COMPUTATIONAL SIMULATION OF HEAT TRANSFER ENHANCEMENT BY USING NANOFLUID IN A RECTANGULAR HEATED PIPE

HOUMAN YARMAND

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Abstract

Thermal characteristics of turbulent nanofluid flow in rectangular pipe are numerically investigated. The continuity, momentum and energy equations were solved by means of a finite volume method (FVM). The upper wall of channel is heated at constant heat flux while the bottom wall is symmetry, the left side is subjected to velocity inlet, and the right side of the channel is subjected to pressure outlet. Four different types of nanoparticles Al_2O_3, ZnO, CuO and SiO_2 with different volume fractions in the range of 1% to 5% are used. In this project, effect of different Reynolds number in range of $5000 < Re < 25000$ were investigated to identify the effect on the heat transfer characteristics. The numerical results indicate that water-SiO_2 has the highest Nusselt number compared with other nanofluids. The Nusselt number increases as the Reynolds number is increased .Similarly, the Nusselt number increases as the volume fraction is increased.

Abstrak

Ciri-ciri haba aliran nanobendalir bergelora di dalam paip segiempat yang dikaji dari segi bilangan. Persamaan keterusan, momentum dan tenaga telah diselesaikan dengan menggunakan kaedah jumlah terhingga (FVM). Dinding atas saluran dipanaskan pada fluks haba berterusan manakala dinding bawah adalah simetri, sebelah kiri adalah tertakluk kepada halaju masuk, dan sebelah kanan saluran adalah tertakluk kepada tekanan injap keluaran udara. Empat jenis nanopartikel Al_2O_3, ZnO, CuO dan SiO_2 dengan pecahan isipadu yang berbeza dalam julat 1% hingga 5% digunakan. Dalam projek ini, kesan nombor Reynolds yang berbeza dalam julat $5000 < Re < 25000$ disiasat untuk mengenal pasti kesan ke atas ciri-ciri pemindahan haba. Keputusan berangka menunjukkan bahawa air SiO_2 mempunyai nombor Nusselt tertinggi berbanding dengan nanofluids lain. Nombor Nusselt meningkatkan sebagaimana nombor Reynolds meningkat dan nombor Nusselt meningkat sebagaimana pecahan isipadu meningkat.

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Nomenclature

Roman symbols

Nu Nusselt number

Q Heat Flux (W/m^2)

k turbulence kinetic energy (m^2/s^2)

u velocity (m/s)

x,y,z Cartesian coordinates (m)

p static pressure (N/m^2)

m mass (kg)

c_p specific heat capacity at constant pressure (J/kg.K)

T temperature (K)

Pr Prandtl number

y distance from the wall

SiO_2 Silicon Dioxide

TiO_2 Titanium Dioxide

ZnO Zinc Oxide

Al_2O_3 Alumina

Greek letters

β modeling function

ε dissipation rate of turbulence kinetic energy (m^2/s^3)

μ dynamic viscosity (kg/m s)

μ_t turbulent viscosity (kg/m s)

ρ density (kg/m^3)

τ shear stress (Pa)

κ thermal conductivity (W/m K)

ν kinematic viscosity

φ particle volume fraction

Chapter 1

1. INTRODUCTION

1.1 Research Background

Forced convection heat transfer in a channel had been a topic of interest in lots of research study. Recently, researchers have shown an increased interest in forced convection heat transfer to nanofluids. There are need to enhance the convective heat transfer in thermal systems used in the engineering applications to reduce weight, size and price of heat exchangers. People tried to increase the heat transfer in heat exchangers by using turbulent forced convection flow. There various geometric shape of channel had been studied in the past decade. The more common geometric shapes are circular, rectangular, triangular and square. All these types of channels were used in many engineering applications such as gas cooled reactors, nuclear reactor cooling, cross-flow heat exchanger, gas turbine airfoil cooling and solar air bade cooling system.

In the new decade, there has been a more interest in nanofluids .Nanofluid is a blend of a base pure fluid with suspended metal or nanometal nanoparticles. Nanofluid is prepared from a very small size of particles suspended in pure base fluid or in a blend of fluids as base fluid. The convective heat transfer properties of nanofluids are influenced by the thermo-physical properties of the base fluid and the tiny particles. It also depends on the constituents of the fluid, the shape of the particles, the volume fraction of suspended material and the dimension of these particles. Addition of a small amount of nanoparticles in the conventional fluids increase their thermal conductivity incomparison to the base fluids. The function of the specific nanofluids for a heat transfer application can be recognized by using suitable model of the convective transport in the nanofluid. As the prospect of nanofluid is very encouraging, many studies on convective heat transfer by using nanofluid have been done in recent years. (Rostamani, Hosseinizadeh, Gorji, & Khodadadi, 2010)The nanofluid makes a huge change on the heat transfer enhancement when used as a working fluid. There are a few types of nanoparticles such as AL_2O_3, Sio_2, oxide carbon ,ceramics and the other functionalized nanoparticles. Also, the base liquid for nanofluids are usually ethylene glycol, pure water, oil and other lubricant, polymer solution etc. Different kinds of nanofluids have altered values of thermal conductivity different sustainability, safety and cost.

Most recent research works were done either theoretically, numerically or experimentally. Experimental research works were commonly used to validate the numerical outcomes in the real life. Experimental research is normally more expensive than the other types of research. Hence, some of the researches consider adopting numerical or theoretical method. Theoretical studies are generally associated with the theoretical equations. On the other hand, many of the earlier studies preferred the numerical study by using computational simulation. In addition, due to the high cost and time require for the experimental studies, the numerical method makes a tolerably good and suitable results compared to the other methods.

1.2 Problem Statement and Scope of Study

The forced heat convection heat transfer in channels is appeared in large number of engineering devises with the base heat transfer fluids such as water, air, or ethylene glycol. The limitation in thermal conductivity has always been the primary challenge in the development of energy efficiency of heat transfer fluids and performance of many industrial equipment such as heat exchangers and electronic devices. To solve this problem, there is a strong effort to enhance heat transfer fluids with upper thermal conductivity. In new decade, nanofluids have used more and more for cooling in many industrial applications. These recent generations of heat transfer fluids involve in suspension of nanoparticles having higher suspension stability compared to micrometer or millimeter sized particles. Different types of powder such as non-metallic, metallic and polymeric particles can be added into pure fluids. Thus, the heat transfer properties could be regulated by using nanofluids. The effects and advantages of nanofluids, or nanoparticles on heat transfer were thoroughly investigated. Heat transfer improvements for different types of nanoparticles (AL_2O_3, CuO, ZnO and SiO_2) with the water as base fluid are investigated in the present study. The volume fraction (concentration) of nanoparticles and Reynolds number Re were consider in the range of $0 \leq \varphi \leq 5\%$ and $5000 \leq Re \leq 20000$ respectively.

1.3 Application of the Study

Heat transfer and flow simulation in channel is one of the most commonly studied problems in thermo-fluids area. This is because the different channel configurations are encountered in large number of engineering applications. One of the important configurations is the rectangular channel, which has applications in condensers, gas turbine blade cooling channels, evaporators and nuclear reactor heat exchangers.

The values of heat transfer coefficient in pipe depend strongly on different geometrical parameters and the kind of fluid flow. An improved heat transfer method should be utilized to keep the temperature of electronic devices at lower safe limits. An enhanced heat transfer can be reached by using higher thermal conductivity nanoparticles into the pure liquid within the tube. Conceptually, it is expected that the introducing of the nanomaterial in the nanofluid improves the thermal conductivity and as a result substantially increase the heat transfer characteristics (Das, Choi, Yu, & Pradeep, 2008; Eiamsa-ard & Promvonge, 2008).

Research investigation on a different kind of nanofluids performance is to be concluded. Nanofluids have most of the applications in thermal organizations of consumer and industrial products. New researches have shown the capability of nanofluids to improve the performance of real-world systems and devices such as automatic transmissions. One of the most applications in the field of nanofluids is improving an advanced cooling technology to cool crystal mirrors, nuclear reactor cooling, vehicle equipment cooling and cooling the electronic devices. In addition, nanofluids technology can make better oils and lubricants(Das et al., 2008). Latest nanofluid activity includes the application of nanoparticles in the lubricants to increase the tribological properties of the lubricants, such as antiwear, load-carrying capacity and friction decreasing properties between moving mechanical parts. Furthermore, nanofluids can be used for medical applications, especially in cancer therapy. In future nanofluids can be used to preserve a high gradiant in thermo electrics that would transform waste heat to a valuable electric energy. In buildings, nanofluids can enhance the energy efficiency without the requirement of usage of a greater pump. In addition, in the renewable energy engineering, nanofluids can be utilized to increase the heat transfer from solar collector to storage tanks and enhance the energy output. (Eiamsa-ard & Promvonge, 2008)

1.4 Objective of the study

The objective of this research is to investigate numerically the thermal and hydraulic characteristic of flow in a rectangular tube ;

- to study the effect of using nanoparticles in base fluid on thermal conductivity of the fluid
- to analyze the effect of dissimilar type of nanoparticles (AL_2O_3 , CuO, ZnO, SiO_2) with base fluid (water) flowing in the rectangular tube at different volume fractions in the range of 1% to 5 %
- To investigate the effect of different Reynolds numbers on thermal performance of the base nanofluids at Re in the range of 5000 to 25000

1.5 Dissertation outlines

This thesis contains of five chapters:

Chapter 1 shows an introduction on the current research. This chapter includes research background, problem statement, scope of thesis, application of the research study and objectives of the dissertation.

Chapter 2 involves two major parts. The first part shows a brief introduction about fundamentals of nanofluids such as production of nanoparticles and nanofluids. The second section reviews the effect of using nanofluids on laminar and turbulent fluid flow.

Chapter 3 includes the research methodology and it incorporating six sections. The first section introduces the numerical methodology of the present research. The second part gives a cover view of CFD. The third section gives the CFD forming process. The fourth part involves the physical model and the assumptions of the physical method, boundary conditions and governing equations. The fifth section presents the thermo physical properties of nanofluids. The final section displays the finite volume method (FVM). The sixth part also outlines the fluid flow computation by SIMPLE algorithm.

Chapter 4 gives the results and discussions of this thesis. This chapter is divided into three parts. The first part is code validation and grid independence study. The second section is divided into four parts to show the effect of different nanofluids , the effect of different volume fraction of nanoparticles and finally the effect of various Reynolds number on the thermal characteristics of flow.

Lastly the final Chapter involves the conclusion of this thesis. List of suggestions for future work is stated in this chapter.

Chapter 2

2 LITERETURE REVIEW

2.1 Introduction

Due to increase of energy demands in a wide range of applications such as: automotive, solar heater, heat exchanger for refrigeration, the aim of industries related to heat transfer improvement are finding methods to transfer the heat energy efficiently. Convective mechanism of heat transfer enhancement is one of the main important heat transfer parameters which can be implemented by various approaches. Turbulent regime a away utilizes dispersion mechanism which can make heat transfer augmentation by increase in bulk motion. On the other hand, recently use of nanoparticles has been found as another method of heat transfer enhancement. In spite of many studies have been done on turbulent flow of nanofluid, this concept has remained open-ended until now.

2.2 Fundamentals of Nanofluids

Since conventional fluids are weak in thermal conductivity, they hold a restriction on heat transfer full capacity. Hence it is about one and a half century from

Maxwell (Yu & Choi, 2003) study that and a lot of scientists put their efforts on this basic restriction with suspension of particles in millimeter or micrometer sizes in the different fluids. However, settlement problems of large particle sizes made another disability for this technique. On the other hand, increase of heat transfer surfaces has provoked scientists to turn to enhance thermo-physical properties of fluids. Consequently nanofluid technique associated with design and cost policy of new industries in one hand has become applicable for new kinds of technology like micro-electro-mechanical systems (MEMS). Although the Maxwell's method is not a cutting edge method, it sparked new idea leading to nanoparticles suspension with fewer problems of clogging and settlement in comparison to problem faced by large particles (Das et al., 2008).

2.2.1 Production of Nanoparticles and nanofluids

New nanotechnology has aided the production of nanometallic or metallic nanoparticles of usual crystallite size less than 100 (nm).The optical, mechanical, electrical and thermal properties of nanoparticles are superior to those of convectional materials having coarse grain structure. Recognizing a good opportunity on application of nanotechnology to thermal engineering, Choi(C. Choi, Yoo, & Oh, 2008) conceived the new theory of nanofluids by hypothesizing that it is potential to break down these century –old technical obstacles by developing the unique properties of nano particles. Nanofluids are a modern class of nanotechnology-based heat transfer fluids engineered by decreasing nanometer sized particles with usual length on the other of 1 to 100nm (Das et al., 2008).

The heat transfer enhancement of AL2O3-water nanofluid employed as a coolant for a micro-electronic component was investigated by Nguyen et al (Nguyen et al., 2008)experimentally. According to the results of this study, a significant enhancement of the cooling block convective heat transfer coefficient was observed due to the use of nano powder materials. The heat transfer coefficient was enhanced by a 6.8 vol.% concentration , as much as 40% compared to that of the base fluid. Additionally it was reported that nanofluid with 36nm size provides a greater convective heat transfer coefficient than the other one with 47nm particle size (Nguyen et al., 2008).

Wang et al worked experimentally on heat transfer changes of R 22 refrigerant by adding AL_2O_3 nanomaterial. It was reported that the nanoparticles augmented heat transfer characteristics with the reduction of bubble sizes moving rapidly close the heat transfer surface (S. U. Choi, Xu, & Wang, 2012).

Lee et al.(Lee, Choi, Li, & Eastman, 1999) and Wang et al. (Hwang et al., 2006) investigated the nanoparticle volume fraction influence by using 23 and 24 nm CuO particle on a water base fluid. They observed that the thermal conductivity rises linearly as concentration of particles increases. For example the thermal conductivity ratio increased by 34% with 10% volume fraction. The effect of nano particle volume concentration by using ethylene glycol as base fluid with CuO particles was also carried out. Increase of 15% volume concentration of CuO nanoparticles provides thermal conductivity ratio increase close to 50%.

Xuan et al. (Li, Xuan, Jiang, & Xu, 2005)analyzed Cu-water nanofluid by using particle size of 26 nm and volume concentration of 0.5-2 %. Based on the results of the experiment, the Nusselt number improved proportionately with the Reynolds numbers, and also for the similar Reynolds number the ratio of Nusselt number of CU-water to water varies from 1.06 to 1.39 as the volume fraction of copper nanoparticles increased from 0.5 to 2%.

2.2.2 Thermo Physical Properties of Nanofluid

Since nanofluids have shown the capability of transmiting or conducting heat, people are interested in thermal conductivity which in to be utilized by many applications such as air conditioning and refrigeration. Thus many researches have been focusing in this area. Kulkarni et al. (Kulkarni, Das, & Chukwu, 2006) represented the use of nanofluids lead to a decrease of cogeneration efficiency. They observed that the decrease in specific heat had introduced an impact on the remaining heat recovery from the engine. Moreover, it was found that the efficiency of waste heat recovery of heat exchanger was greater than before with nanofluids because of enhancement of convective heat transfer coefficient. Consequently it was presented that the heat exchanger efficiency increased with nanofluids owing to the higher heat transfer coefficient of nanofluids in comparison to the conventional base fluids.(Kulkarni et al., 2006)

The thermal conductivity of Cu-water nanofluids producing by chemical reduction process was assessed by Liu et al. (Liu, Lin, Tsai, & Wang, 2006) The investigation results pronounced 23.8% improvement at 0.1% volume fraction of copper particles. In fact larger surface area and higher thermal conductivity of cooper nanoparticles were caused by this augmentation. Furthermore, it was represented that thermal conductivity increases with particle volume fraction, although it decreases as time is passed.

Hwang et al (Hwang et al., 2006)reported that the thermal conductivity improvement of nanofluids was remarkably effected by thermal conductivity of both base fluids and nanoparticles. As an example, thermal conductivities of nanofluids produced by multi-walled carbon nanotube and water were of much greater than that of SiO_2 –water nanofluid (Vajjha & Das, 2009).

Yoo et al.(Yoo, Hong, & Yang, 2007) pointed out that surface to volume ratio of nanoparticles could be a main factor influencing the nanofluids thermal conductivity. The thermal conductivity of cu-water nanofluid was measured by Jana et al. (Jana, Salehi-Khojin, & Zhong, 2007). They finally showed a 70% increase in thermal conductivity corresponding to 0.3% Cu nanoparticles fraction. On the other hand, Kang et al. (Kang, Kim, & Oh, 2006) found a 75% thermal conductivity improvement for ethylene glycol with 1.2% diamond nanoparticles from 30 to 50 nm diameter. In contrast to these improvements, there are some researchers found some abnormal results.

Terivedi (Trivedi, 2008) found that decreasing the temperature of a component increases its performance such as reliability. In addition to lowering the junction temperatures within a component, it was sometimes important to reduce the temperature difference between components that were electronically connected in order to gain the optimal performance. Thermal considerations are an important part of electronic equipment because of increased heat flux. Thermal management of electronic components may employ dissimilar heat transfer modes simultaneously and enable an altered level. Nanofluids have shown the capability as a new generation of coolants for many technological services such as automobile thermal management due to their greater thermal conductivities than the base fluids (C. Choi et al., 2008). Thus the nanofluids are made associated with vehicles to meet the heat rejection requirements due to continually increase in trends of high fuel consumption led to energy crisis.

Vahjjha and Das (Vajjha & Das, 2009) presented the dependency of thermal conductivity to both temperature and nanoparticles concentration. They found that, it would be more beneficial if nanofluids are employed in high temperature applications.

Peng et al. (Peng, Ding, Jiang, Hu, & Gao, 2009) observed that the frictional pressure drop on nanorefrigerant boiling flow inside a horizontal smooth pipe is more than that of conventional pure refrigerant, and increases as the volume fraction of nanoparticles increases. The maximum improvement of frictional pressure drop is 20.8% under the experimental conditions.

The effect of particle shape on the thermal conductivity was considered in the investigation of Xie et al. (Xie et al., 2002) . Accordingly the results were compared to the geometric shape of the particle with the similar material and base fluid. The report illustrated that the lengthened particles show better enhancement of the thermal conductivity. Due to the experiments, they suspended 26 nm cylindrical particles of SIC in ethylene glycol based fluid. Consequently they found that at 3% volume fiction, the thermal conductivity ratio of 1.16 and 1.10 was obtained for cylindrical and spherical particles respectively.

Touloukian et al.(Touloukian & Makita, 1970) found experimentally that with the volume fraction increase or the nanoparticles size decrease cause the raise of nanofluid average temperature and decline the heat transfer coefficient ratio of nanofluids to that base fluids. Khanfar et al. (Khanafer, Vafai, & Lightstone, 2003) studied the heat transfer augmentation in a two dimensional enclosure by using nanofluid. The effective thermal conductivity considered in this study has employed the combination of the conventional thermal conductivity theory and a dispersion thermal conductivity while in many of the numerical studies on convective heat transfer; the temperature influence on the thermal conductivity was ignored.The viscosity of the nanofluids measured by Pantzali et al. (Pantzali, Mouza, & Paras, 2009) showed a two time increase relative to pure water. An increase in the pressure drop and therefore increase in the pumping power was observed. They obtained that the pumping power increased about 40% compared to water for a given flow rate. Moreover, they found that for a given heat duty the necessary volumetric flow rates for both water and nanofluid were practically equal, whereas the necessary pumping power in the case of the nanofluid up to two fold higher than the corresponding value for water lead to the higher kinematic viscosity of the fluid.

2.2.3 Application of Nanofluids

The increase in necessity of high heat flow processes has resulted in a considerable demand for cutting-edge methods to enhance the heat transfer. For example, as microprocessors are being smaller and more powerful, and also due to increase in heat flow demands, gradually new challenges in thermal management systems are being important. Likewise, there is an increasing trend for improving the efficiency of existing heat transfer processes. For example in automotive systems heat transfer improvement could result in smaller cooling system, and consequently vehicle weight reduction. Many techniques are available to improve the heat transfer processes used by different systems (Saidur, Leong, & Mohammad, 2011).

The performance and challenge of nanofluids were reviewed by Saidur (Saidur et al., 2011). In this study several applications of nanofluids in solar such as water heating, cooling of electronic devices, and improvement of diesel generator efficiency of chiller, cooling systems of heat exchanger devices, domestic refrigerator-freezers, cooling in machining cooling of nuclear reactor and defense and space were taken into account. Ultimately the future guidelines and problems of nanofluid applications were pronounced. They reported that nanofluids are of a close temperature- dependent to thermal conductivity at very low particle concentrations than conventional fluids. Nanofluids also have shown proper cooling characteristics for welding equipment, automobile engine and high heat-flux devices such as high power microwave tubes and laser diodes arrays. Moreover nanofluids as a coolant could flow through so narrow passage in MEMS without clogging effect. For nuclear application the measurement of nanofluids critical heat flux (CHF) in a forced convection loop is vital. It was shown that an augmentation of chiller heat transfer efficiency by 1% from using nanofluids provides about a saving of 320 billion kWH of electricity or an equivalent 5.5 million barrels of oil per year would be realized in the USA alone. Nanofluids could be used in deep drilling application as well. A nanofluid could be employed for strengthening the dielectric and increasing life of the transformer oil by dispersing nano diamond particles (Saidur et al., 2011; Sridhara, Gowrishankar, Snehalatha, & Satapathy, 2009). According to Kostic (KOSTIC & NETEMEYER) nanofluids were utilized in the following specific areas: Heat-transfer nanofluids, surfactant and coating nanofluids, chemical nanofluids, process/extraction nanofluids, environmental (pollution cleaning) nanofluids, tribological nanofluids and medical nanofluids. During the numerical study, Oliver et al. (Ollivier, Bellettre, Tazerout, & Roy, 2006)found that the ability of nanofluids as a jacket water coolant in a gas spark ignition engine. They considered in their simulations an unsteady heat transfer through the cylinder and inside the coolant flow. They concluded that as a result of higher thermal diffusivity of nanofluids, the thermal signal change for knock detection is increased by 15% like the data obtained for using water solely. Choi (C. Choi et al., 2008; S. U. Choi et al., 2012) presented that in USA by taking a project to investigate fuel saving for the HV industry through the improvement of energy efficient nanofluids lighter and lighter radiators. In fact, deminishing the size and weight of the HV cooling system was a major purpose of the project. On the other hand nanofluids are of the potential to make higher duty coolant and more heat elimination in HVs. A high performance radiator with greater temperature gradiant could decrease the dimensional size by30%.. This could be lead to 10% fuel saving. Buongiorno and Hu (Buongiorno & Hu, 2009) carried out a project in nuclear reactor to find the CHF of nanofluids and mechanisms of heat transfer enhancement. They concluded that the results were essential and state-of-the-art because they expanded the nanofluids CHF database to a flow situation in nuclear applications. The development of various nanofluids for using in water-cooled nuclear systems could lead to an enormous improvement of their financial performance or security restrictions.

2.3 Heat Transfer Enhancement By Using Nanofluid

2.3.1 Heat Transfer Enhancement in Laminar Flow

Haghshenas et al. (Haghshenas Fard, Esfahany, & Talaie, 2010) studied heat transfer efficiency numerically in case of laminar convective heat transfer of nanofluids in a uniformly heated-wall pipe numerically. They employed both the single-phase and two-phase models for estimation of temperature, velocity, and heat transfer coefficient. The two-phase model was reported to be more accurate than the single-phase model. Furthermore, it is found that the heat transfer coefficient of nanofluids increases with the volume fraction of nanofluids and Peclet number raise. On the other hand, it was observed that at the fixed Peclet number of 6500, the heat transfer coefficient for 3%

CuO-Water nanofluid increases by 1.54 times relative to the base conventional fluid. Finally increasing the nanofluid volume fraction from 0.2% to 3%, leads to 27.8% increase in the heat transfer coefficient. At a particular volume fraction, CuO-Water nanofluid is of higher heat transfer coefficient as well. Similarly laminar mixed convection of Al_2O_3-water nanofluid in a horizontal tube with heating at the top half surface of a copper tube was investigated numerically by Allahyari et al.(Allahyari, Behzadmehr, & Hosseini Sarvari, 2011). Two-phase mixture model have been employed to assess hydrodynamic and thermal performance of the nanofluid over a wide range of nanoparticle volume fraction. They have shown that increasing the nanoparticle concentration remarkably enhances the heat transfer coefficient whereas the skin friction coefficient was not considerably influenced. The natural convection in an isosceles triangular enclosure where a heat source locating at the bottom wall and filling with a Cu- Ethylene Glycol nanofluid was simulated by Aminossadati et al. (Aminossadati & Ghasemi, 2011). A heat transfer enhancement was observed by them when the solid volume fraction and Rayleigh number were used. Mahmoudi et al.(Mahmoudi, Shahi, Raouf, & Ghasemian, 2010) simulated a cooling system working with natural convection as a heat sink horizontally installed to the left vertical wall of a cavity filled with Cu-water nanofluid while the left vertical wall was kept at the constant temperature, and the rest ones were kept adiabatic. According to the conclusions of this study the average Nusselt number increases linearly with the increase of solid volume fraction of nanoparticles. Mansour et al.(Mansour, Mohamed, Abd-Elaziz, & Ahmed, 2010) numerically studied a mixed convection flow in a square lid-driven cavity partially heated from below and filled with different water-based nanofluids such as Cu, Ag, Al_2O_3 and TiO_2 to find the effect of particles type and concentration on heat transfer. Finite difference method was adopted to solve the dimensionless governing equations of the problem. They reported that increase in solid volume fraction raises the corresponding average Nusselt number. Moreover, the results depicted Nusselt numbers of base fluid where it was enhanced by the addition of alumina (Al_2O_3) nanoparticles more than that of enhancement done by adding titanium oxide (TiO_2) nanoparticles to the same base fluid. Shahi et al .(Shahi, Mahmoudi, & Talebi, 2011) analyzed the heat transfer enhancement of a nanofluid by simulation of an annular tube driven by inner heat generating solid cylinder. The finite volume method was employed with using SIMPLE algorithm on the collocated arrangement. It has been shown that the averages Nusselt numbers were increasingly depend on the solid concentration. The investigation of the effect of the inclination angle indicated that the maximum average Nusselt number and the minimum level of fluid temperature are obtained at $\gamma=0°$ .In addition, Izadi et al. (Izadi, Behzadmehr, & Jalali-Vahida, 2009) worked on forced convection AL_2O_3-water nanofluid flow in an annular tube by simulation.They reported that the nanoparticle concentration impact on the nanofluid is significant. In general the higher nanoparticle volume fraction is added to base fluid, the more convective heat transfer coefficient is resulted. On the other hand, at the higher Reynolds number in which the momentum and energy increases this dependency on the nanoparticle volume fraction declines.

2.3.2 Heat Transfer Enhancement in Turbulent Flow

Namburu et al. (Namburu, Das, Tanguturi, & Vajjha, 2009) simulated turbulent flow and heat transfer enhancement for three kinds of nanoparticles (CuO, Al_2O_3 and SiO_2) added to both ethylene glycol and water mixture flowing through a circular pipe under constant wall-heat flux condition. They considered the impacts of different nanoparticles and also particles concentration. In this study κ–ε turbulent model proposed by Launder and Spalding (Launder & Spalding, 1974) was adopted. The κ–ε

turbulent model offers two extra equations namely turbulent kinetic energy (κ) and rate of dissipation (ε). The conclusions illustrated that an increase in concentration of nanofluid is led to rise of the average NUSSELT number. Furthermore the result depicted that at a specific Reynolds number of 20000, Nusselt number for 6% CuO concentration increases by 1.35 times more than the base fluid.. Finally for the same concentration of CuO, Al_2O_3 and SiO_2, at a specific Reynolds number, the research results reported CuO nanofluid is of the highest heat transfer rate. Lotfi et al. (Lotfi, Saboohi, & Rashidi, 2010) reported the effect of different models of nanoparticle simulation on forced convection turbulent flow in a circular tube. They made comparisons among three different single-phase, two-phase mixture and Eulerian models. Comparison of the experimental values showed that the mixture model is the most accurate one. Finally they concluded that the rate of thermal enhancement decreases with the increase of nanoparticles volume concentration. Ghaffari et al.(Ghaffari, Behzadmehr, & Ajam, 2010) studied numerically the turbulent mixed convection heat transfer of an Al_2O_3-water nanofluid with particles size of about 28 nm throughout a horizontal curved pipe. They have applied two-phase mixture model for the simulation. The effect of the buoyancy force, centrifugal force, and nanoparticles concentration are assessed in this study. The result illustrated that increases of the nanoparticle volume fraction enhanced the Nusselt number even though its impact on the skin friction coefficient was not remarkable. Nanoparticle concentration increase also strengthened the secondary flow and indirectly influenced the skin friction coefficient. Additionally although at the low Gr turbulent intensity was insignificant, as the Gr increased the effect became more remarkable. The turbulent flow of nanofluids with different volume fractions of nanoparticles flowing through a two-dimensional duct under constant heat flux condition was simulated by Rostami et al (Rostamani et al., 2010). The mixtures of copper oxide (CuO), alumina (Al_2O_3) and oxide titanium

(TiO_2) nanoparticles and water were selected to be bended as a kind of nanofluid. The results show that both the Nusselt number and the heat transfer coefficient of nanofluid are strongly dependent on nanoparticles and increase by increasing of the volume concentration of nanoparticles. In addition the results presented that by increasing the volume fraction, the shear stress increases. The result depicted for a constant volume concentration and Reynolds number, CuO nanoparticles show the most influence to augment the Nusselt number.

Chapter 3

3 METHODOLOGY

3.1 Introduction

In the present study, the Computational Fluid Dynamics (CFD) method is being applied to predict the heat transfer characteristics of nanofluid flow in rectangular. Configuration the finite volume method is employed as the principal behind the CFD method. CFD is utilized to optimize or predict thermo-physical phenomena by post-processing some fundamental data such as: temperature, velocity and pressure.

The finite volume method is utilized by FLUENT solver. In this study, the given model is divided into a finite set of the control volumes called cells by meshing process. After that the governing equations involving mass, energy and momentum equations are solved for a set of control volume systems. All algebraic equations produced by finite volume method from governing differential equations are solved iteratively to find the solution field. The CFD procedures, assumptions, physical model, governing equations, boundary conditions, finite volume method (FVM) and fluid flow computation by using SIMPLE algorithm are presented in the following parts.

3.2 CFD Theories

Computational fluid dynamic (CFD) method is used to solve both fluid flow and heat transfer questions in rectangular channel. CFD is computer based tool for simulate the treatment of system involving heat transfer, fluid flow and other physical operation. It acts by solving the equations of fluid flow over an area of concentration, with known boundary conditions of that zone. The CFD is important in industry because it can solve different system configurations in the specified time with a lower cost compared to experimental options. In this numerical study CFD method was used to find the effect of using nanoparticles in working fluid in a rectangular heated pipe.

3.3 The CFD modeling process

Generally CFD problem can be following by the common procedure as shown in Fig 3-1. As a general rule, after understanding the fluid flow phenomena in the rectangular heated pipe and heat transfer, the CFD method can be classified into some main steps: Pre-Process, solution and post-processing stages. The geometry of case study was created in CFD point of view and then mesh generated. After that physical model, initial condition, boundary conditions and extra appropriate factors were defined in the model for solution. Finally, the result can be gained in the step of post-processing.

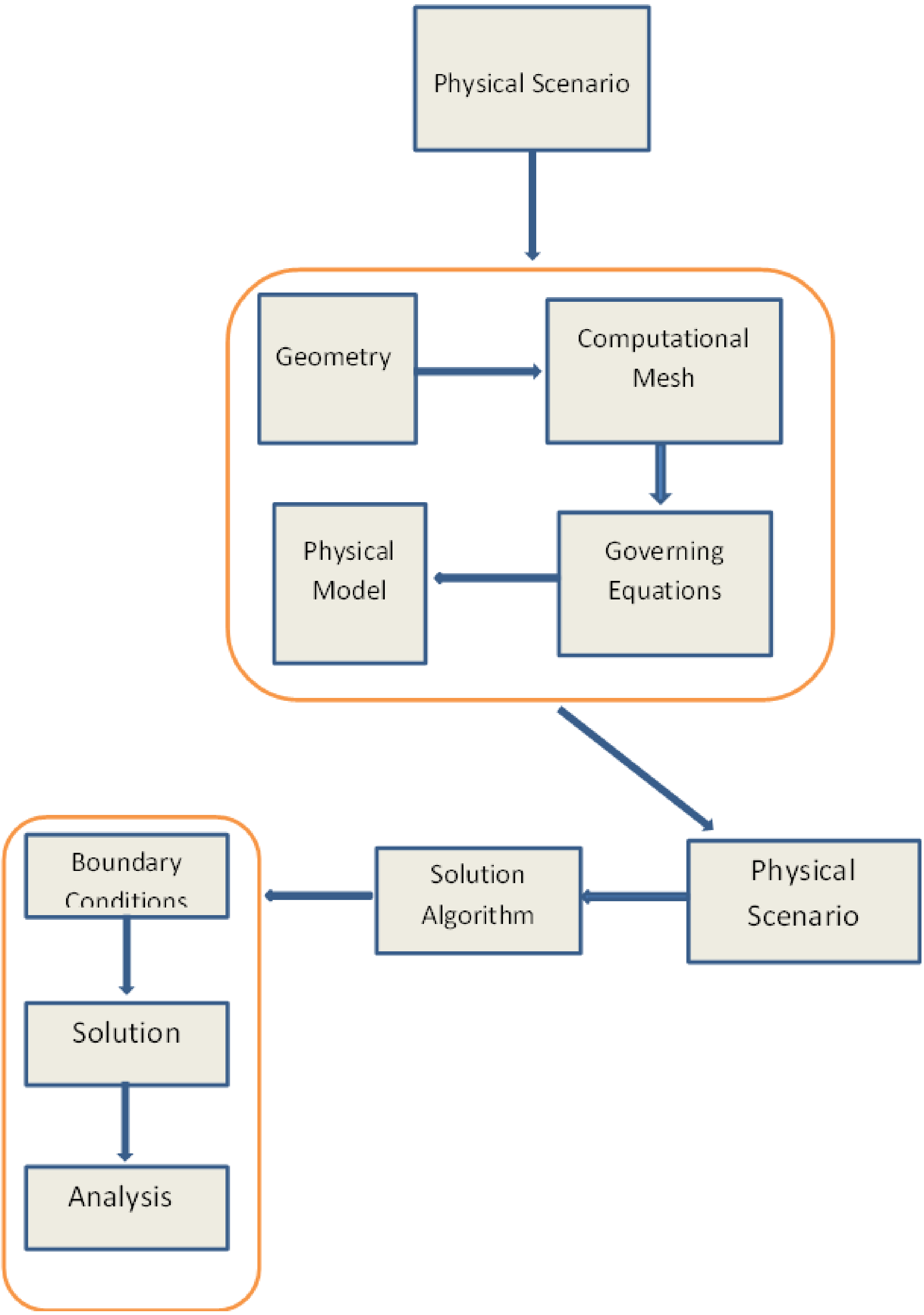

Figure 3-1 – The general modeling process

3.4 Physical Model and Assumption

3.4.1 Physical model

Schematic diagram of a rectangular heated pipe for geometrical model is shown in *Figure 3-2*.A horizontal rectangular pipe of length 2(m) , the channel height is set to be H=10 (mm).To sure a fully developed flow, the $L \geq 10$ H. as shown in *Figure 3-3*.the left side is subjected to velocity inlet and the right side is subjected to pressure outlet, the up wall is subjected to uniform heat flux Q=20000 (W/m^2) , and the down wall is symmetry.

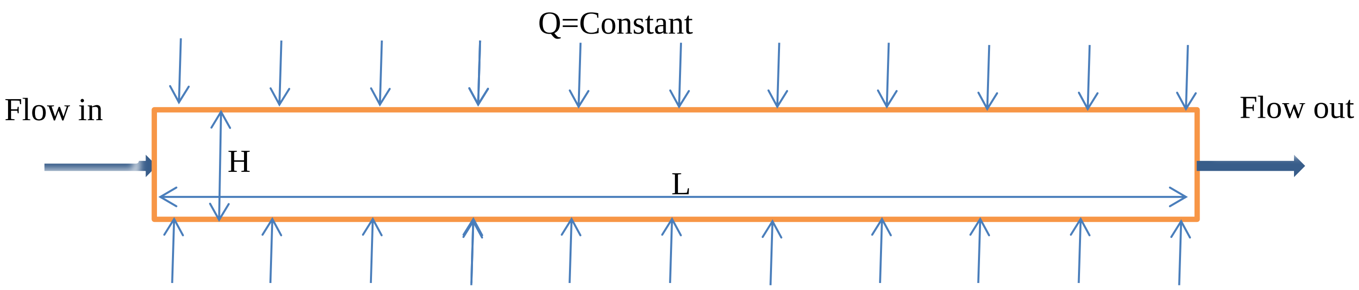

Figure 3-2 : Schematic diagram of the rectangular channel

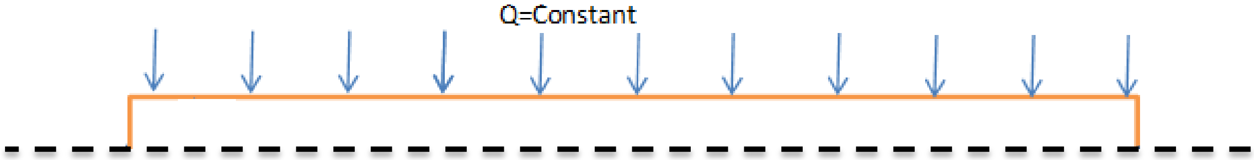

Figure 3-3 : Symmetric diagram of the rectangular channel

3.4.2 Governing Equations

It is very important to set the governing equations (momentum, continuity and energy) to complete the CFD analysis of the rectangular channel. The phenomenon under consideration is governed by the steady two-dimensional from of the continuity, the time-averaged incompressible Navier-Stokes equation and energy equation. In the certain tensor system these equation can be written as :

Continuity equation:

$$\frac{\partial}{\partial x_i}\left(\rho u_i\right)=0 \tag{2.1}$$

Momentum equation:

$$\frac{\partial\left(\rho u_i u_j\right)}{\partial x_i}=-\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_i}+\frac{\partial u_j}{\partial x_i}\right)\right]+\frac{\partial}{\partial x_j}(-\overline{\rho u' u'_j}) \tag{2.2}$$

Energy equation:

$$\frac{\partial}{\partial x_i}\left(\rho u_i T\right)=\frac{\partial}{\partial X_J}\left(\Gamma+\Gamma_t\right)\frac{\partial T}{\partial x_j} \tag{2.3}$$

Where Γ and Γ_t are molecular thermal diffusivity and turbulent thermal diffusivity, they are given by

$$\Gamma=\frac{\mu}{p_{r'}} \text{ and } \Gamma_t=\frac{\mu_t}{p_{r_t}} \tag{2.4}$$

The Reynolds averaged approach to turbulence modeling requires that the Reynolds stresses $\left(-\rho\overline{u_i' u'_j}\right)$ in Eq.(3.2) need to be modeled. The k-ε turbulence model was chosen. A common method employs the Boussinesq hypothesis to relate the Reynolds stresses to the mean velocity gradients Eq.(3.5) :

$$-\rho\overline{U_I' U_J'} = \mu_T\left(\frac{\partial U_I}{\partial X_J} + \frac{\partial U_J}{\partial X_J}\right) \tag{2.5}$$

The turbulent viscosity term μ_t is to be computed from an appropriate turbulence model. The expression for the turbulent viscosity is given as Eq. (3.6).

$$\mu_t = \rho c_\mu \frac{k^2}{\varepsilon} \tag{2.6}$$

The modeled equation of the Turbulant Kinetic Energy (K) is written as Eq.(3.7)

$$\frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j}\left[(\mu + \frac{u_t}{\sigma_k})\frac{\partial k}{\partial x_j}\right] + G_k - \rho\varepsilon \tag{2.7}$$

Similarity the dissipation rate of TKE, ε is given by the following equation Eq. (3.8) :

$$\frac{\partial}{\partial_x}\left(\rho\varepsilon u_i\right) = \frac{\partial}{\partial x_j}\left[(\mu + \frac{u_t}{\sigma_k})\frac{\partial \varepsilon}{\partial x_j}\right] + c_1\frac{\varepsilon}{k}G_K - C_{2\varepsilon}\rho\frac{\varepsilon^2}{K} \tag{2.8}$$

Where G_K is the rate of generation of the TKE while $\rho\varepsilon$ is the destruction rate. G_K is written as equation (3.9) :

$$G_K = -\rho\overline{U_I' U_J'}\frac{\partial U_J}{\partial X_I} \tag{2.9}$$

The Reynolds number, the Nusselt number, the friction factor, and the thermal augmentation factor are presented by the following relations(Incropera & DeWitt, 1985) equations (3.10 – 3.12)

$$\mathrm{Re}_{av} = \frac{u_{av} D_h}{v} \tag{2.10}$$

$$Nu_{av} = \frac{hD_h}{k} \tag{2.11}$$

Where T_W and T_b are the wall temperature and bulk temperature.

$$f = 2\triangle p \frac{D_h}{L} \frac{1}{\rho u_{av}{}^2} \tag{2.12}$$

Re_{av} is the average Reynolds number, Nu_{av} is the average Nusselt number of channel and f is friction factor for channel.

3.4.3 Boundary Conditions

The boundary conditions for this study work are specified for the computational domain as shown in Fig 3.2 and Fig 3.3. These figures show the rectangular pipe model which is selected for the present study. The top wall is subjected to a constant heat flux of 20000 (W/m^2) and the bottom wall is symmetry to the top wall. The left side is subjected to velocity inlet and the right side is subjected to pressure outlet. The turbulent model is very important to accommodate the flow behavior of each application to reach an exact estimate in the rectangular pipe,the standard $k-\varepsilon$ turbulent model ,the Renormalized Group $k-\varepsilon$ turbulence method were used. The time independent incompressible Navier-Stokes equations and the turbulence model analysis were solved using finite volume model. To evaluate the pressure field, the pressure-velocity coupling algorithm SIMPLE (Semi Implicit Method for Pressure-Linked Equations) was chosen the inlet and the uniform velocity profile has been imposed. The turbulent intensity was 1% at the inlet. The solutions are considered to be converged when the normalized residual values reach 10^{-5} for all parameters.

3.4.4 Thermo Physical Properties of Nanofluids

In order to find numerical simulation for nanofluids, the effective thermo physical properties of nano fluids must be determined first.in this study, the nanoparticles being used are AL_2O_3, ZNO, CuO , SiO_2.generally the required properties for numerical simulations are effective thermal conductivity ($\boldsymbol{k}_{eff}$),effective dynamic viscosity, effective mass density (ρ_{eff}), and effective specific heat (C_{peff}). Regarding these, the effective properties of mass density, specific heat, thermal conductivity, and viscosity are truly calculated base on the mixing theory.

The density of nanofluid, ρ_{nf} can be obtained from the equation (3.13) , (Hoffmann & Chiang, 2000).

$$\rho_{nf} = (1-\phi)\rho_f + \phi\rho_{np} \tag{3.13}$$

Where ρ_f and ρ_{np} are the mass densities of the based fluid and the solid nanoparticles, respectively.

The effective heat capacity at constant pressure of nanofluid, $(\rho\ C_p)_{nf}$ can be calculated from the equation (3.14),(Bianco, Chiacchio, Manca, & Nardini, 2009) .

$$\left(\rho\ C_p\right)_{nf} = (1-\phi)(\rho\ C_p)_f + \phi\ (\rho\ C_p)_{np} \tag{3.14}$$

Where $(\rho\, C_p)_f$ and $(\rho\, C_p)_{np}$ are heat capacities of the based fluid and the solid nanoparticles

The effective viscosity equation (3.15 -3.15.1) can be obtained by using the following mean empirical correlation (Bianco et al., 2009).

$$\mu_{eff} = \mu_f * \frac{1}{(1-34.87\,(d_p/d_f)^{-0.3} * \varphi^{1.03})} \tag{3.15}$$

$$d_f = \left[\frac{6M}{N\pi\rho_{fo}}\right] \tag{3.15.1}$$

Where: M is the molecular weight of base fluid, N is the Avogadro number = $6.022*10^{23}$ mol^{-1} and ρ_{f0} is the mass density of the base fluid calculated at temperature T_0=293K.

By using Brownian motion of nanoparticles in channel, the effective thermal conductivity can be obtained by using the following mean empirical correlation equations (3.16 – 3.17), (Vajjha & Das, 2009) :

$$K_{eff} = k_{static} + k_{Brwnian} \tag{3.16}$$

(3.17)

$$k_{brawnian} = 5*10^4 \beta\phi\rho_f c_{p,f} \sqrt{\frac{kT}{2\rho_{np} R_{np}}}\, f(T,\phi) \tag{3.18}$$

Where :

Boltzman constant : $k = 1.3807*10^{-23} \quad J/K$

Table 3-1 shows the thermo physical properties of water and nanoparticles at 3% concentration and Re= 15000.

Table 3-1: *The thermo-physical properties of water and different nanoparticle at T=300K, φ=3% and Re =15000*

Property	Al2 O3	CuO	SiO2	ZnO	WATER
Density, ρ (kg/m^3)	1090	1160	1030	1140	998
Specific heat, cp (J/kg.K)	3807.73	3570.00	3960.00	3640.00	4182.00
Thermal conductivity, k(W/m.K)	0.7000	0.7003	0.6199	0.6880	0.5984
viscosity, μ (Ns/m^2)	0.0014087	0.0014087	0.0014087	0.0014087	0.0010020
VELOCITY (m/s)	0.9720	0.9080	1.0200	0.9300	0.7536

Table 3-2 shows the thermo-physical properties of SiO2 for different range of concentrations from 1% to 5%

Table 3-2 *:The thermo-physical properties of SiO_2 with different concentrations*

Concentration	Density	Viscosity	Specific Heat	Conductivity
0%	998	0.001002	4182	0.5984
1%	1010.00	0.0011057	4110.00	0.6065718
2%	1020.00	0.0012378	4030.00	0.6132
3%	1030.00	0.0014087	3960.00	0.6198846
4%	1050.00	0.00164	3890.00	0.62662594
5%	1060.00	0.0019566	3820.00	0.6334241

Table 3-3 shows the velocity of SiO_2-Water corresponding to a wide range of ϕ from 1% to 5% and Reynolds numbers from 5000 to 25000.

Table 3-3: *The measurement of velocity for corresponding Re and φ*

SIO_2 (VELOCITY)						
Re	Water	1%	2%	3%	4%	5%
5000	0.25120216	0.2740	0.3030	0.3410	0.3910	0.4620
10000	0.50240433	0.5470	0.6050	0.6810	0.7820	0.9240
15000	0.75360649	0.8210	0.9080	1.0200	1.1700	1.3900
20000	1.00480866	1.0900	1.2100	1.3600	1.5600	1.8500
25000	1.25601082	1.3700	1.5100	1.7000	1.9600	2.3100

3.5 Finite Volume Method (FVM)

The finite volume method is a discretization method which was well suited for the numerical simulation of various types (elliptic, parabolic, and hyperbolic) of conservation laws. Some of the important features of finite volume method were similar to those of finite element method. It may be used on arbitrary geometries, using structured or unstructured meshes and it leaded to robust schemes. Feature is the local conservative numerical fluxes from one discretization cell to its neighbor. Finite volume method is locally conservative because it was based on a balance approach: local balance was written on each discretization cell which often called "control volume" by divergence formula an integral formulation of the fluxes over the boundary of the control volume was then obtained. The fluxes of boundary were discretized with respect to the discrete unknown. The principle of the finite volume method was given a number of the discretization points which may be defined by mesh, to assign one discrete unknown per discretization point and to write one equation per discretization point. At each discretization point the derivatives of the unknown were replaced by finite volume through the use of Taylor expansions(Vajjha & Das, 2009) .

Quantity ϕ as a conservative property is taken into account and the velocity field and the fluid density are supposed to be known. The consecutive equations based on steady convection and diffusion could be offered as general form of transport equation for property ϕ as below:

$$div\ (\rho\varphi u) = (\Gamma\nabla\varphi) + S_{\varphi} \tag{3.19}$$

Eq.3.16 is utilized as a beginning step for computational process in FVM to predict thermo-fluid phenomena. The channel geometry is divided into so many finite control volumes known as cells. This procedure is called as the geometrical discretization of the domain (Quatember & Mühlthaler, 2003) The different steps of general finite volume method used for prediction of steady convective-diffusive problems should be done one after another in this study as follows:

i) The transport equation for momentum, mass and energy are used to each cell, computational domain and discretized as depicted in *Figure 3-4*.

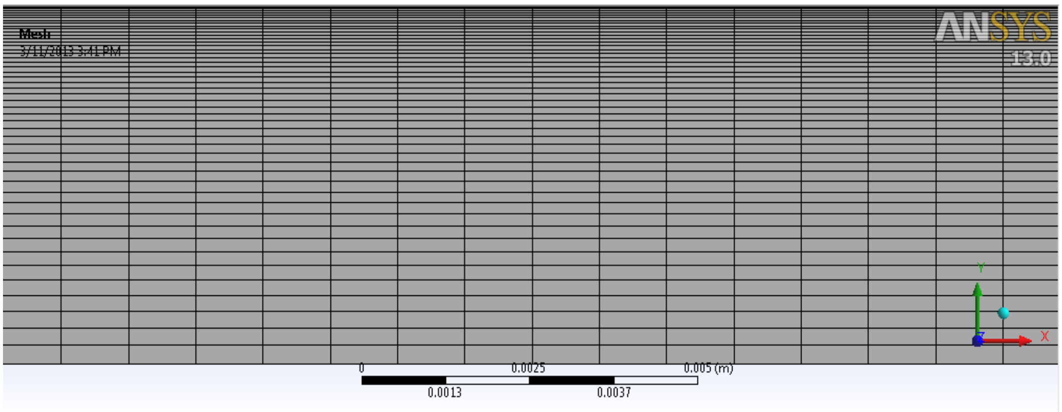

Figure 3-4: Rectangular channel into finite set of control volumes (mesh)

ii) By integrating from Eq.(3.9) over two-dimensional control volume one can write:

$$\int_v div\,(\rho\varphi u) = \int_v (\Gamma\nabla\varphi)dV + \int_v S_\phi dV \tag{3.20}$$

The first term on the left side is called convection while the first term on the right side is called the generation term. The middle term called diffusion is re-written as surface integrals over bounding surface of control volume based on Gauss divergence theorem. For vector *b* this theorem can be presented as:

$$\int_v div\,(\rho\varphi u) = \int_A div\,(m.bdA) \tag{3.21}$$

The physical interpolation of *m.b* is the component of vector *b* in the direction of vector volume is equal to component of *b* in the direction normal to the surface which bounds the volume summed integrated over all boundary surface *A*.

iii) Eq.(3.15) could be resulted by utilizing Gauss divergence theorem as equation (3.22).

$$\int_A div\,(\rho\varphi u)dA = \int_A m.div\,(\Gamma\nabla\varphi)dA + \int_v S_\phi dV \tag{3.22}$$

iv) Then for cell p as shown in Figure *3-5*, each transport equation is converted into algebraic form.

$$\int_f^{N_{faces}} \rho_f . \varphi_f . u_f . A_f = \sum_f^{f\ N_{faces}} \Gamma_f \nabla \varphi_f A_f + S_\varphi \Delta V \tag{3.23}$$

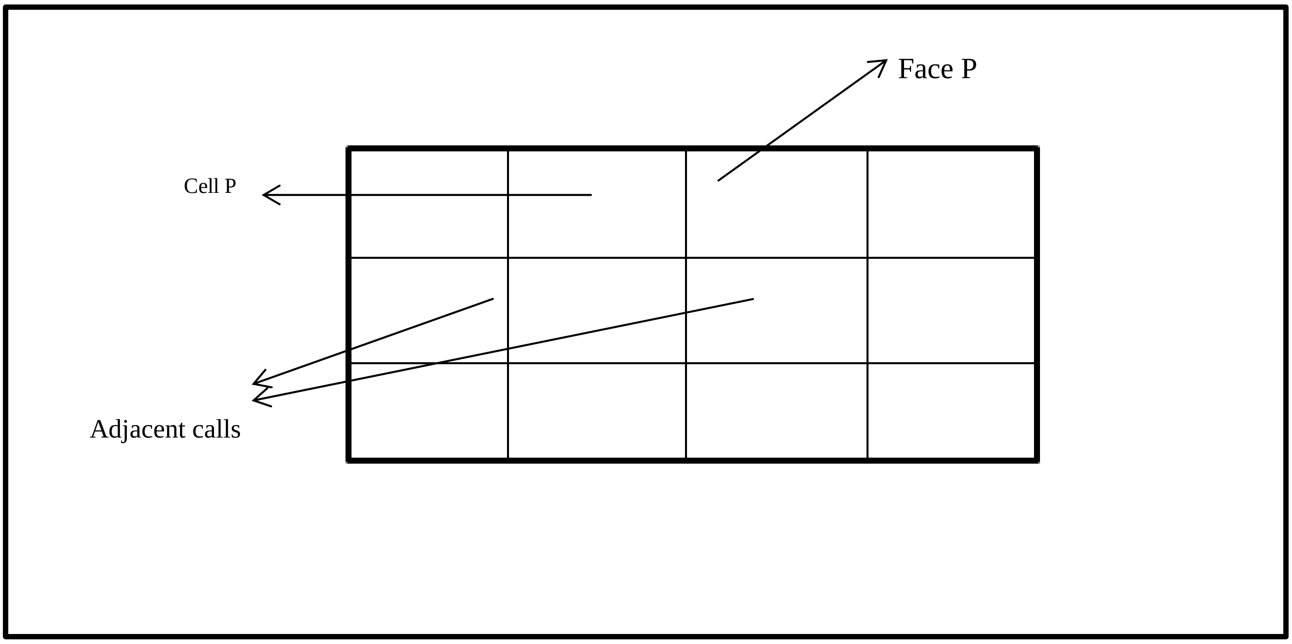

Figure 3-5 : A part of two dimensional grids

v) Owing to the data at both centers and faces are required, all material properties and velocities should be stored at cell centers whereas face values are calculated by interpolation within the local and adjacent cell values.

vi) Since field variable those which stores at cell centers must be employed by interpolation for the face control volume data calculations, there are several interpolation algorithm used for convection parameter which are first order upwind, power law, second order upwind and quadratic upwind interpolation (QUICK). Due to the second order upwind as the easiest way for convergence and proper for Reynolds numbers of 5000-20000 according this study, it has been selected to interpolate the conversion parameter.

vii) For evaluation of diffusive flux, velocity derivatives and higher order discretization schemes the gradient of solution variables are needed. In this study the gradients of solution variables at cell centers are calculated by Green Gauss method and also Taylor series expansion is utilized for Gradients of solution variables at faces.

viii) It is clear that the discretized equation could be presented as bellow equation (3.24).

$$a_p\varphi_p + \sum_{nb} a_{np}\varphi_{np} = b_p \qquad (3.24)$$

Coefficients a_b and a_{np} are typically functions of solution variables (nonlinear and coupled). The following steps are followed to develop a numerical method for calculating the fluid flow velocities as shown in the section 3.6.

3.6 Fluid Flow Computational by SIMPLE Algorithm

The convection of a scalar variable ϕ depends on the magnitude and direction of the velocity field. The velocity field was assumed to develop the methods in the previous sections. However, the velocity field generally unknown and emerges as part of overall solution process along with all other flow variables. In this section, the popular strategies for computing the entire flow field will be discussed. The pressure gradient term, which forms the main momentum source term in most flows of engineering importance, has been written separately to facilitate the discussion. Solutions of Eq.3.1 and Eq.3.2 are introduced to two new problems. These problems are the convective terms of the momentum equations which contain nonlinear and both equations are intricately coupled. This is because every component appears in each momentum equation and continuity equation(Vajjha, Das, & Kulkarni, 2010) The most complex problem to solve is the role played by the pressure. It appears in both momentum equations, but there is evidently an (transport or other) equation for pressure. Therefore, both problems associated with the nonlinear in equation set and the pressure-velocity linking could be resolved by utilizing an iterative solution strategy such as the SIMPLE algorithm of Toro. SIMPLE is one of the algorithms avertible in FLUENT and was chosen in the current study, because it allows faster convergence for turbulent flow with physical models employed and it is robust(Corcione, 2010).

The Semi-Implicit Method for Pressure-Linked Equations (SIMPLE) is basically a Guess and correction procedure for pressure calculation. Generally in this procedure, the momentum equation through entire domain is discretized by assuming a known pressure distribution and the velocity field for *u*,*v* and *w* is guessed. Then the obtained velocity field must satisfy the continuity equation. If the velocity satisfies continuity equation the pressure field assumed is correct otherwise a new pressure distribution should be proposed. This algorithm is continued iteratively until this goal is achieved. The SIMPLE algorithm is shown as below:

i) Purposing the pressure field P^*. A suitable pressure field P^* should be considered for the whole domain. This pressure distribution can be the estimation for finding the correct pressure distribution achieved iteratively.

ii) The momentum equation solution for gaining dimensionless velocity of u^*, v^*, and w^* from the momentum equations due to the guessed pressure distribution. As mentioned before., the velocity distribution, after solving the momentum equations are calculated according to the proposed pressure distribution. This velocity field will recommend an estimate for the right velocity field.

iii) Estimate pressure relation *p'* for the whole domain. In fact *p'* as a variable will be defined temporarily. This variable is compared by a semi-linear algebraic system of equation for the whole domain , so that the coefficients for the linear system of equations are created by mass conservation equation integration over control volume according to the assumed velocity measures computed in step no. 2. The major association of this temporary variable is to employ the correct pressure and velocity field.

iv) Update the pressure field p. The field of pressure p is updated by the addition of the guessed pressure P^* and the pressure formula *p'* as :

$$P=P'+P^* \qquad (3.24,1)$$

As mentioned before, P' is equal to zero, if the proposed pressure field can meet the continuity equation requirements.

v) Update u, v, and w from their estimated measures according to the velocity correlation:

$$u_i = u_i^* + \frac{area}{a_p}\,(\Delta P') \qquad (3.25)$$

where $(\Delta P')$ offers the pressure (according to step 3) applying over each velocity component and a_b is the nodal point coefficient. As can be deducted from Eq.3.20, the precise velocity field is the addition of the started velocity field (step 2) and a coefficient that represents the driving potential for fluid motion (second term on the right hand side).

vi) Solution of any other φ properties. Once the velocity field calculated any other scalar variable such as temperature, concentration, etc. including in the problem could be achieved.

vii) The flow field solution should be assessed to meet the requirements of the mass conservation equation. In the step pressure and velocity field are examined for satisfying the continuity to reach convergence. Convergence guaranties that the conservation has been gained and changes in solution variables has become minor from iteration. Otherwise the algorithm is repeated and the right pressure P in step 1 is applied as the new pressure field P^* suggestion and the steps are recurrent until convergence. During the solution of momentum equation it should be noticed to determine the nodal location for all the variables in the equation. Figure 3.6 illustrates the flow diagram of the solution process (Versteeg & Malalasekera, 2007).

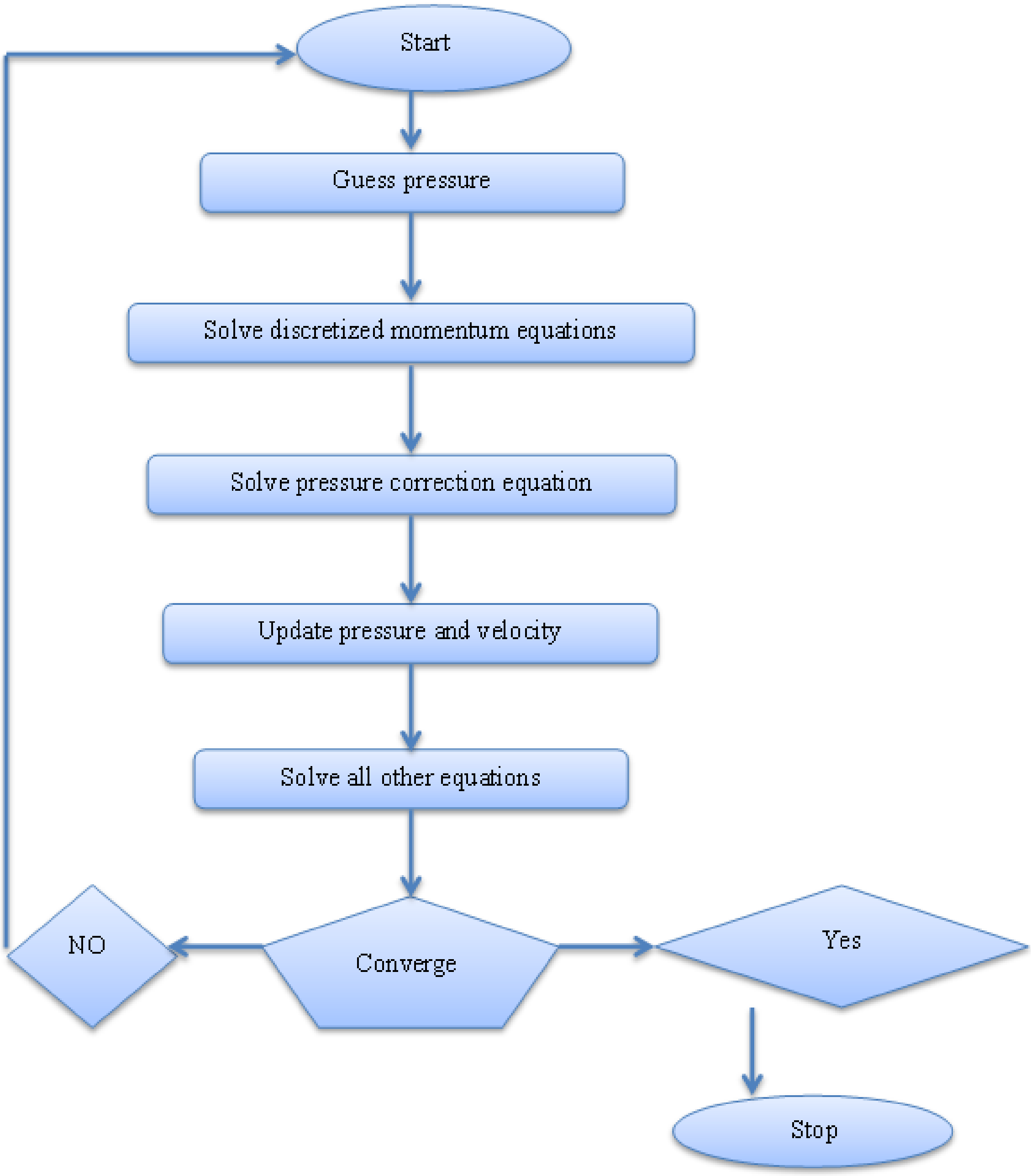

Figure 3-6 :*SIMPLE algorithm fiow chart*

3.7 Summary

The dimensionless governing conversation equations corresponding to the studied geometry are based on the fundamental governing equations in CFD were presented. The numerical computation was carried out by solving the governing conservation equations by using FVM. The proper boundary conditions have been set carefully as the setting are extremely important in CFD. The pressure-velocity coupling in the governing equations is achieved by using the well-known SIMPLE algorithm method for numerical computations. The calculation methods for all the effective nanofluids properties are essential in order to carry out the simulations that utilize nanofluids. The results in terms of isotherms, streamlines and Nu number and its discussions for rectangular channel are elaborated in the next chapter.

Chapter 4 :

4 Result & Discussions

4.1 Validation of Present Simulation

According to the earlier proposed geometry, the fixed duct height 10 mm ,length 2 m and a width of 1 m. The fluid that going into the entrance of duct with a constant inlet temperature,T_0 of 300 K and at uniform axial velocity,V_0 .The Reynolds number range was varied from 10000 to 40000 .To ensure that the computational model is fully justified, the numerical results obtained were compared with those of theoretical datas discussed before. In order to do so, the NU number from the Dittus Boelter equation is evaluated from equation (4.1):

$$Nu = 0.023 \mathrm{Re}^{0.8} \mathrm{Pr}^{0.4} \qquad (3.1)$$

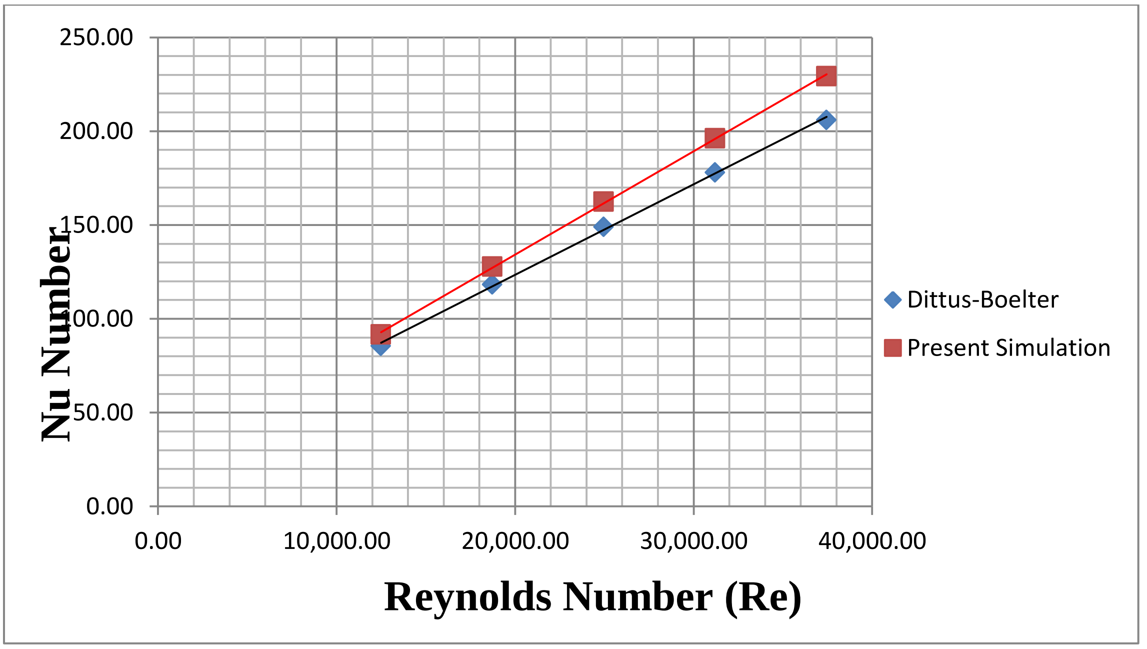

Figure 4-1: Comparison of Nusselt number for water with Dittus-Boelter formula in turbulent regime

Figure ***4-1*** displays the comparison of Nusselt number from Dittus-Boelter formula and computed value from the present simulations. A moderate agreement is observed with maximum deviation and average deviation of computed values from theoretical equation being 23.38 and 9.61%, respectively because the errors obtained could possibly be happened due to unpredictable variation in thermal conductivity, assumptions of empirical equation (Dittus-Boltter) and simulation errors.

Next, the comparison of Nusselt number from the water could be compared with the Nu value obtained from the empirical Gnielinski equation (4.2) .

$$Nu = \frac{(f/8)(\mathrm{Re}-1000)\,\mathrm{Pr}}{1+12.7(f/8)^{\frac{1}{2}}(\mathrm{Pr}^{\frac{2}{3}}-1)} \qquad (3.2)$$

Where f could be calculated from equation (4.3).

$$f = (0.790 Ln\,\mathrm{Re}_D 1.64)^{-2} \qquad (3.3)$$

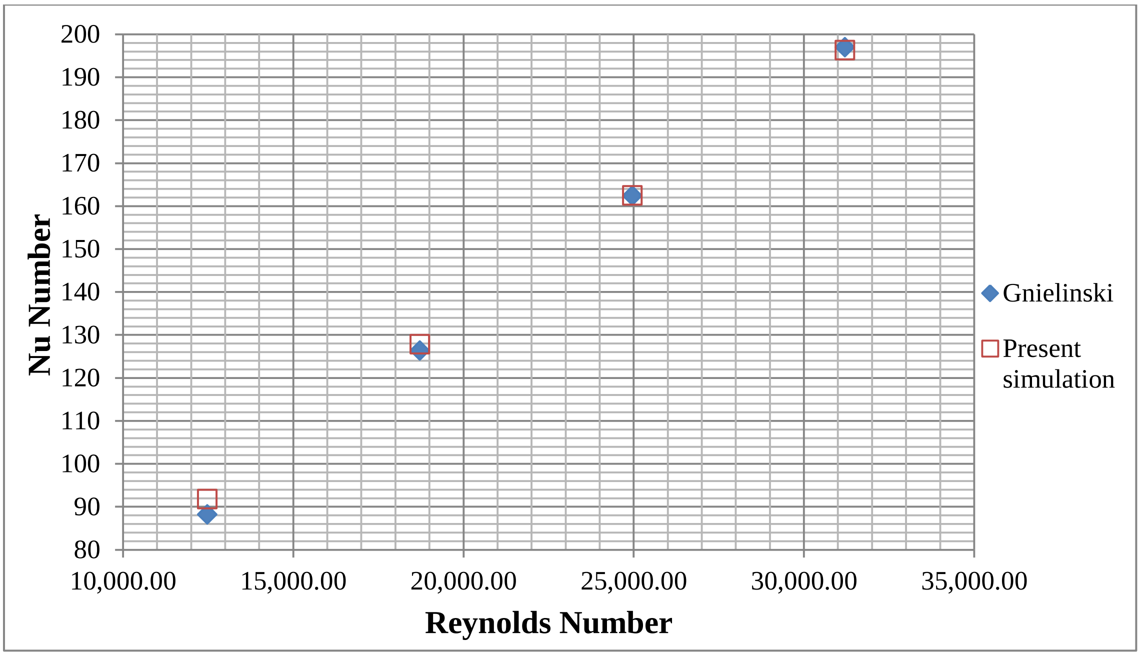

Figure 4-2: Comparison of Nusselt number for water with Ginielinski formula in turbulent regime

Figure 4-2 shows The maximum deviation happened at the first selected Reynolds number and the average error are 3.59 and 0.45% error respectively.

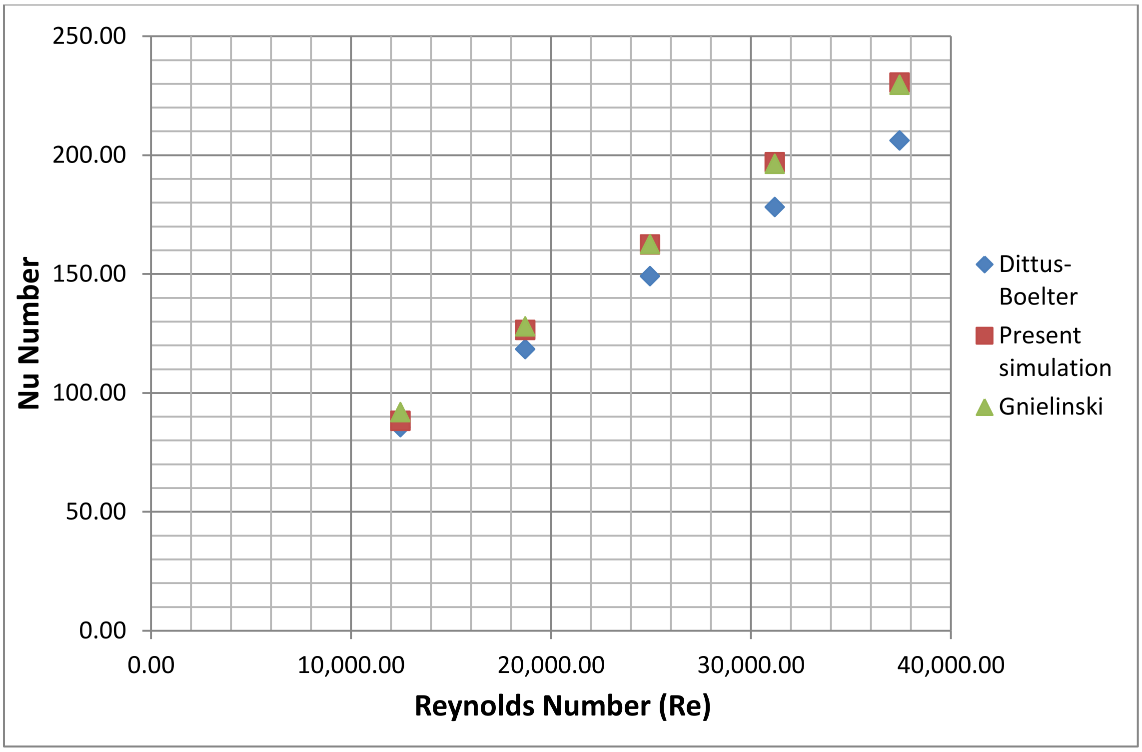

Figure 4-3 :Comparison of numerical Nu for water with Gnielinski correlation and Dittus-Boelter formula in turbulent regime

From Figure ***4-3***, at the In lower range of Reynolds numbers the agreement of predicted Nusselt number is very close to the data obtained from the two correlation when compared at higher range of Reynolds numbers. It can be seen that numerical Nusselt number are in very good agreement with that of the correlation values, from the Gnienlski and Dittus – Boelter correlations, only slight deviation occur which being considered negligible for validation. however the simulated data fits much accurately with the data of Gnielinski as it is more accurate correlation and fits well with the experimental data incomparision to Dittus-Boelter.

4.2 Effect of Nanoparticle Volume Concentration on the Nusselt Number and Heat Transfer Coefficient

*Figure 4-4*shows the effect of different volume concentrations of SiO_2 nanoparticles on the Nusselt number. In single phase simulation, the nanofluid being chosen SIO_2 – water. In this case, the Reynold number was selected 15000 constant for observation with different volume concentrations, such as 1% , 2% , 3% , 4% and 5% . The results show that with the increase concentration of Nanoparticle in the base fluid the Nusselt number increases.

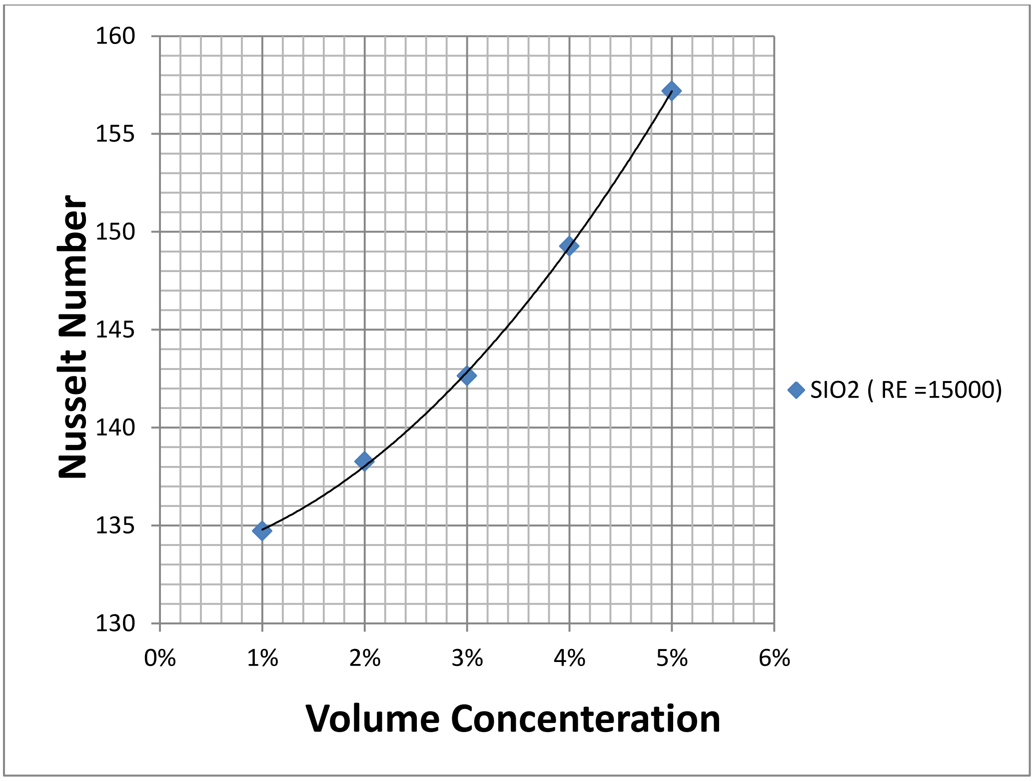

Figure 4-4 :*Effect of using different volume concentrations of nano particles on Nusselt number*

Figure 4-5 shows the effect of different concentration of SiO_2 nano particle on heat taransfer coefficient of nano fluid. The Reynolds number and concentration are chosen same as before. The results show as volume concentration increases from 1% to 5%, the heat transfer coefficient also increases.

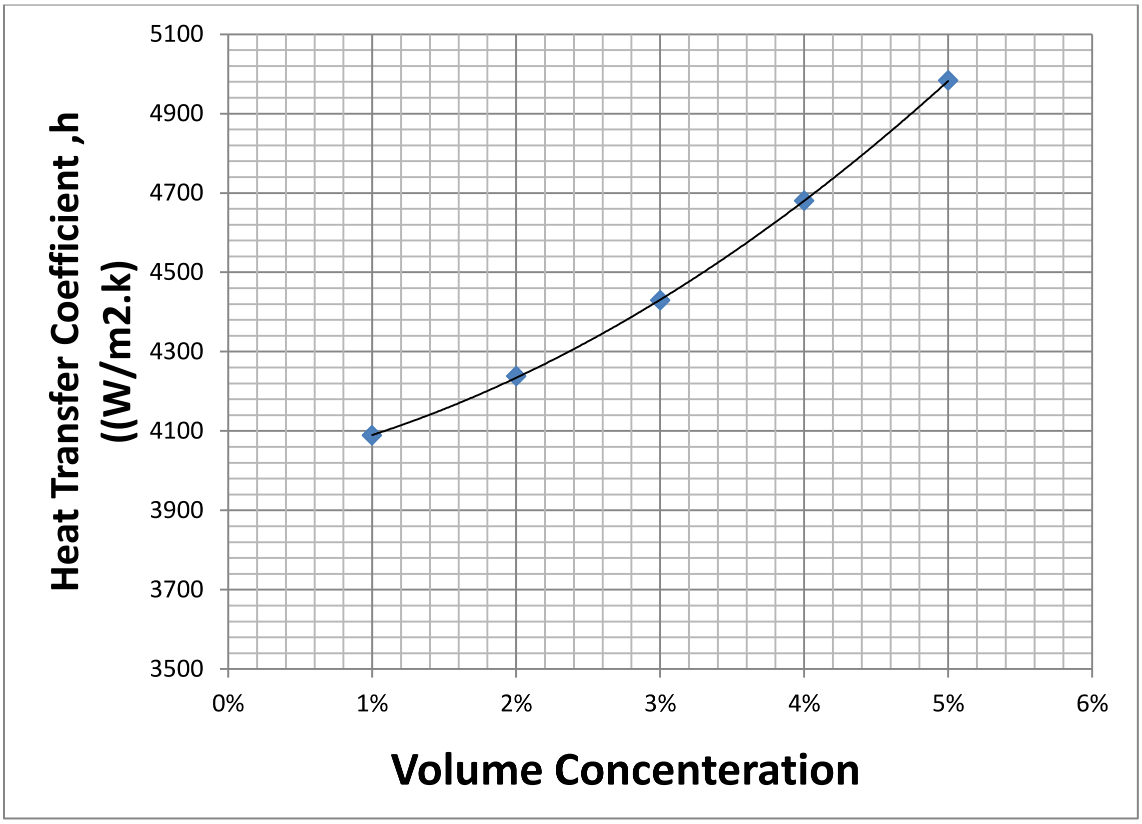

Figure 4-5 :Effect of using different volume concentrations of nano particles on Heat Transfer Coefficient

4.3 The effect of Different types of Nanoparticles on the Nusselt Number and Heat Transfer Coefficient

In this part, four various types of nanoparticles such as (CuO,SiO_2, ZnO and Al_2O_3) and pure water as the base liquid are examined. In order to find the effect of different nanoparticles on the heat transfer improvement all other parameters of the system must be fixed (Re=15000 and 3% concentration of nanoparticles). The Nusselt numbers for different nanofluids are shown in *Figure 4-6*. This figure shows, that all the nanofluids provide the higher Nusselt number compared to pure water. But in the case of comparison between the four types of nanofluids, it is clear that nanofluid with SiO_2 has higher Nusselt number, followed by Al_2O_3 ,ZnO and CuO respectively. This result can be attributed to the fact that the high values of inverse of thermal properties of Water-SiO_2 compared to smaller inverse thermal conductivity properties of other particle type nanofluids.

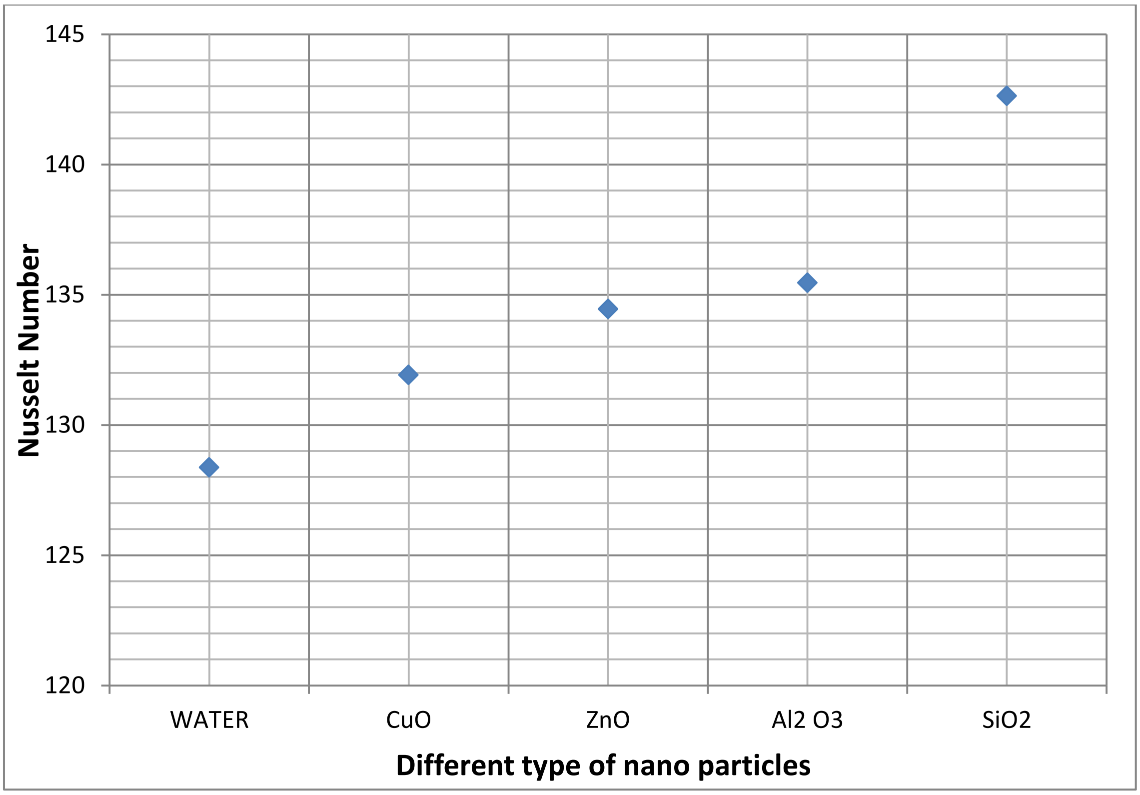

Figure 4-6 : *Effect of different types of Nanofluids on Nusselt Number*

Figure 4-7 illustrations effect of using different types of nano particle (CuO, ZnO, Al_2O_3 and SiO_2) on heat transfer coefficient of Nanofluids with water as a base fluid. The results for all the particles are in Reynolds number 15000 and 3% volume concentration. The results show all the nano fluids have higher heat transfer coefficient than that of pure water. The highest heat transfer coefficient is for AL_2O_{3}-water nanofluid. In addition, due to the impact of effective thermal conductivity on calculation of heat transfer coefficient, the figure 4-6 cannot show the accurate effect of different types of nano particles on heat transfer enhancement. For example, although SiO_2 has the highest Nusselt number, it has the lowest heat transfer coefficient.

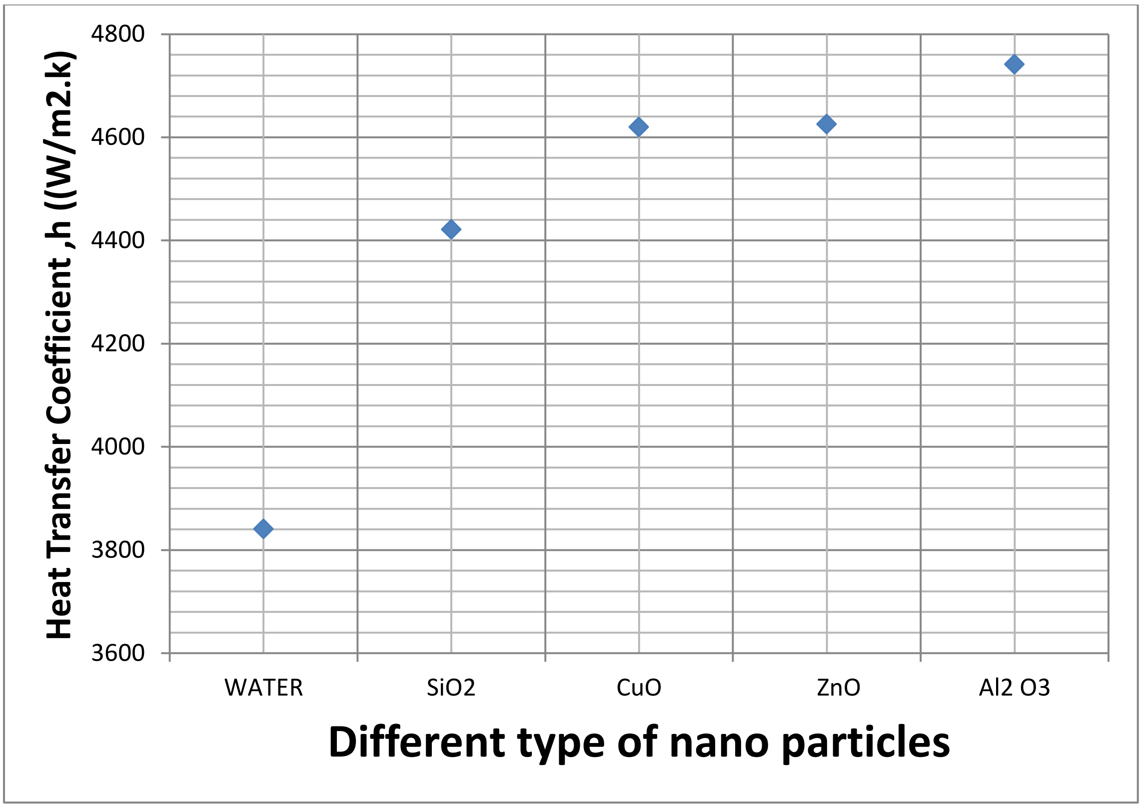

Figure 4-7:Effect of different types of Nanofluids on Heat Transfer Coefficient

Figure 4-8 shows the contour of temperature for water and figure 4-9 shows the temperature contour for AL_2O_3 -water. The wall temperature for pure water is more than the wall temperature for AL_2O_3 nanofluid. This difference is because of nano particle suspension in pure water increasing the heat transfer coefficient of fluid and leads to more heat transfer from wall to fluid.

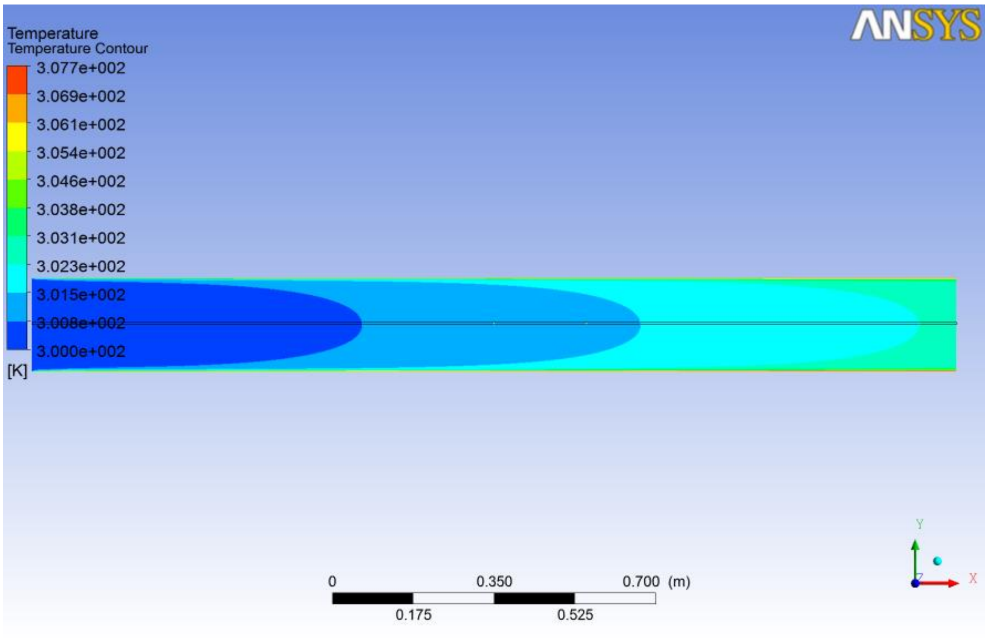

Figure 4-8 : Temperature Contour for Pure Water

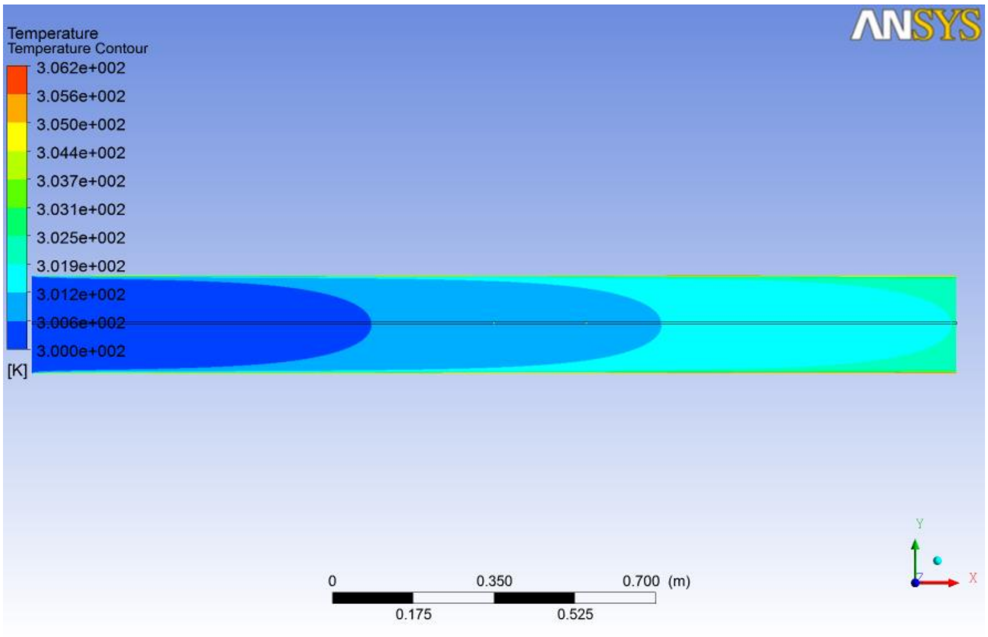

Figure 4-9 : Temperature Contour for Al_2O_3 -Water

4.4 The Effect of Reynolds Number of Nanofluids on the Nusselt number and Heat Transfer Coefficient

This study was done at Reynolds number in the range of 5000 – 25000. *Figure 4-10* can be shows that with the increase of Reynolds number the Nusselt number also increases. The Reynolds number Re=25000 provided the highest Nusselt number rather than other Reynolds numbers.

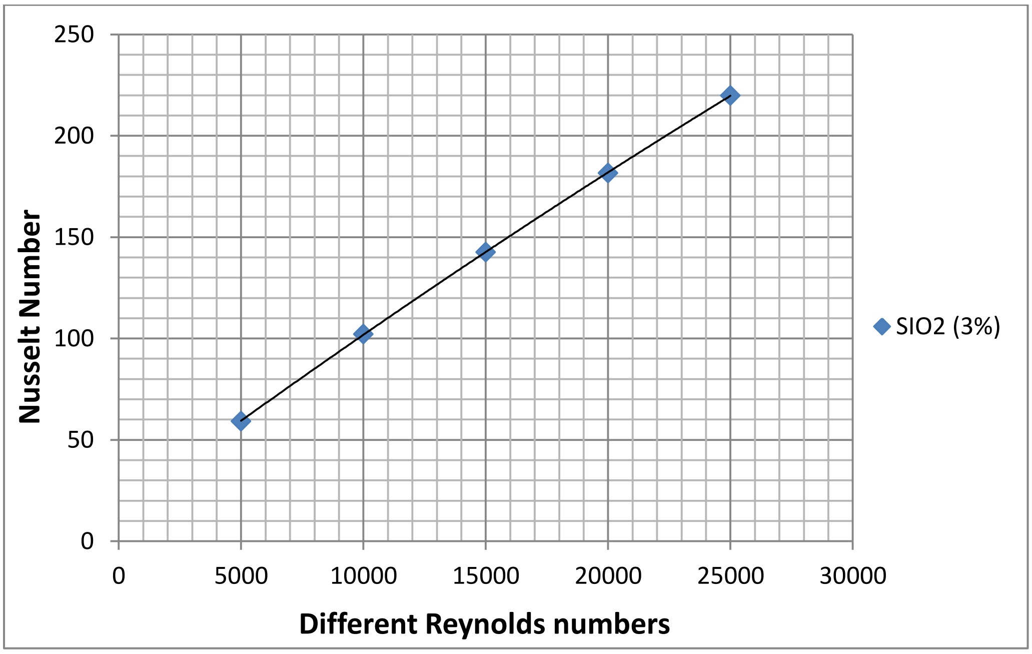

Figure 4-10 *Nusselt number with different Reynolds numbers*

Figure 4-11 display effect of increasing Reynolds number on heat transfer coefficient of nano fluid. The result is for SiO_2 with 3% volume concentration in range of 5000 -25000 Reynolds number. Results show with increasing Reynolds number the heat transfer coefficient increase respectively. The highest heat transfer coefficient is for 25000 Reynolds number.

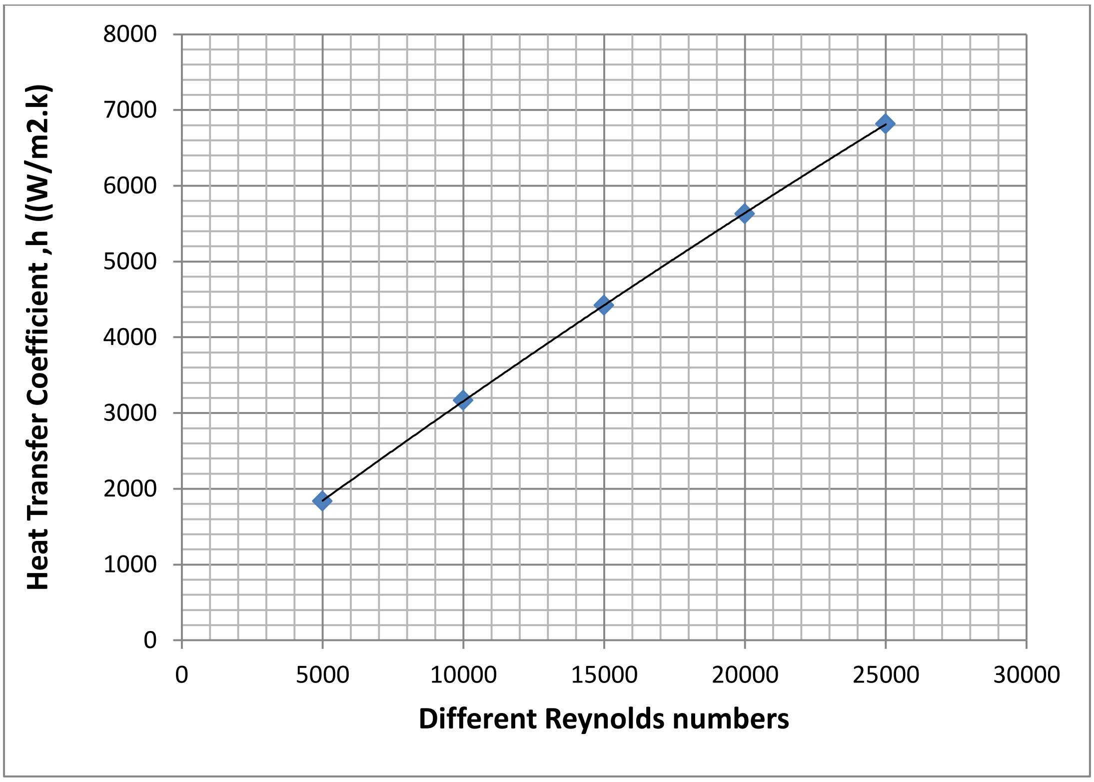

Figure 4-11 : Heat transfer coefficient with different Reynolds numbers

4.5 Summary

This chapter has presented the code validation results, the effect of different type of nanoparticles, different Reynolds numbers and different nanoparticle concentrations. The results can be fully utilized in many engineering applications in order to obtain an optimum and maximum heat transfer enhancement .The conclusions from this project and recommendation for future work are presented in the last chapter.

Chapter 5

5 CONCLUSION

5.1 Introduction

Numerical simulation of turbulent forced convection heat transfer in a rectangular heated pipe was carried out. The emphasis was given on the heat transfer enhancement resulting from various parameters which include different type of nanofluids (Al_2O_3 ,CuO,SiO_2 and ZnO) , volume fraction of nanoparticles in the range of $1\% < \varphi < 5\%$ and the Reynolds number in the range of $5000 < Re < 25000$. The governing equations were solved utilizing finite volume method with certain assumptions and appropriate boundary conditions to provide a clear understanding of the modeling aims and conditions for the present study. CFD software (ANSYS – FLUENT) has been employed in this study to simulate the current results.

5.2 Conclusions

The main purpose of this project is to investigate the thermal characteristics of turbulent nanofluid flowing in rectangular channel at a constant temperature. A number of conclusions were drawn from the current work as follows:

1. By changing the types of nanoparticles , (Al_2O_3 ,CuO,SiO2 and ZnO) , the result shows that SiO_2 generates the highest Nusselt number followed by Al_2O_3 , ZnO and CuO respectively while pure water provides the lowest Nusselt number.

2. The Nusselt number increases with the increase of the volume fraction of nanoparticles.

3. The Nusselt number increases gradually when the Reynolds number increased.

Finally, the study shows that the use of nanofluids is very effective in the enhancement of heat transfer rate compared to conventional heat transfer fluids. The investigations of this work will definitely solve many heat transfer related problems in the near future and very likely to be applied in the numerous practical energy transports process devices.

5.3 Recommendations for Future work

Based on this project, the following recommendations could be made for future work:

1. An experimental set up could be constructed to validate the numerical results of this study.
2. The heat transfer performance of other channels of different shape could be evaluated.
3. The study of the same geometry with different boundary conditions can be performed.
4. The study of effect of rib- grove could be considered.
5. The study of the nanofluids in the present geometry could be further extended with the use of other nanoparticles (such as Ag ,Au, SiN and SiC)

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