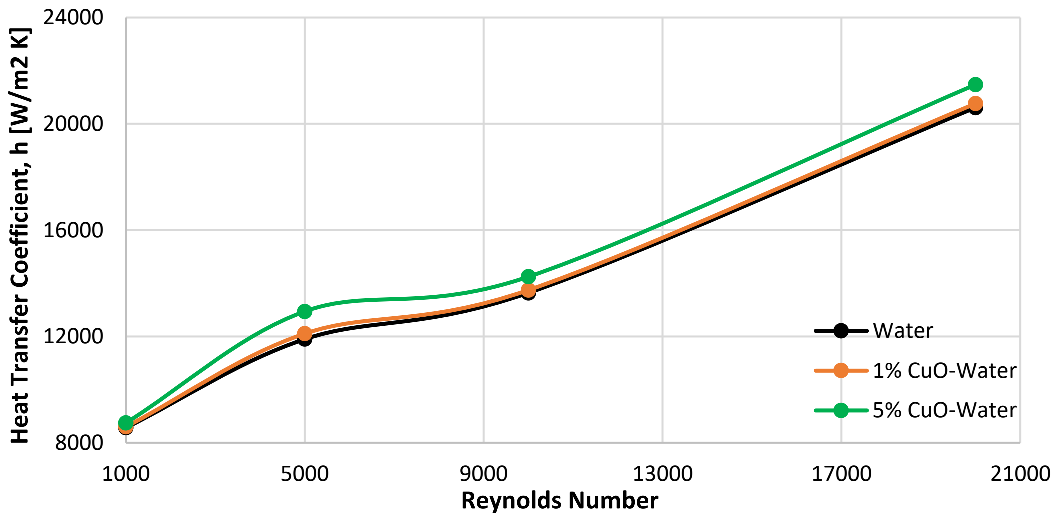


Figure 4.3: Comparison of heat transfer coefficient for different volume fraction of CuO-Water nanofluids.

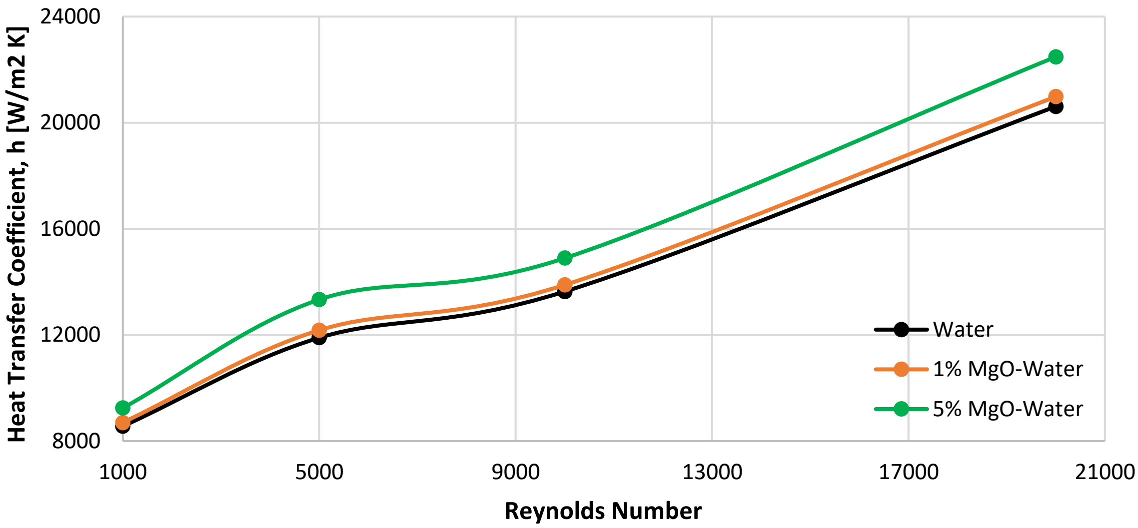


Figure 4.4: Comparison of heat transfer coefficient for different volume fraction of MgO-Water nanofluids.

It is found that 1% and 5% MgO-water show enhancement of 1.91% and 9.6% where as Al_2O_3-Water indicates 1.76% and 9.13% enhancement and CuO-Water represents 0.92% and 4.9% enhancement in heat transfer coefficient respectively when compared to pure water.

4.2.2 Heat transfer coefficient for constant volume fraction of different nanofluids

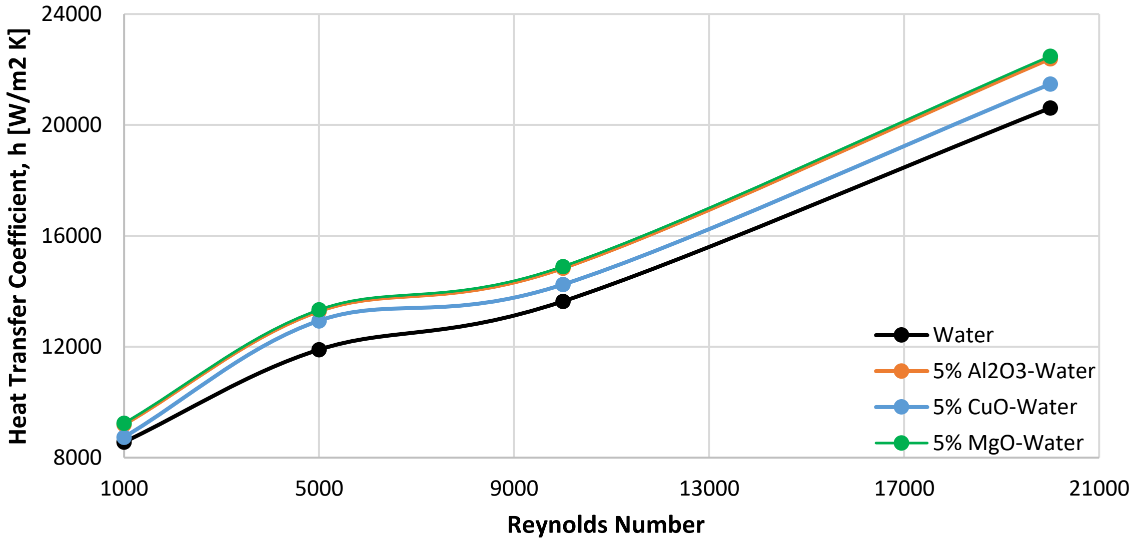


Figure 4.5: Comparison of heat transfer coefficient for 5% volume fraction of metal oxides nanoparticles.

Figure 4.5 indicates the effect of 5% volume fraction of different metal oxide nanoparticles (Al_2O_3, CuO, MgO) on heat transfer coefficient. Here, MgO-water nanofluid is enhancing the heat transfer characteristics over Al_2O_3- water and CuO-water by 0.42% and 4.45% respectively. In figure 4.6 to 4.8, the effects of multispecies nanofluids are shown and compared with that of water and single species nanofluids. It is found that the highest heat transfer coefficient is obtained in case of metal-oxide single species nanoparticles.

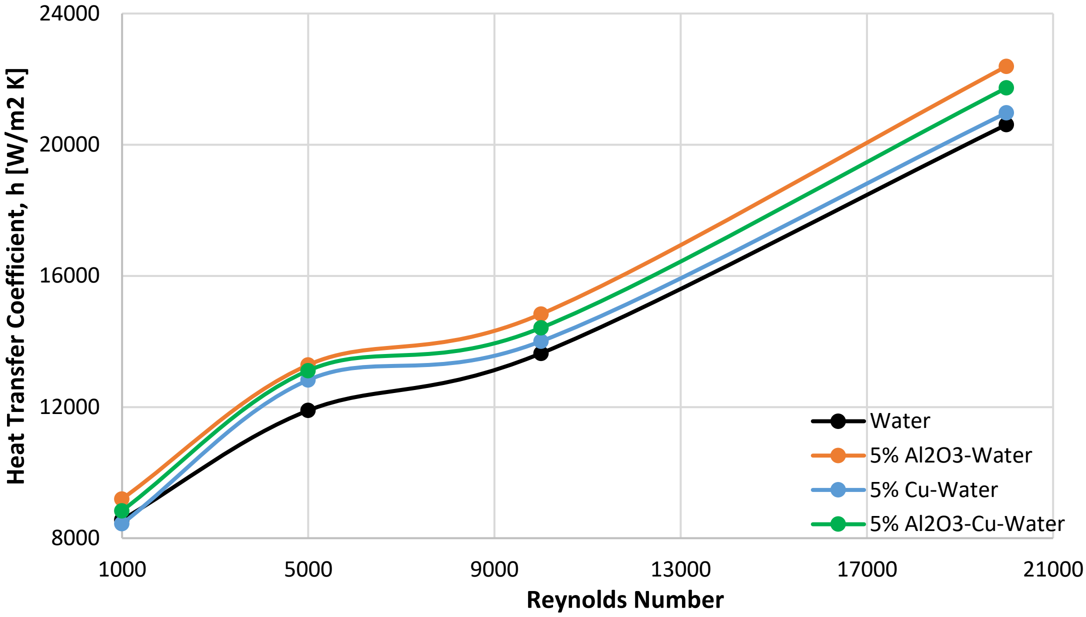


Figure 4.6: Comparison of heat transfer coefficient for 5% volume fraction of single and multispecies nanoparticles.

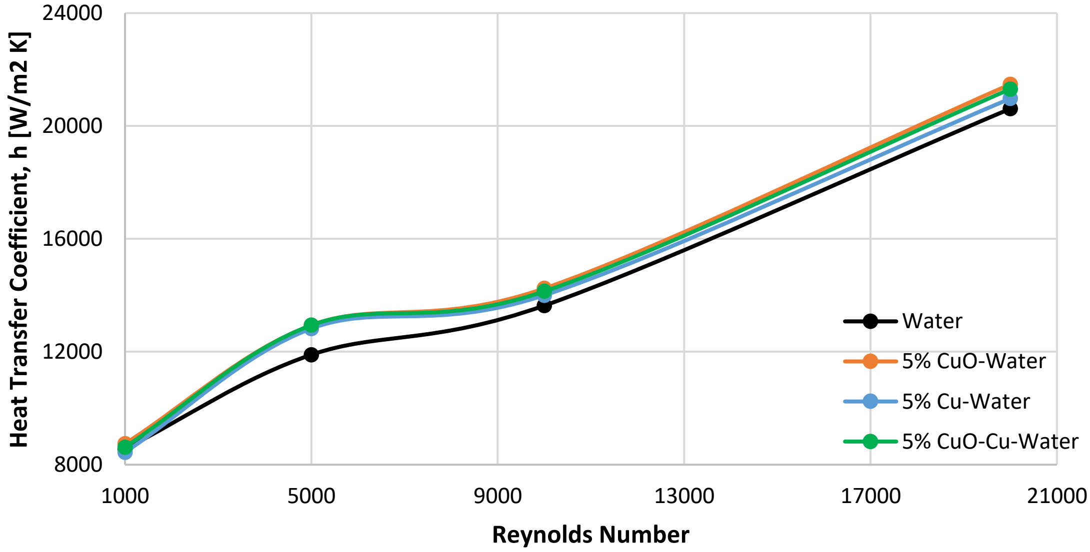


Figure 4.7: Comparison of heat transfer coefficient for 5% volume fraction of single and multispecies nanoparticles.

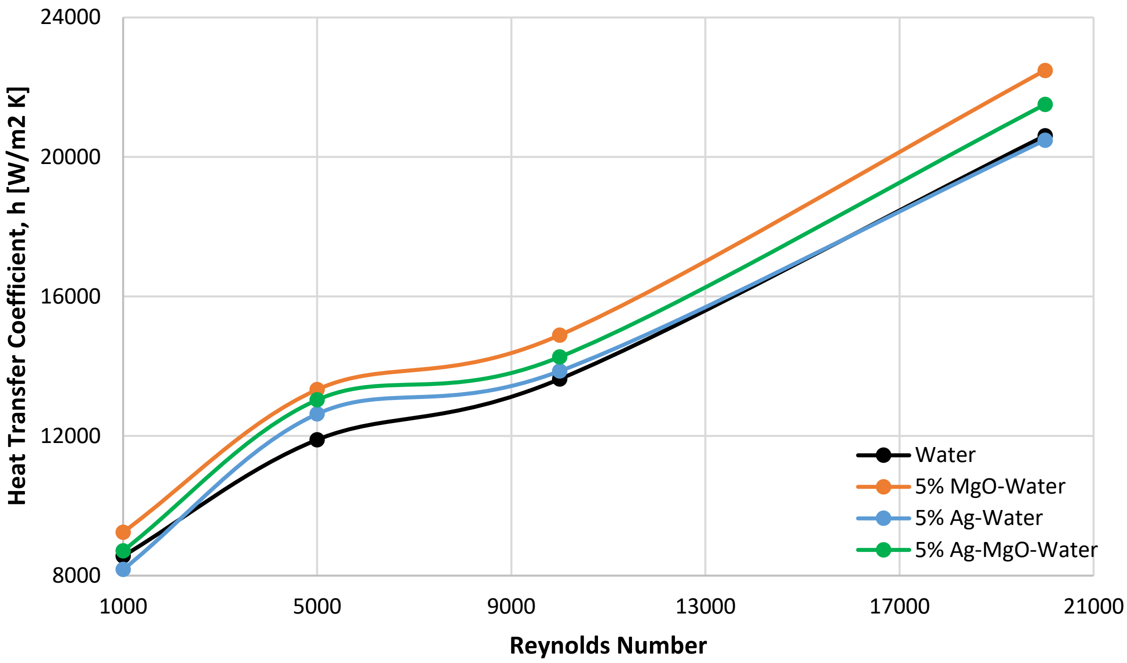


Figure 4.8: Comparison of heat transfer coefficient for 5% volume fraction of single and multispecies nanoparticles.

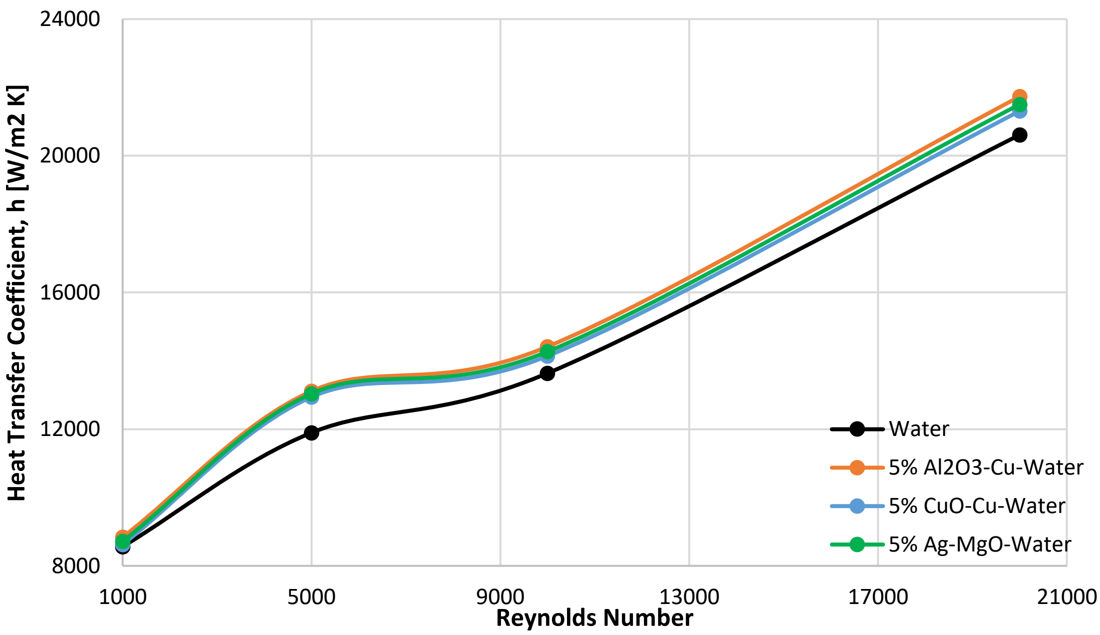


Figure 4.9: Comparison of heat transfer coefficient for 5% volume fraction of different multispecies nanoparticles.

Figure 4.9 represents the heat transfer enhancement using three different multispecies nanofluids among which Al_2O_3- Cu-water nanofluid is showing better performance over Ag-MgO-water and CuO-Cu-water nanofluids by 0.99% and 1.9% respectively.

4.2.3 Nusselt number for different volume fraction of nanofluids

The figure 4.10, 4.11 and 4.12 represents the effect of volume fraction of nanofluids on Nusselt number at different Reynolds number. From the figures, it is observed that the Nusselt number increases with the increase of volume concentration and Reynolds number for all the nanofluids. As the convective heat transfer coefficient is increasing with the addition of nanofluids, the Nusselt number is increasing gradually assuming the thermal conductivity as constant.

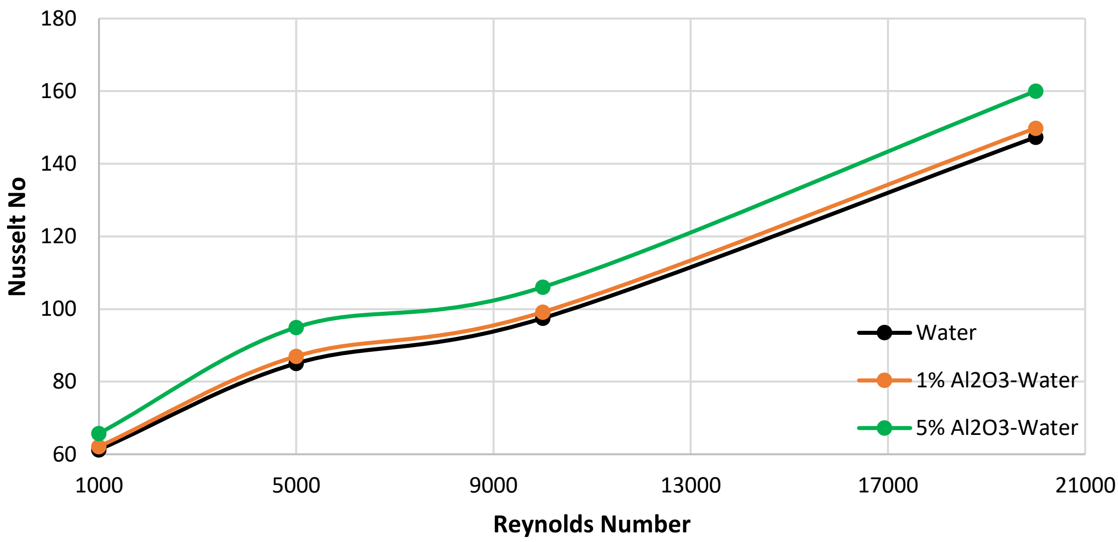


Figure 4.10: Comparison of Nusselt number for different volume fraction of Al_2O_3-Water nanofluids.

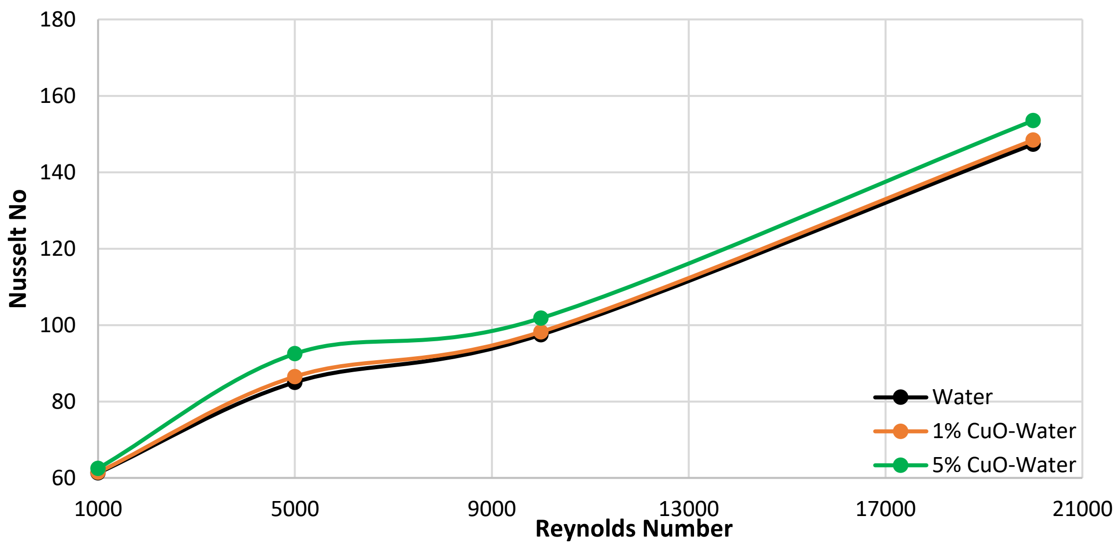


Figure 4.11: Comparison of Nusselt number for different volume fraction of CuO-Water nanofluids.

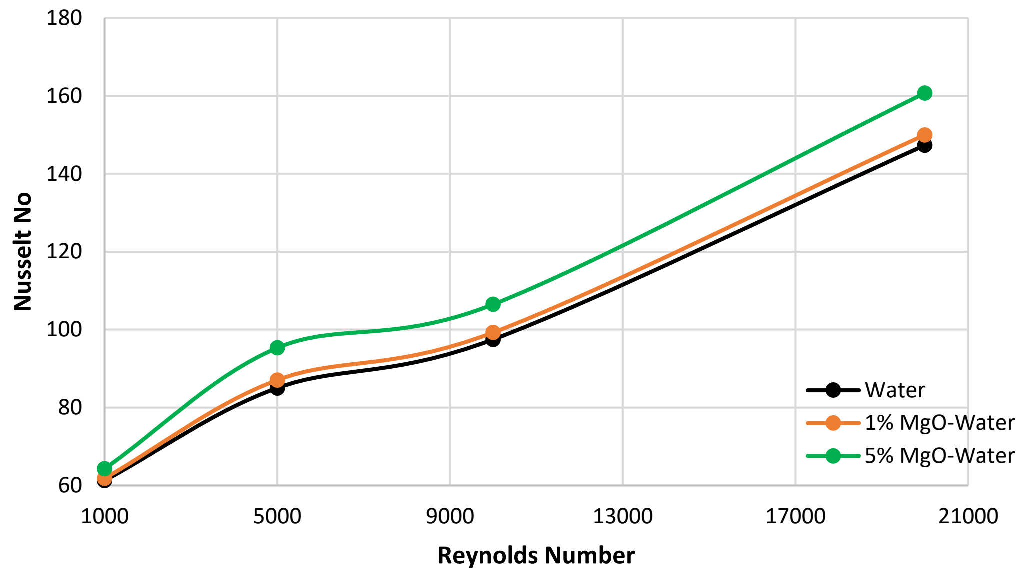


Figure 4.12: Comparison of Nusselt number for different volume fraction of MgO-Water nanofluids.

The enhancement in Nusselt number is almost similar to that of convective heat transfer coefficient, but not exactly same as because of the change in thermal conductivity of nanofluids.

4.2.4 Nusselt number for constant volume fraction of different nanofluids

Figure 4.13 indicates the effect of 5% volume fraction of different metal oxide nanoparticles (Al_2O_3, CuO, MgO) on Nusselt number respectively. It is found that MgO-Water shows the highest Nusselt number followed by Al_2O_3-Water and CuO-Water in comparison with water.

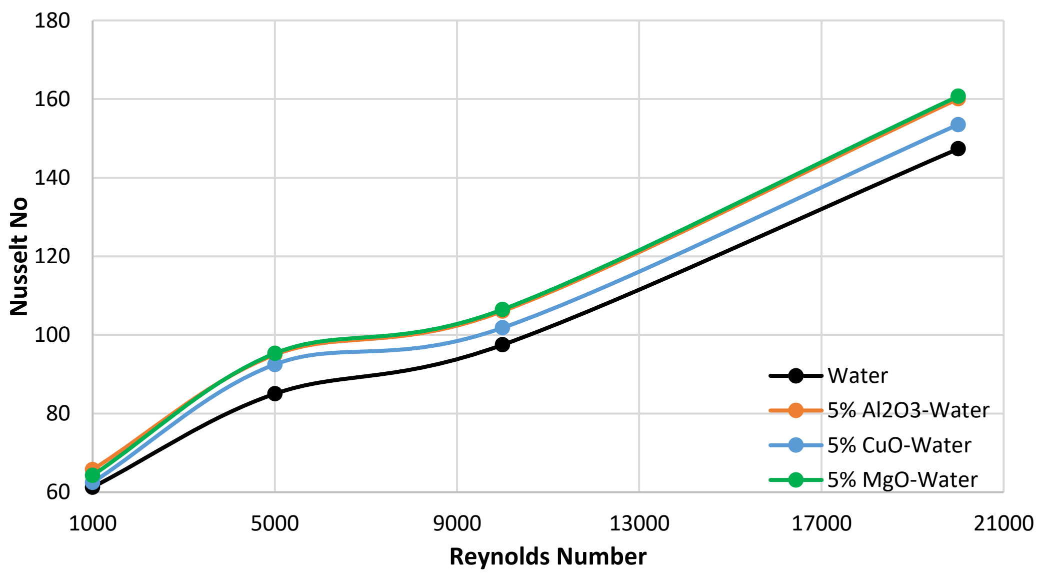


Figure 4.13: Comparison of Nusselt number for 5% volume fraction of metal oxides nanoparticles.

In the following figures 4.14 to 4.17, the effects of multispecies nanofluids are shown and compared with that of water and single species nanofluids. It is found that the Nusselt number is increased by 5.79% for 5% volume fraction of Al_2O_3-Cu- Water compared to water. However, Ag-MgO-Water signifies the Nusselt number by 0.82% and 4.9% and CuO-Cu-Water enhances by 0.64% and 4.1% respectively.

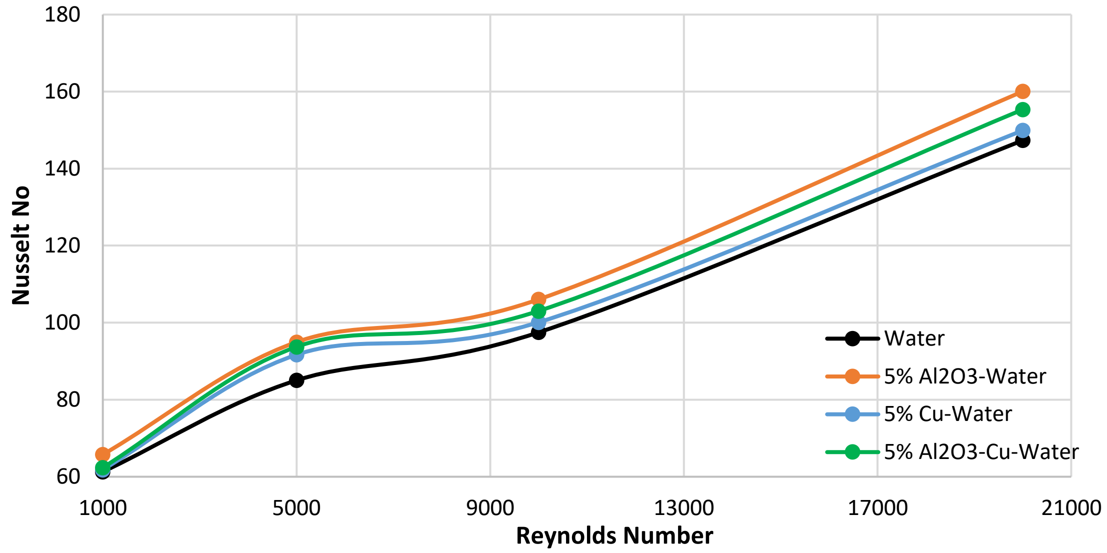


Figure 4.14: Comparison of Nusselt number for 5% volume fraction of single and multispecies nanoparticles.

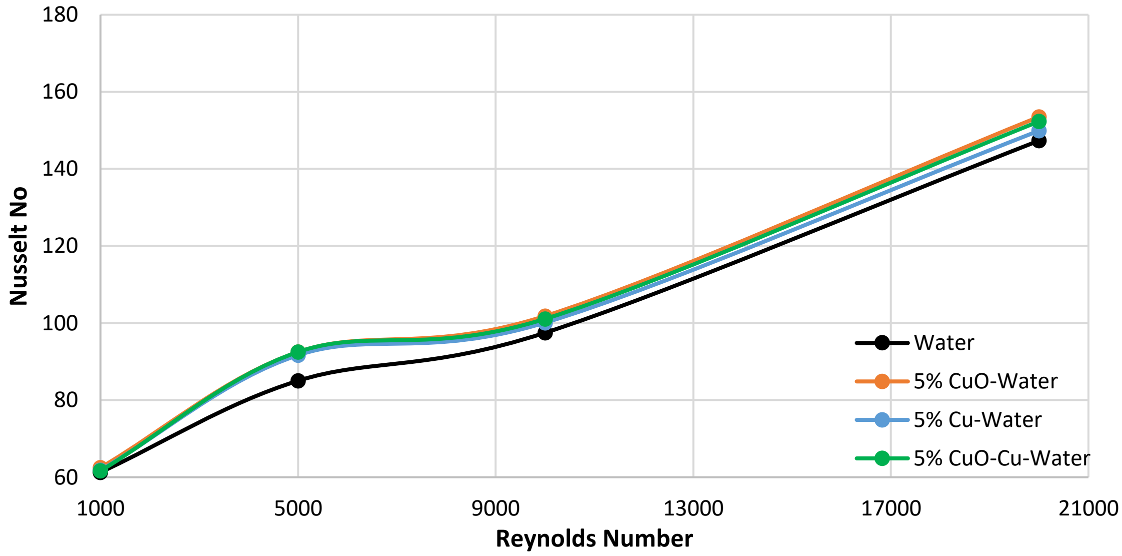


Figure 4.15: Comparison of Nusselt number for 5% volume fraction of single and multispecies nanoparticles.

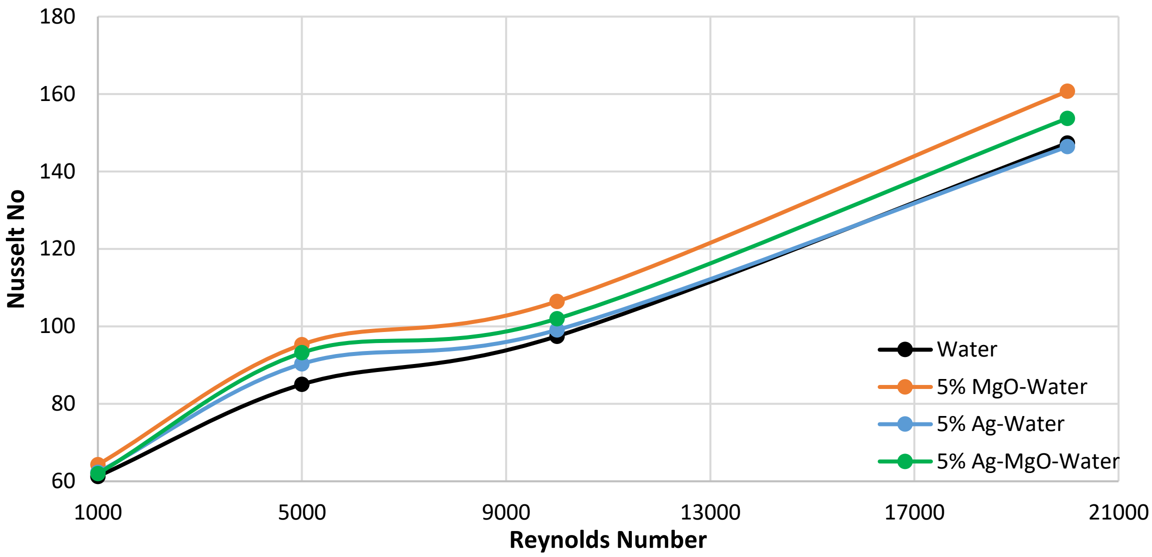


Figure 4.16: Comparison of Nusselt number for 5% volume fraction of single and multispecies nanoparticles.

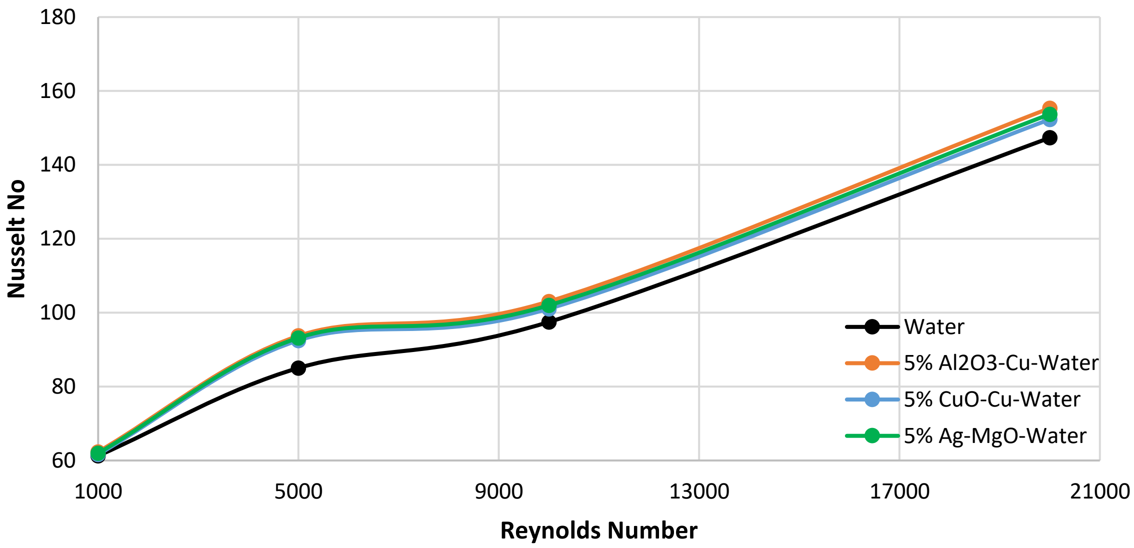


Figure 4.17: Comparison of Nusselt number for 5% volume fraction of different multispecies nanoparticles.

From figure 4.17, it is evident that the highest Nusselt number is obtained in case of Al_2O_3-Cu- Water and it is dominating over Ag-MgO-water and CuO-Cu-water nanofluids by 0.87% and 1.7% respectively.

4.3 Design Specification for Heat Exchanger

A compact heat exchanger has been designed suitable for aviation use where the temperature rise is restricted to 70 ˚C. Its hydraulic diameter is 4.3 mm and length is 2.30

m and the heat pipe is dipped into a 30×30 cm heat sink. Design specifications and test conditions for a compact heat exchanger are mentioned below:

Table 4.4: Design parameters of heat exchanger

Parameters	**Range**
Temperature Rise, ΔT	70 ˚C
Inlet Temperature	286 K
Heat Transfer Rate, Q	1 kW- 100 kW
Dimension	50×2.25×2300 mm
Reynold number	500-20000
Inlet Velocity	Dependent on Re and thermo-physical properties of nanofluids

4.3.1 Geometry of the computational domain

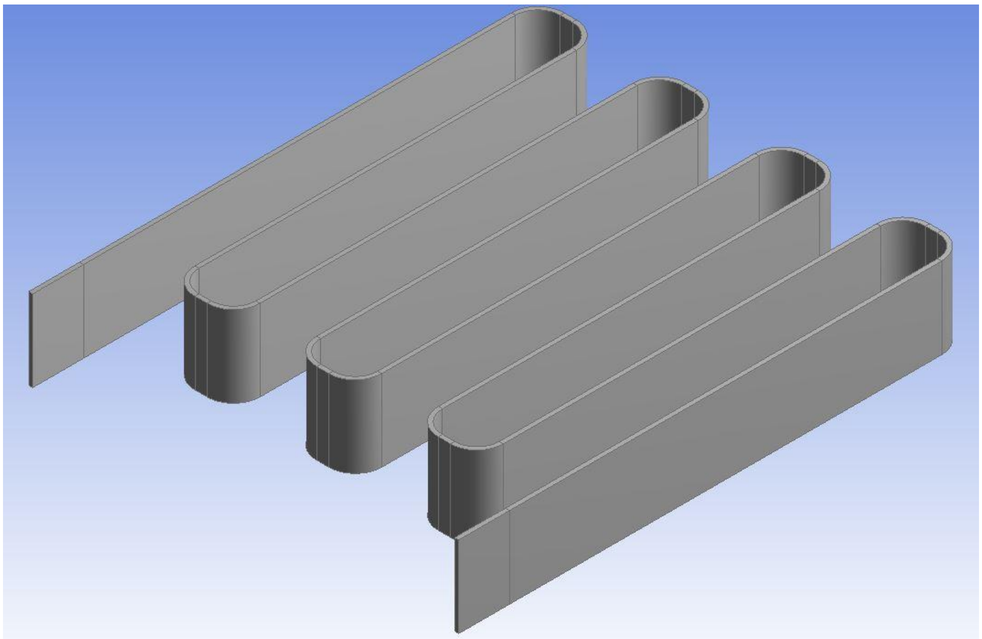


Figure 4.18: Computational domain of a rectangular heat exchanger.

The inlet velocity, *u (*m/s) is dependent on Reynolds number and the thermophysical properties of different nanofluids used. The micro channel is made of oxygen free copper.

4.3.2 Meshing of the computational domain

Structured mesh method was used for meshing the geometry as shown in Figure 4.18. 574387 nodes were created including all three-dimensional coordinates. Figure 4.19 represents three-dimensional geometry of rectangular micro channel with structured mesh.

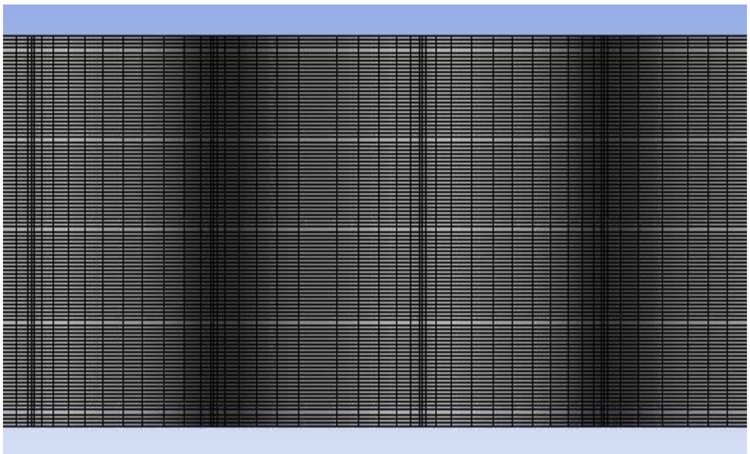


Figure 4.19: A 2D view of rectangular micro channel with structured mesh.

4.3.3 Physical model

Physical model for simulation of a rectangular heat exchanger is same as discussed for validation in chapter 3.

4.3.4 Boundary conditions

A no slip boundary condition was assigned for the surfaces, where both velocity components were set to zero at that boundary. A constant temperature and pressure were assigned at the channel inlet. Assuming the thermal power range 1- 100 kW, a constant heat flux is obtained and that effective heat flux is assigned to all the sink surfaces and other surfaces out of heat sink are subjected to adiabatic conditions (heat flux is zero).

4.3.5 Method of solutions

The specified solver uses a pressure correction based iterative SIMPLE algorithm with 1st order upwind scheme for discretization of convective transport terms. The convergence criteria for all the dependent variables are specified as 0.000001. The default values of under relaxation factor as shown for validation in Table 3 are used in the simulation work.

4.4 Result and Discussions

Among the above mentioned nanofluids, MgO-Water and Al_2O_3-Cu-Water is used as the coolant in the heat exchanger to find the convective heat transfer characteristics and required pumping power. It is evident from the following figure 4.21 that the requirement of the pumping power increases exponentially with the increase of Reynolds number. Therefore, it is relatable that, to enhance the convective heat transfer coefficient as well as Nusselt number, a higher pumping power is required.

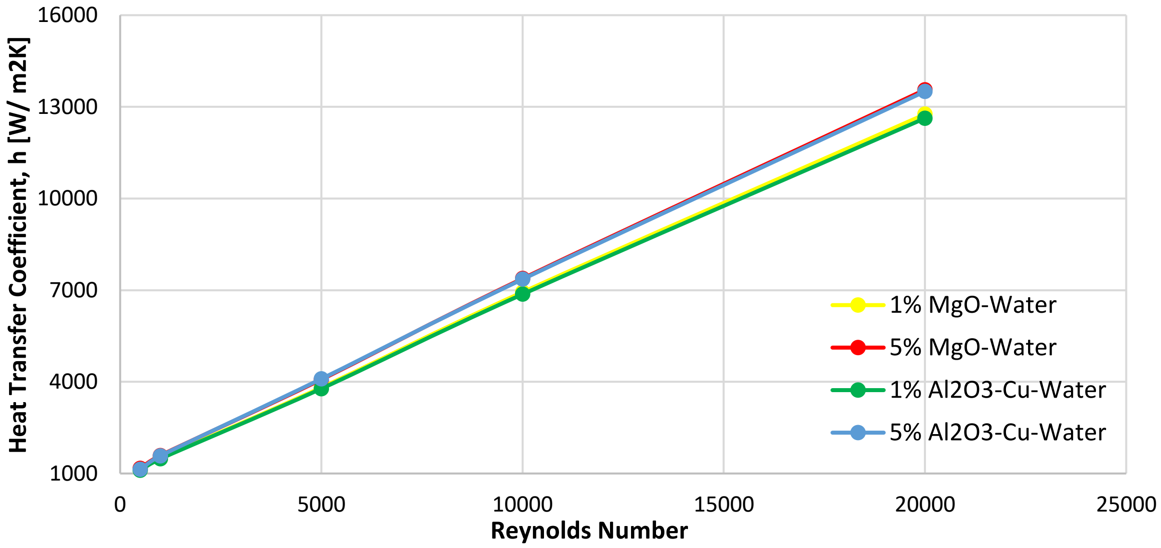


Figure 4.20: Comparison of heat transfer coefficient for nanofluids through heat exchanger.

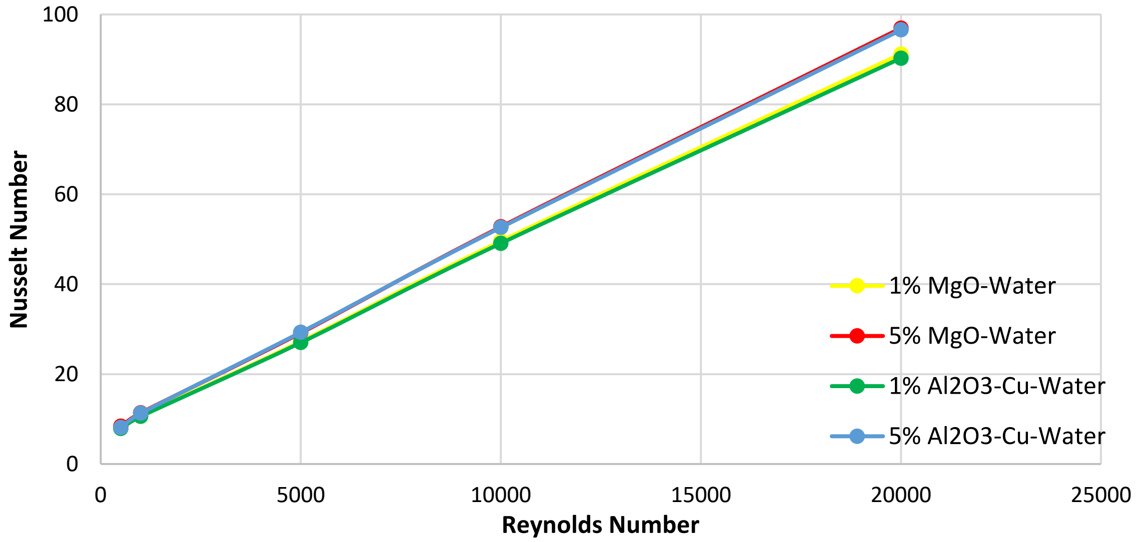


Figure 4.21: Comparison of Nusselt number for nanofluids through heat exchanger.

Here in figure 4.20 and 4.21, the enhancement is more in case of 5% MgO nanoparticles compared to 5% Al_2O_3-Cu nanoparticles by 1.26% and 0.904% for heat transfer coefficient and Nusselt number respectively.

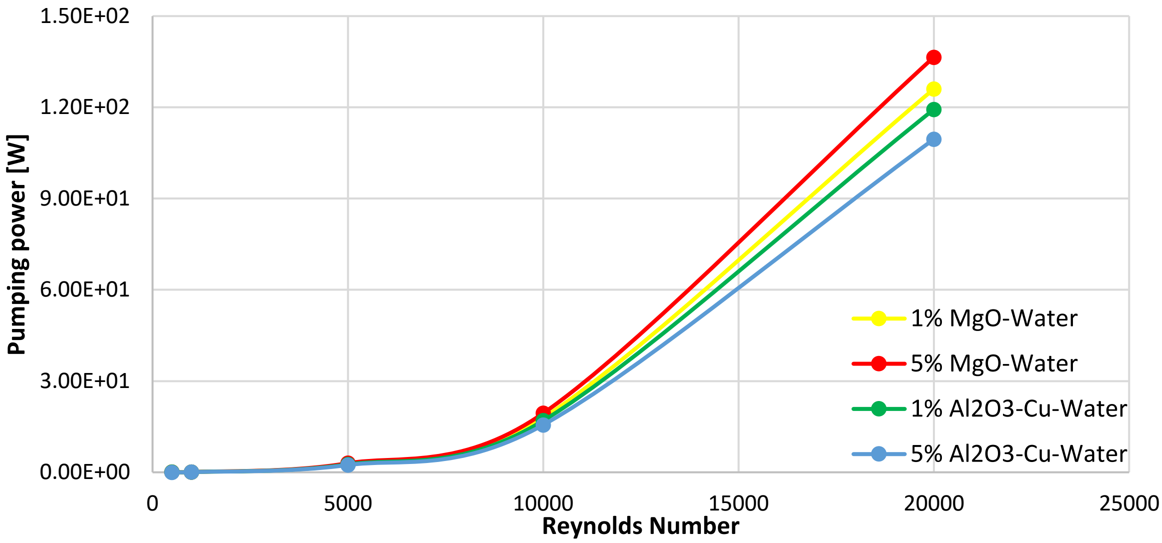


Figure 4.22: Requirement of pumping power of the heat exchanger for Al_2O_3-Cu-Water nanofluids.

It is seen from figure 4.22 that the pumping power requirement increases with the addition of nanofluids. In case of MgO-water, as the volume concentration increases from 1% to 5%, the requirement of pumping power increases by 8.3% and in case of Al_2O_3-Cu-water, the increment is 8.94%. If it is compared between 5% MgO-water and Al_2O_3-Cu-water, it is Al_2O_3-Cu-water that provides reduces power requirement by 24.6%. Moreover, the increment in power is exponential with the increase of Reynold number.

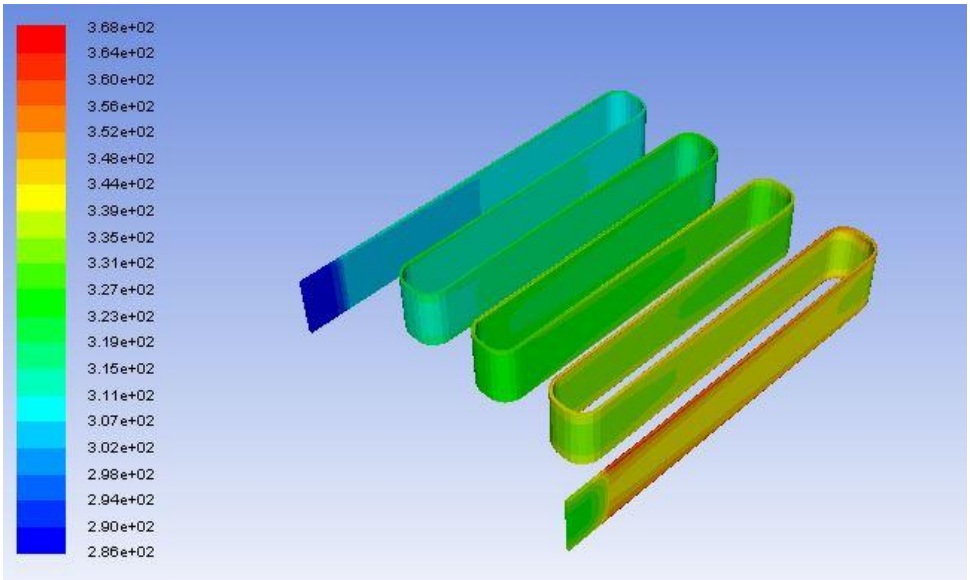


Figure 4.23: Temperature contour for 5% MgO-Water nanofluids

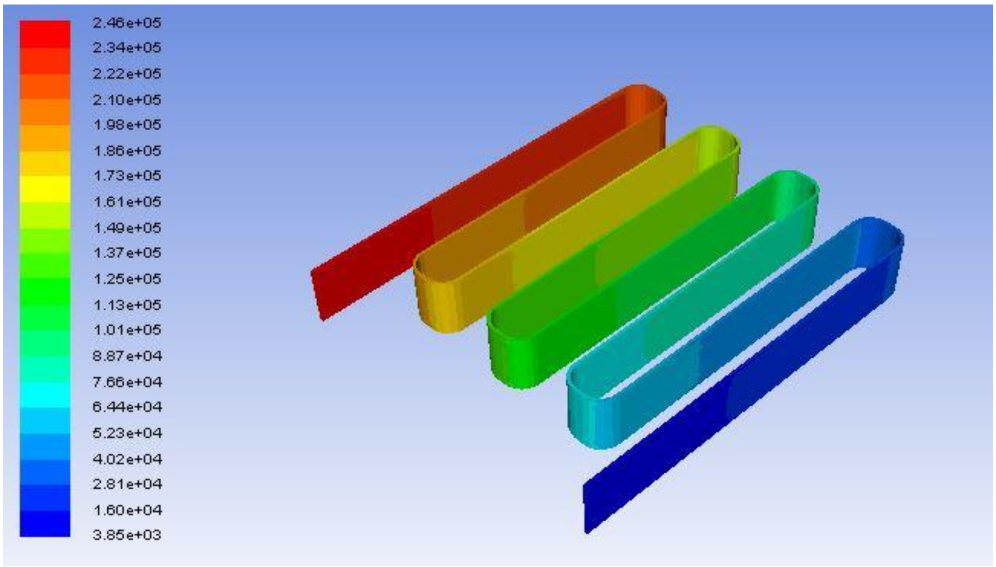


Figure 4.24: Pressure contour for 5% MgO-Water nanofluids.

Figure 4.23 and 4.24 represent the temperature and pressure distribution throughout the heat pipe respectively. In case of temperature contour, it is seen that the temperature is maximum near the wall due to the boundary layer effect and the maximum temperature rise is restricted before the boiling point. In case of pressure contour, the pressure decreases gradually from inlet to outlet as it drives the fluid flow.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

The extensive CFD analysis of different combinations of nano-fluids to identify efficient nanofluid for enhanced heat transfer has been undertaken and the results are presented in previous chapters. The summery of the results and the conclusions drawn and the recommendations for further study are presented in this chapter.

5.1 Conclusion

The numerical analysis of heat transfer has been undertaken in present study. The possibility to enhance the heat transfer characteristics using different nanofluids has been undertaken. The salient results from the extensive CFD analysis are as presented below:

a) A total of three metal oxides, two metal nano particles in two volume concentration were considered to create a total sixteen combinations of nano-fluids for four different Reynolds numbers. The thermal performance of the nano fluids is compared w.r.t. convective heat transfer coefficient and Nusselt number.

b) The results suggest that the convective heat transfer coefficient and Nusselt number increase with the increase of volume concentrations of nanofluids as well as the Reynolds number.

c) Among the three multispecies nanofluids, Al_2O_3-Cu-water indicates the better enhancement, MgO-water performs better out of all the oxide nanofluids and overall among all combinations. Thus, MgO-water is identified as the best coolant irrespective of volume concentration.

d) Based on comparative performance of nanofluids in microchannels, a compact microchannel heat exchanger is designed for different heat capacities (i.e. 1 kW, 50 kW and 100 kW) which is suitable for aviation use and where the maximum temperature rise is restricted to 70ºC. The thermo- hydraulic behavior of the compact micro-channel heat exchanger is analyzed with the more efficient coolants out of all the single and multispecies nanofluids studied in the present work.

e) As mentioned above, MgO-water acts as the best coolant showing convective heat transfer enhancement of 1.91% and 9.6% compared to water for 1% and 5% volume concentration respectively and among the multispecies nanofluids, Al_2O_3-

Cu-water performs better. Therefore, MgO-water and Al_2O_3-Cu-water have been used in the heat exchanger to investigate its thermal and hydraulic performance and to determine the requirement of pumping power.

f) The pumping power required is calculated based on the volume flow rate and total pressure loss. It is observed that the pumping power increases with increasing Reynolds number nearly exponentially.

5.2 Recommendations

Due to lack of sufficient experimental facilities, only a numerical study of single-phase flow is carried out to investigate the thermal and hydraulic performance of nanofluids as coolant. However, the following recommendations are made to extend present research further in future:

a) A similar study can be conducted using different shape of heat exchangers like double pipe heat exchangers or shell and tube heat exchangers and so on.

b) To reduce the pumping power, various wick structures can be used inside the heat pipe so that the wick itself can create a capillary pressure and drive the nanofluids from inlet to outlet.

c) As the nanofluids are prepared by suspension of solid nanoparticles in liquid, its performance can be well investigated if the fluid is considered as a two phase non-homogeneous flow. Thus, a discrete phase model (DPM) can be applied to understand the interaction between solid and liquid phase and also its adverse effects can be acknowledged.

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APPENDIX A: Calculation of convective heat transfer coefficient and Nusselt number using nanofluids.

For Water				
Re=5000				
Tw [K]	Tm [K]	Tw-Tm [K]	$h = \frac{Heat\ Flux}{T_w - T_m}$ [W/ m^{2}K]	$Nu = \frac{D_h \times h}{K_w}$
292.88425	290.04105	2.8432	11430.78	81.70808
292.59235	289.76199	2.83036	11482.64	82.07875
292.15451	289.35082	2.80369	11591.87	82.85952
291.71667	288.93871	2.77796	11699.23	83.62698
291.27884	288.52563	2.75321	11804.4	84.37875
290.84103	288.11152	2.72951	11906.9	85.1114
290.4032	287.69638	2.70682	12006.71	85.82485
289.96536	287.2801	2.68526	12103.11	86.51394
289.52756	286.86271	2.66485	12195.81	87.17655
289.08957	286.44414	2.64543	12285.34	87.81651
288.94217	286.3098	2.63237	12346.29	88.25219

For 5% MgO-Water				
Re=5000				
Tw [K]	Tm [K]	Tw-Tm [K]	$h = \frac{Heat\ Flux}{T_w - T_m}$ [W/ m^{2}K]	$Nu = \frac{D_h \times h}{K_w}$
292.67245	290.12143	2.551	12740	91.06648
292.37476	289.83681	2.5379	12805.61	91.53546
291.92819	289.41747	2.5107	12944.49	92.52821
291.48166	288.99716	2.4845	13081.1	93.5047
291.03513	288.57587	2.4593	13215.36	94.46436
290.58856	288.15352	2.435	13346.8	95.40394
290.14203	287.73012	2.4119	13474.8	96.31886
289.6955	287.30556	2.3899	13598.67	97.20429
289.24893	286.87986	2.3691	13718.46	98.0606
288.80222	286.45296	2.3493	13834.14	98.88749