EVALUATION OF HEAT TRANSFER PERFORMANCE OF HELICAL COIL HEAT EXCHANGER

by

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16892

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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
Mechanical Engineering Programme
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(MECHANICAL)

Approved by,

____

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May 2015

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

MOHD FATHUR RAHMAN BIN MOHD NADZIM

ABSTRACT

Helical tubes have been widely used in the industry due to their compactness in structure, high heat transfer rate and ease of manufacture. They are commonly found in industry as heat exchangers or chemical reactors. Aside from the industrial application, fluid flow phenomena in coiled duct have also attracted much attention from engineering researchers. So, basically the aim of this project is to investigate the heat transfer performance of various configurations of helical coil heat exchanger such as circular concentric and eccentric tube-in-tube and with different cross sections geometry which are square concentric and eccentric. Moreover, parametric study will also be conducted for several parameters like co-flow in tube, different fluid inlet velocity in inner tubes and annulus region then different in cross-sections geometry for helical heat exchangers by using numerical investigation of well-known ANSYS Fluent CFD software. By commanding boundary conditions and choosing of an applicable mesh, the obtained results were compared and validated with existing experimental results in open literature. The result designates that augments of heat transfer by increasing the inner Dean number, annulus flow rates and concentricity of tube-in-tubes helical heat exchanger. However, the square cross section also can be a part of parameter that going to affect in increasing heat transfer.

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In the name of Allah, The Most Merciful and Most Compassionate

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CHAPTER 1: INTRODUCTION

1.1 Background

Nowadays, heat exchanger has become a major equipment that needed to ease our daily life. For instance heat exchanger application in our surrounding that commonly being used are air conditioners, refrigerators and cooling system in car engine. Basically the mechanism in heat exchanger is a system designed to transfer heat between two fluids. The function is to control the temperature of the fluids whether in high or low temperature. A heat exchanger could remove or add thermal energy to a system where the process require a certain temperature to work properly without mixing the fluid together. For instance, if we want to add thermal energy to the liquid such as gases, we cannot simply heated like the solid where it cannot be mix together but for gases it will be difficult because the best way to heat-up is by using heat exchanger. However, there are many design of heat exchanger such as shell-and-tube, open-flow, plate contact and double pipe heat exchanger. In this project, double pipe heat exchanger was the main focusing type that going to be compare with helical coil heat exchanger.

Helical coil has been known as one types of heat exchanger. Therefore, helical coil has been known to offer high heat transfer when compared to straight tube heat exchanger (Fernández-Seara et al., 2013), (Kurnia et al., 2012) & (Prabhanjan et al., 2002). As such, helical coil heat exchangers are widely used in industry such as steam generators, refrigerators, nuclear reactors, chemical plants and domestic hot water systems as heat exchanger or chemical reactors. It is due to their compactness in structure, ease of manufacture, ease of maintenance and improved thermal efficiency. The inner and outer tubes was transfer heat by convection process, but for conduction process it is through the tube wall and fouling resistances where there are on the inner and outer tubes that will affects the performance of heat transfer.

Aside from the industrial application, transport phenomena in coiled duct have also attracted much intention from engineering researchers. Therefore for this evaluation of heat transfer performance of helical coil heat exchanger project, the purpose is to investigate the heat transfer performance of various configuration of helical coil heat exchanger which are tube-in-tube and side-by-side. Moreover, the parametric study will also be conducted for various flow such as co-flow and counter flow, fluid inlet velocity and cross-section geometry. This study will be using numerical method to evaluate the heat transfer rates for non-circular tubes and eccentricity of helical coil.

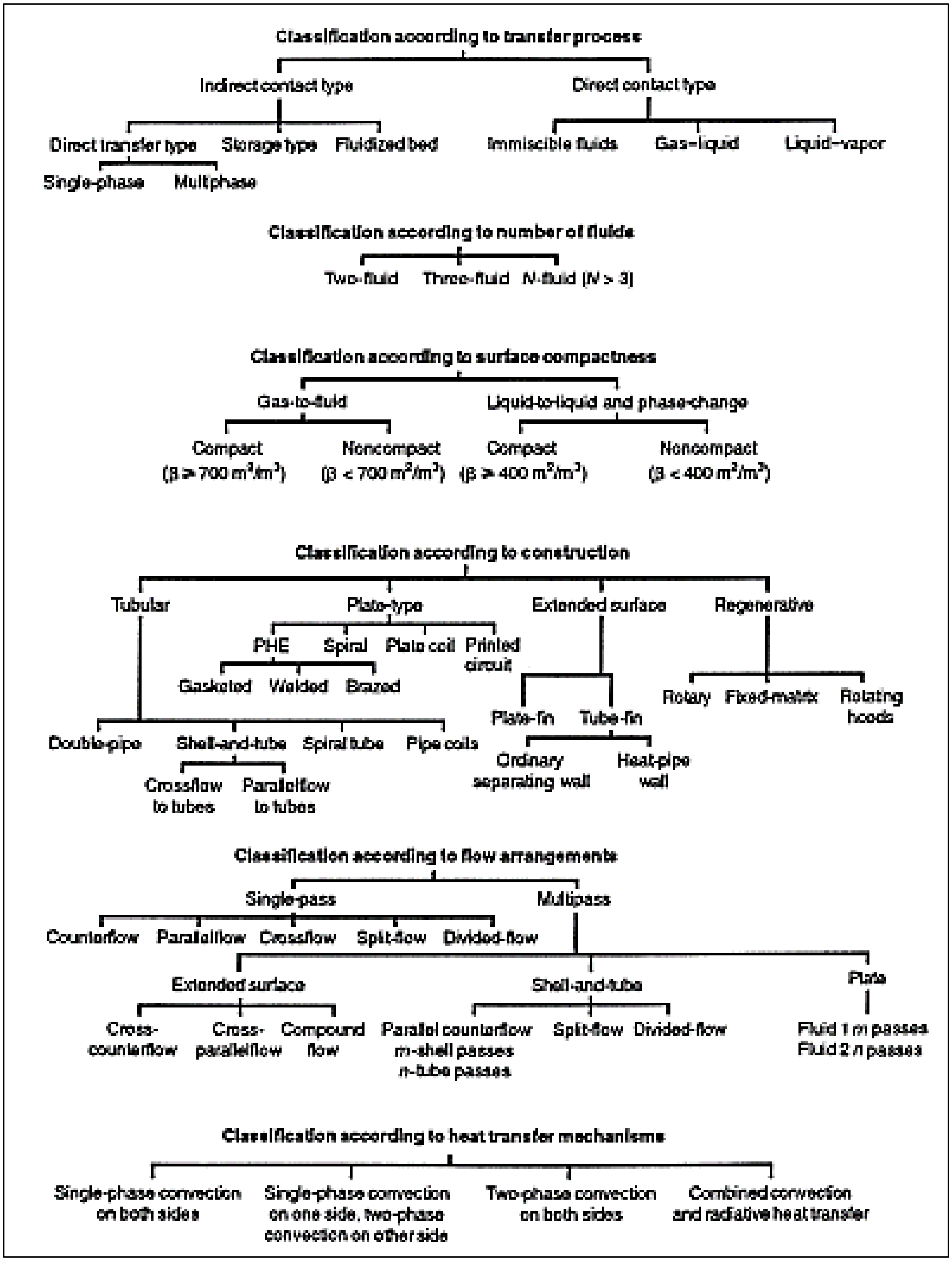

Figure 1.0: Classification of heat exchanger (Shah, 1981)

1.2 Problem Statement

Increasing heat transfer performance has been a major focus of engineering. Due to large application in industries make the people think about an improvement to the heat exchanger itself which the main function is to transfer heat whether from high to low temperature or vice versa. However, other than focusing about the heat transfers rate, they also trying to compare the heat exchanger of straight tube with the heat exchanger of helical coil due to the compactness of geometries and heat transfer rates. Many research have been done to prove that they can make the same efficiency of heat transfers with smaller geometry of heat exchanger. One of the result found is by using helical coil heat exchanger.

Therefore, helical coil design offers higher heat transfer rate due to the presence of secondary flow induced by coil curvature. However, the previous studies focus on circular cross-section and concentricity for inner tube. Only limited study dealt with non-circular cross-section and its eccentricity of helical tube heat exchanger. Hence this study conducted to evaluate the heat transfer performance in helical heat exchanger.

1.3 Objectives

There are the objectives of project entitle of evaluation of heat transfer performance of helical coil heat exchanger which focusing in different cross-section of tube.

- To investigate flow characteristics and heat transfers phenomena in helical coil heat exchanger
- To identify and evaluate several parameters affecting the performance of a helical coil heat exchanger

1.4 Scope of Study

The scope of study of this project is to evaluate the heat transfer performance of helical coil heat exchanger which more focusing on the type of double pipe heat exchanger and also includes:

I. The considered working fluid in this study is water.
II. Only one turn of helical heat exchanger is studied in this study.
III. The cross-section area is fixed with different cross-section geometry.
IV. This study dealt with laminar flow in helical tube.
V. Water properties in only function of temperature.

CHAPTER 2: LITERATURE REVIEW

2.1 Heat Transfer

Heat transfer involved in two flowing fluids but it has been separated with the solid wall. From the inside of tube that contain hot fluid, the heat will transfer to the wall by convection. While through the wall it is by conduction and again from the wall to the cold fluid is by convection. That the basic working principle of heat transfers inside the tube-in-tube heat exchanger.

The investigated hydrodynamic and heat transfer characteristic of tube-in-tube helical heat exchanger was done (Kumar et al., 2006). They are doing experimental of counter flow heat exchanger and evaluated the heat transfer coefficients. For the Nusselt number and friction factor coefficient was found and validated with the numerical value from computer fluid dynamic (CFD) package (FLUENT). The result shown that the overall heat transfer coefficient increase with increasing the inner coil tube Dean Number for constant flow rate in annulus region.

Other than that, it has been noted by Naphon and Wongwise et al. (2005), which study of the heat characteristics of a compact spiral coil heat exchanger under wet-surface condition and they had done the numerical and experimental studies. It is to find out the heat transfer rate and predict the performance of the spiral coil heat exchangers. The analysis is using cooling and dehumidifying conditions. They found the mass flow rate and temperature of air values at the inlet and outlet. The increase in mass flow rate of water give the outlet temperature of air and water decrease. Hence with the increase of mass flow rate of air and water gives the enthalpy and humidity effectiveness decrease.

In addition, the study of difference sizes of the inner tubes of double pipe helical heat exchanger that affect the heat transfer characteristics (Rennie, 2006). The result showed an increasing the overall heat transfer coefficient as increasing the inner dean number which considering an annulus region have stronger influence in term of thermal resistance. Other than that, by increasing the inner tube size result in lower thermal resistance in annulus region because thermal resistance in inner tube remained fairly constant.

However, by referring to Jianqin and Shoapeng, (2000), they numerically studies about the different eccentricity ratio of TTHC heat exchanger and performed with helium in the inner tube and annulus. The increasing value of eccentricity, the inner Nusselt number will decrease while the annulus Nusselt number increase rapidly. Eccentricity is good for heat transfer in annulus region of tube-in-tube helically coiled (TTHC) heat exchanger, especially in large Reynolds number region.

In other work, Naphon (2007) conducted a study on a complex heat exchanger involving two helical coil banks with fins attached to the coils, inside of a sectioned shell. The heat exchanger was operated in the counter flow configuration. Hot water and cold water were used for the coil and shell sides, respectively. Though no results were presented on inner or outer heat transfer coefficients, results were given relating the heat exchanger effectiveness versus shell and coil flowrates. For low hot water mass flowrates, the heat exchanger effectiveness was seen to increase with increasing coil hot water inlet temperature. At higher hot water flowrates, the effectiveness converges onto a single value, regardless of hot water inlet temperature. The highest effectiveness is seen with the largest shell side flowrate and the lowest coil side flowrate, and the lowest effectiveness is seen when the inverse situation occurs.

2.2 Helical Coil

Many studies have been done to replace the straight inner tubes with helical coil heat exchanger due to the compactness of geometry with same heat transfer amount. This is cause by larger heat transfer surface area and small in a geometry of tubes but increase the pressure drop across the heat exchanger. Due to the geometrical configuration of helical coils with more complex flow pattern also report an additional centrifugal force on the inner coil flow and increasing the pressure drop on the shell side. Dean number is often used which represent the ratio of the viscous force acting on a fluid flowing in a curve pipe to the centrifugal force. The Dean number will never be larger than the Reynolds number for same flow (Gaskil, 2011). The effects of the centrifugal force dominate flow as a Dean number approaches the Reynolds number and it effect on the heat transfer.

According to Kurnia et al. (2014), compared between in-plane spirals, conical spiral coils and helical coil, the best heat transfer rate is helical coil and then followed by others. The study have been done for heat transfer performance of power law fluids in coiled square with these three type of coil. For pressure drop which become the important factor that affect heat transfer and the comparison between these three types give the helical coil to the highest pressure drop.

Seban and McLaughlin (1963), they have studied heat transfer through helical coil by using two different curvature diameter ratios which 0.0588 and 0.0096. The curvature diameter ratio is defined as the ratio of the inner diameter of the pipe, *d*, to the curvature diameter of helix, *D*. The flow is in range of 12< Re<65,000 was varied from laminar to turbulent. The local heat transfer coefficients for laminar flow were found to be consistently larger on the outer half (peripherally) than on the inner half. For all cases, a larger heat transfer coefficient was seen relative to a straight tube.

In the other hand, Jamshidi et.al. (2013) study the heat transfer rates in shell and helical tube heat exchanger and finding the optimum design parameters in coiled tube heat exchanger. The result given is by decreasing the coil pitch, in the inner tube the Nusselt number will increase but will decrease in shell side. However, by increasing the coil diameter, for inner tube will increase the Nusselt number and overall heat transfer coefficient but in shell side the Nusselt number will decrease.

2.3 Flow Characteristics

In heat exchanger, the flow inside was the main character that going to effect the performance of heat exchanger. It was supported by Rennie (2004), the location of the maximum axial velocity is recorded toward the outer wall of a curved tube. Therefore, increasing the resistance to flow in curved tube compared to straight tube will related to curvature ratio. Even though it have slight curvature actually it tend to change the critical velocity that is common indicator of transition from laminar to turbulent flow. However, by using ink injections into water flowing through coiled tubes, the secondary flow of pattern was observed. Whenever a fluid flows in a curved pipe or channel, the secondary flow will appears.

Therefore, the study of heat transfer coefficient and parameters profile with numerical and experimental analysis was done (Kumar et al., 2006). The results have been compared between experimental, present numerical and also literature data. In numerical analysis, it shows that the velocity fields for inner and annulus tubes. With different degree of length from the starting tubes (30, 60, 90, 180, 270, and 360) degree, it stated that by increasing the axial distance, the secondary flow is strong for both regions. It also shows the temperature field for both regions. As the higher axial distance, the temperature distribution almost ceases to change which after 180 degree. In experimental, in the annulus side it flow with cold water and in the inner tube it flow with hot water. The result shows heat transfer coefficient increase with increase the flow rate inside.

Hence, the numerical simulation were carried out for a square duct Reynolds number of 1000 and a fixed fluid rate to examine seven different non-circular duct geometries such as straight duct, in-plane spiral ducts with various cross-section geometries including square, 2x1 and 4x1 rectangular, trapezoid, triangular and half circular (Kurnia et al., 2012). From earlier investigations shown that it will lead to significant radial pressure gradient in the flow core region with the presence of centrifugal force due to curvature. The axial velocity and the centrifugal force will approach zero in the proximity of curved duct inner and outer walls. Hence, secondary flow will occur to balance the momentum transport. So, the result from studies shown with higher velocity in generated in the outer wall of spiral ducts for the secondary flow.

Moreover, the study on the flow and heat transfer characteristics in spiral-coil tube heat exchanger was done, (Paisarn, 2005). He did mathematical and experimental study on horizontal spiral-coil tube to predict the flow characteristics whether the heat transfer rate or heat transfer coefficient had affected by the centrifugal force. However, the pressure drop and also the Nusselt number from the spiral tube are increases higher than the straight tube due to centrifugal force.

Other than that, numerical study of tube-in-tube helically coiled heat exchanger by considering compressed air in the inner tube at various operating pressure and cooling water in the outer tube was done, (Kumar et al., 2008). Heat conduction within the tube walls was considers but if compare to the literature they don’t consider it. So, the result showed overall heat transfer coefficient increase with increase the flow rate in the inner tubes when constant in outer and vice versa. Increase in operating pressure in the inner tube also increase the overall heat transfer coefficient.

CHAPTER 3: METHODOLOGY

The methodology for this project was done by using numerical method of study. Computer fluid dynamics (CFD) package was used (ANSYS FLUENT) to control volume finite difference method (CVFDM) and evaluate certain parameters such as pressure, flow rates and temperature due to cross section in helical coil double pipe heat exchanger. Then, validation purposes from the result was identified whether its correlate with experimental literature reviews.

The heat exchanger were numerically modeled consist of two spiral ducts, consist of one inside the other (tube-in-tube) and side-by-side (eccentric). The dimension of the coiled ducts were taken as same as in the literature reviews (Rennie et al., 2005). The hot fluid flows in the inner duct tube, while the cold fluid flows in the annulus region whether in co-flows or counter flows.

Then we set the condition according to literature data with fixed the flow rates in annulus region where in the inner region we run with different flow rates. However, for side-by-side type the flows is depend on the fixed area of calculated region base on the validation data's. We also can evaluate and compare the differences between various cross section inside the square ducts.

3.1 CFD modeling

This project will considered circular eccentric and square cross-section for the coil geometry. The geometries for the helical coil was created in AutoCAD 12 which to get the basic geometries for helical shape before export to pre-processor GAMBIT for meshing, labeling boundary condition and determining the computational domain. It will export as stereo lithography (.sat) files.

In GAMBIT software, a relative mesh is generated. This mesh contains mixed elements (Hexahedral and Hexahedral/wedge) having several types of faces (Map, Submap & Cooper) at the boundaries. Precaution is taken to use structured hexahedral cells as much as probable. It is meant to reduce numerical distribution as much as possible by structuring the mesh in a well mode, mostly near the wall region. Then, a fine mesh is generated. But there are constraint in denser meshed volume regarding this project because ANSYS FLUENT software of students version have limitation in meshed volume which is only up to 512 000. The figure 3.1 shows the CFD modelling that have been developed in Gambit software.

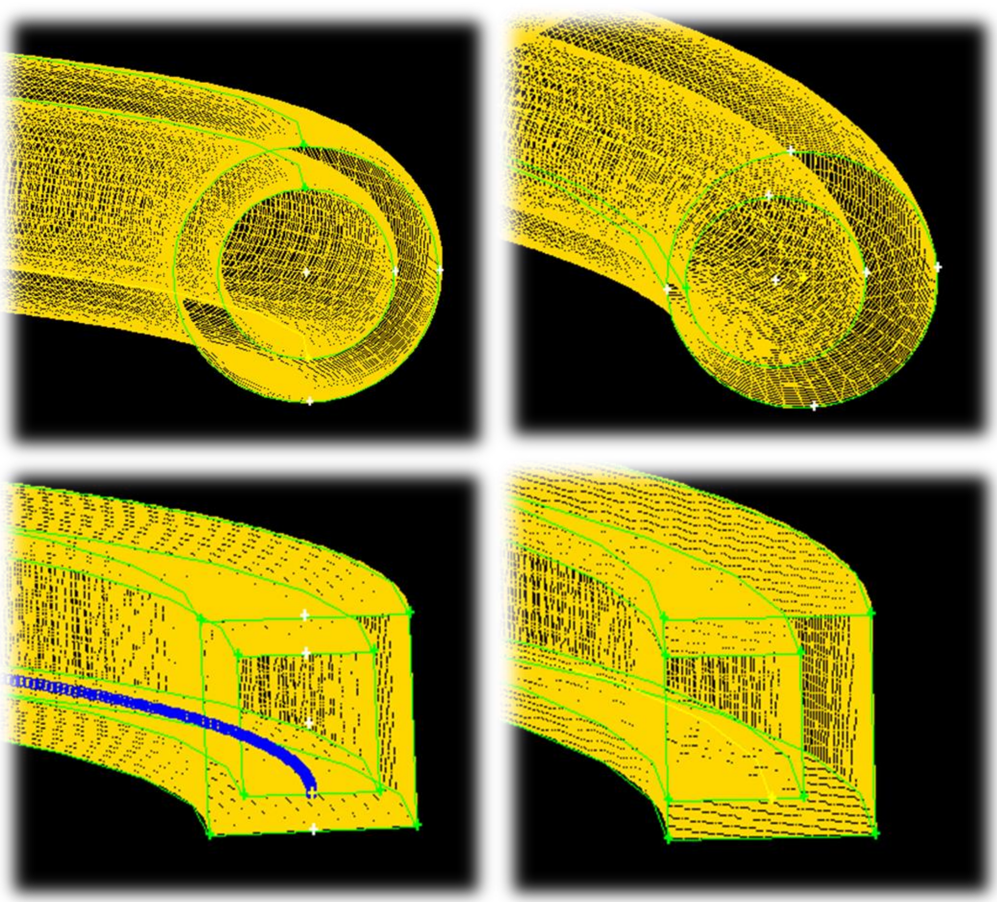

Figure 3.0: The model of helical heat exchanger

In addition, for the numerical method simulations, the step is shown in figures 3.0 & 3.1 below:

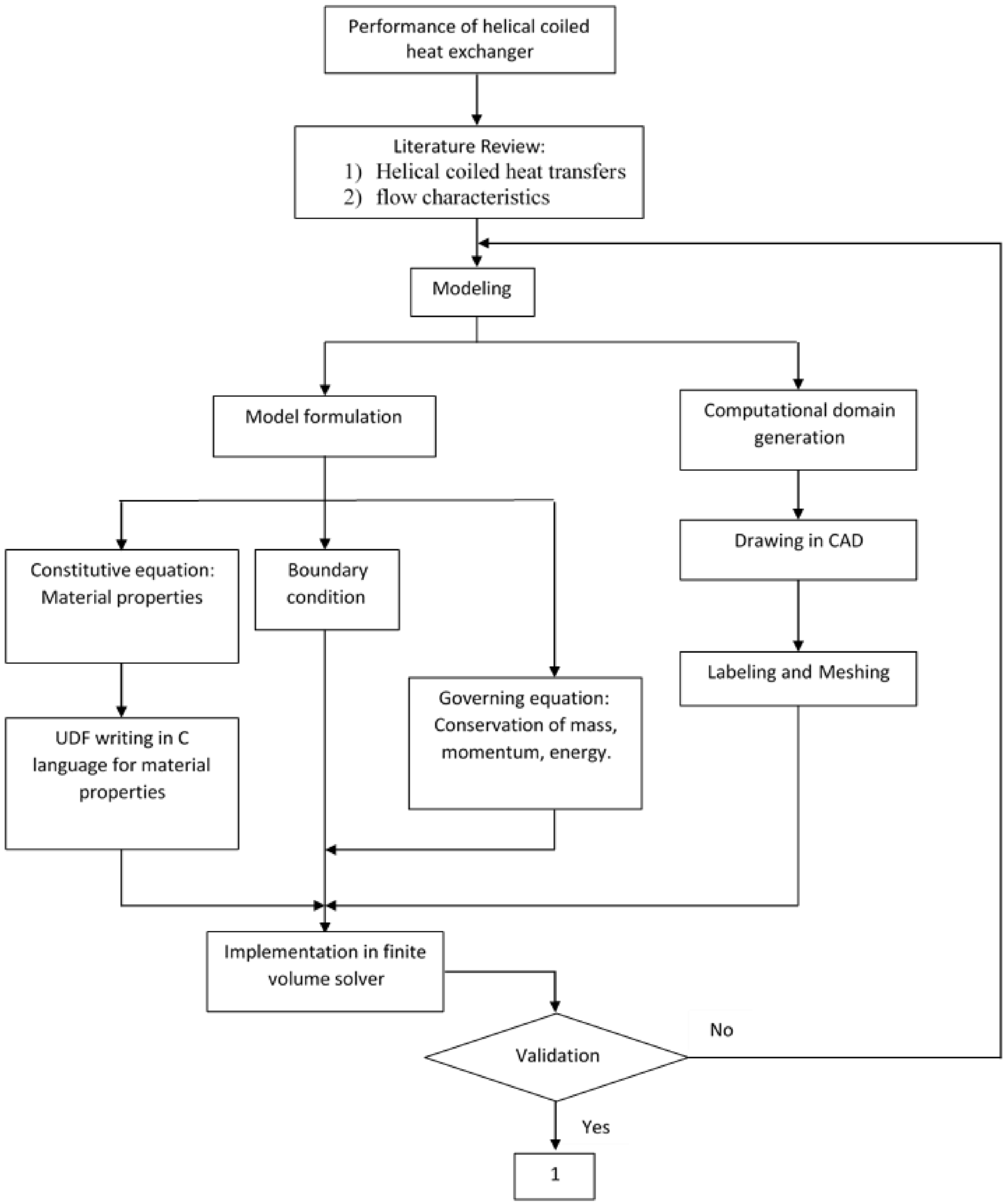

Figure 3.1: Process flow chart

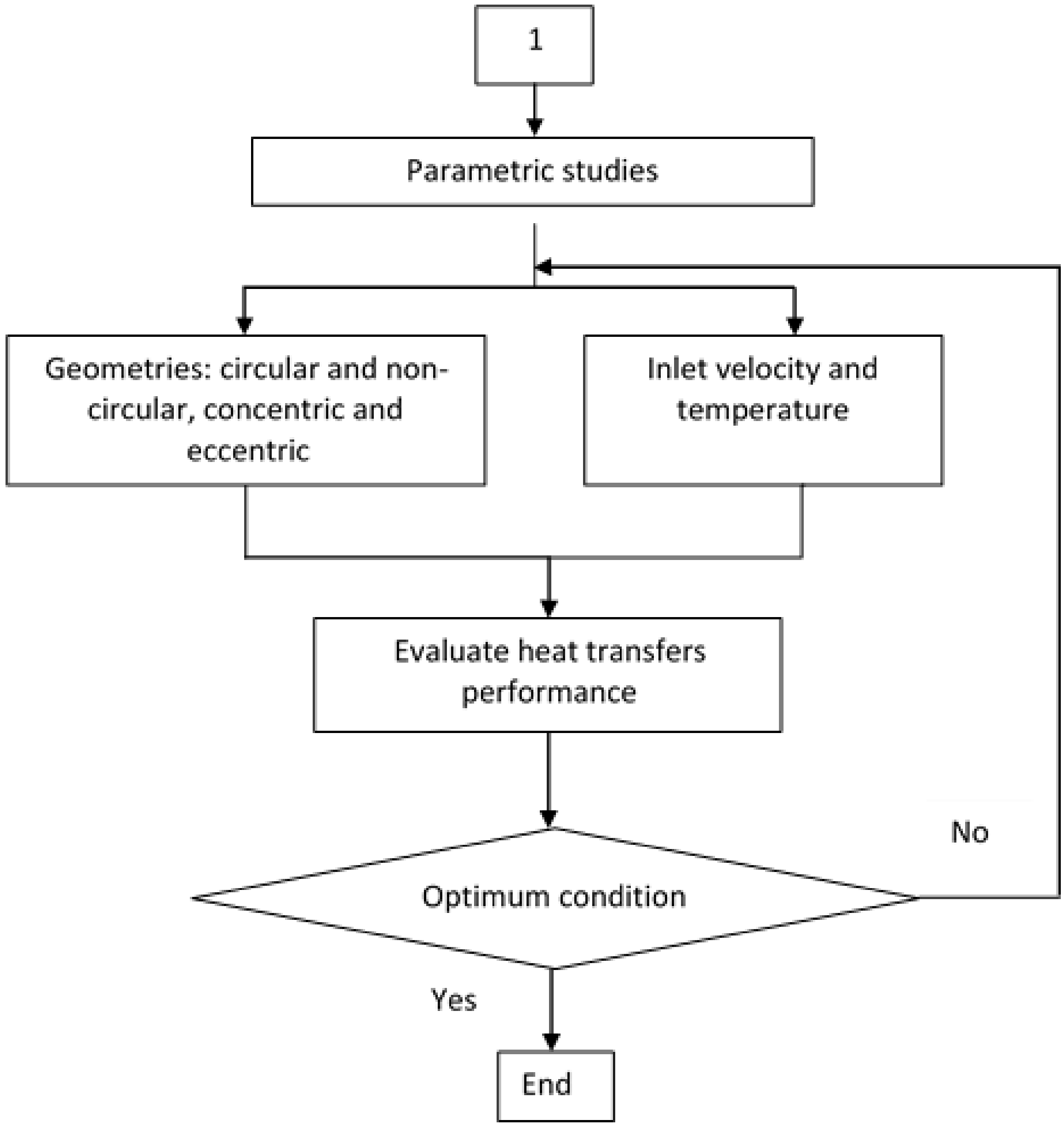

Figure 3.2: Continue process flow chart

The circular heat exchanger was constructed with the outer diameter of outer tube Do, inner diameter of inner tube Di, and is coiled with curvature radius Rc, while the distance between two turns (the pitch) is reported by H. Then, with outer diameter of inner tube do, and inner diameter of inner tube di. In this study, the Certesian coordinate system (x, y, z) is used to present flow in numerical simulation.

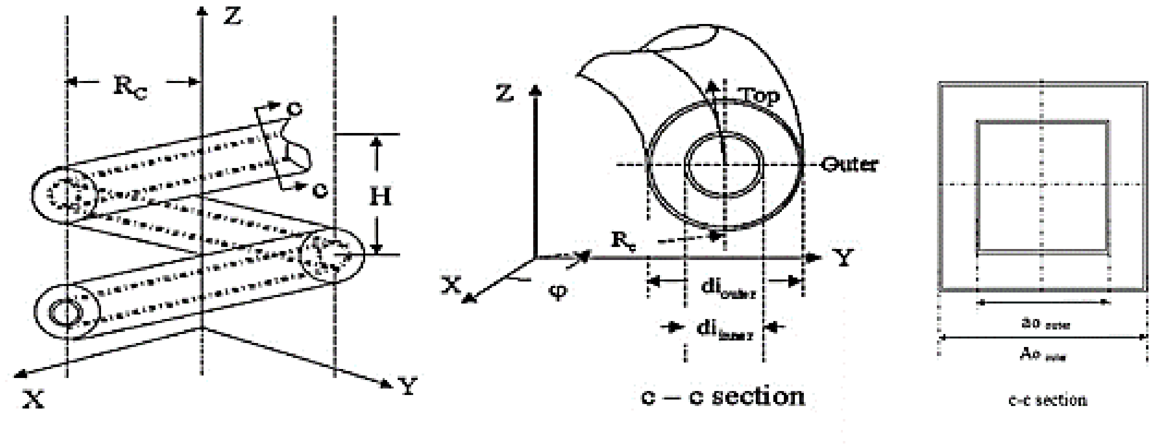

Figure 3.3: Schematic diagram and coordinate system of the circular and square tube-in-tube heat exchanger (Kumar et al., 2006)

Table 3.0: characteristics dimensions for ***circular*** *tube-in-tube heat exchanger*

Dimensional parameters	**Heat Exchanger**
Inner diameter of outer tube, Di (mm)	14.3
Outer diameter of outer tube, Do (mm)	15.9
Inner diameter of inner tube, di (mm)	7.9
Outer diameter of inner tube, do (mm)	9.5
Curvature Radius, Rc (mm)	235.9
The pitch, H (mm)	31.8

Table 3.1: Characteristics dimensions for ***square*** *tube-in-tube heat exchanger*

Dimensional parameters	**Heat Exchanger**
Inner length of outer tube, Ai (mm)	12.8
Outer length of outer tube, Ao (mm)	14.4
Inner length of inner tube, ai (mm)	6.8
Outer length of inner tube, ao (mm)	8.4
Curvature Radius, Rc (mm)	235.9
The pitch, H (mm)	31.8

3.1.1 Governing Equation

Since this analysis looks into laminar flow, a precise numerical solution is sufficient to simulate real life application very closely. Conservation equations for mass, momentum and energy for the flow inside the ducts are given by

$$\nabla \cdot \rho \mathrm{u} = 0, \tag{1}$$

$$\nabla \cdot (\rho u \otimes u) = -\nabla P + \nabla \cdot [\mu (\nabla \mathrm{u} + (\nabla \mathrm{u})^T)], \tag{2}$$

$$\rho c_p u \cdot \nabla \mathrm{T} = \mathrm{k}\nabla^2 T, \tag{3}$$

where ρ is the fluid density, u is the fluid velocity, P is the pressure, μ is the dynamic viscosity of the fluid, c_p is the specific heat of the fluid, k is the thermal conductivity and T is the temperature.

3.1.2 Constitutive relations

In this project, water is the main working fluid used to investigate the performance of heat transfers. The thermo-physical properties of water are given as functions of temperature. The density, viscosity, thermal conductivity and specific heat of water are defined as (Kays et al., 2005)

$$\rho_w = -3.570 \times 10^{-3} T^2 + 1.88T + 753.2, \tag{4}$$

$$\mu_w = 2.591 \times 10^{-5} \times 10^{\frac{238.3}{T-143.2}}, \tag{5}$$

$$k_w = -8.354 \times 10^{-6} T^2 + 6.53 \times 10^{-3} T - 0.5981, \tag{6}$$

$$c_{p,water} = 4200. \tag{7}$$

The mixed mean temperature is given by (Kays et al., 2005)

$$T_{mean} = \frac{1}{VA_c} \int_{A_c} T u dA_c, \tag{8}$$

$$V = \frac{1}{A_c} \int_{A_c} u dA_c. \tag{9}$$

The total heat rate is define as

$$\dot{Q}_{total} = \dot{m} c_p \, (T_{mean,L} - T_{mean,0}) \tag{10}$$

3.1.3 Calculation of the heat transfer coefficient

The overall heat transfer coefficient is obtained using the inlet and outlet temperatures, and the question is:

$$U = \frac{\dot{Q}}{A_o\, \Delta T_{LMTD}}, \tag{11}$$

$$\Delta T_{LMTD} = \frac{\Delta T_2 - \Delta T_1}{ln(\Delta T_2/\Delta T_1)}, \tag{12}$$

Where we have $\Delta T_1$and ΔT_2:

$\Delta T_1 = T_{h,i} - T_{c,i}$

$\Delta T_2 = T_{h,o} - T_{c,o}$

Where $T_{h,i}$ and $T_{h,o}$ are the temperature of hot fluid at the inlet and outlet.

3.2 Numerical Simulation (ANSYS FLUENT)

3.2.1 Problem Setup

The mesh is checked and quality is obtained. The analysis type is changed to Pressure Based type. The velocity formulation is changed to absolute and time to steady state.

3.2.2 Models

Energy is set to ON position. Viscous model is selected as laminar.

3.2.3 Materials

Create or edit option is clicked to add water-liquid and copper to the list of fluid and solid respectively from the fluent database.

3.2.4 Cell zone conditions

The parts are assigned as water and copper as per fluid/solid parts.

3.2.5 Boundary Conditions

Boundary conditions are used according to the need of the model. The inlet and outlet conditions are defined as velocity inlet and pressure outlet. As this is a co-flow with two tubes so there are two inlets and two outlets. The walls are separately specified with respective boundary conditions. No slip condition is considered for each wall and tube walls each wall is set to zero heat flux condition. For inner tubes, the thickness have been fixed to 0.8 mm. The details about all boundary conditions for circular tubes and square ducts can be seen in the table 3.2 & 3.3 as given below.

Table 3.2: Boundary Condition for circular tubes

	Boundary Condition Type	**Temperature (K)**	**Velocity magnitude for different flow rates (m/s)**				
			Re=500	**Re=1000**	**Re=1500**	**Re=2000**	**Re=2300**
Inner inlet	Velocity inlet	333	0.025	0.05	0.075	0.10	0.115
Inner outlet	Pressure outlet	-	-	-	-	-	-
			300 cm^3/min		**500 cm^3/min**		**700 cm^3/min**
Outer inlet	Velocity inlet	295.1	0.056		0.093		0.130
Outer outlet	Pressure outlet	-	-		-		-

Table 3.3: Boundary condition for square ducts

	Boundary Condition Type	**Temperature (K)**	**Velocity magnitude for different flow rates (m/s)**				
			Re=500	**Re=1000**	**Re=1500**	**Re=2000**	**Re=2300**
Inner inlet	Velocity inlet	333	0.028	0.057	0.085	0.113	0.13
Inner outlet	Pressure outlet	-	-	-	-	-	-
			300 cm^3/min		**500 cm^3/min**		**700 cm^3/min**
Outer inlet	Velocity inlet	295.1	0.0536		0.0893		0.125
Outer outlet	Pressure outlet	-	-		-		-

3.2.6 Reference Values

The inner inlet is selected from the drop down list of "compute from". The values are:

- Area = 1 m^2
- Density = 1.225 kg/m^3
- Length = 1000

- Temperature = 288.16 K
- Velocity = 1 m/s
- Viscosity = 1.789 x10^{-5} kg/m-s
- Ratio of specific heats = 1.4

3.2.7 Solution Methods

The solution methods are specified as follows:

- Scheme = Simple
- Gradient = Green Gauss Cell Based
- Pressure = Second Order
- Momentum = Second Order Upwind
- Turbulent Kinetic Energy = Second Order Upwind
- Turbulent Dissipation Rate = Second Order Upwind

3.2.8 Solution Control and Initialization

Under relaxation factors the parameters are

- Pressure = 0.3 Pascal
- Density = 1 kg/m^3
- Body forces = 1 kg/m^2s^2
- Momentum = 0.7 kg-m/s
- Energy = 1 m^2/s^2

Then the solution initialization method is set to Standard Initialization whereas the reference frame is set to Relative cell zone. The inner inlet is selected from the compute from drop down list and the solution is initialized.

3.2.9 Measure of Convergence

It is tried to have a nice convergence throughout the simulation and hence criteria is made strict so as to get an accurate result. For this reason residuals are given as per the table 4 that follows.

Table 3.4: Residuals

Variable	**Residual**
x-velocity	10^{-6}
y-velocity	10^{-6}
z-velocity	10^{-6}
Continuity	10^{-6}

3.210 Run Calculation

Run simulation until convergence criteria is met. It roughly takes around 30 minutes for every single simulation to be done then, the result of surface integral and various contours plots are obtained.

3.2 Gantt Chart

NO	ACTIVITIES	FYP 1													
		1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	FYP Topic Selection														
2	Project Introduction														
3	Extended Proposal - Literature review														
4	Extended Proposal Submission														
5	Proposal Defense Preparation - Prepare for slide presentation														
7	Proposal Defense Evaluation														
8	Experimental early preparation - find information related														
9	Preparation to develop model by software - Learn to be familiar with software														
9	Submission of Interim Draft Report - preparation Interim Report														
10	Submission of Interim Report														

Figure 3.4: Gantt chart for semester 1 (FYP 1)

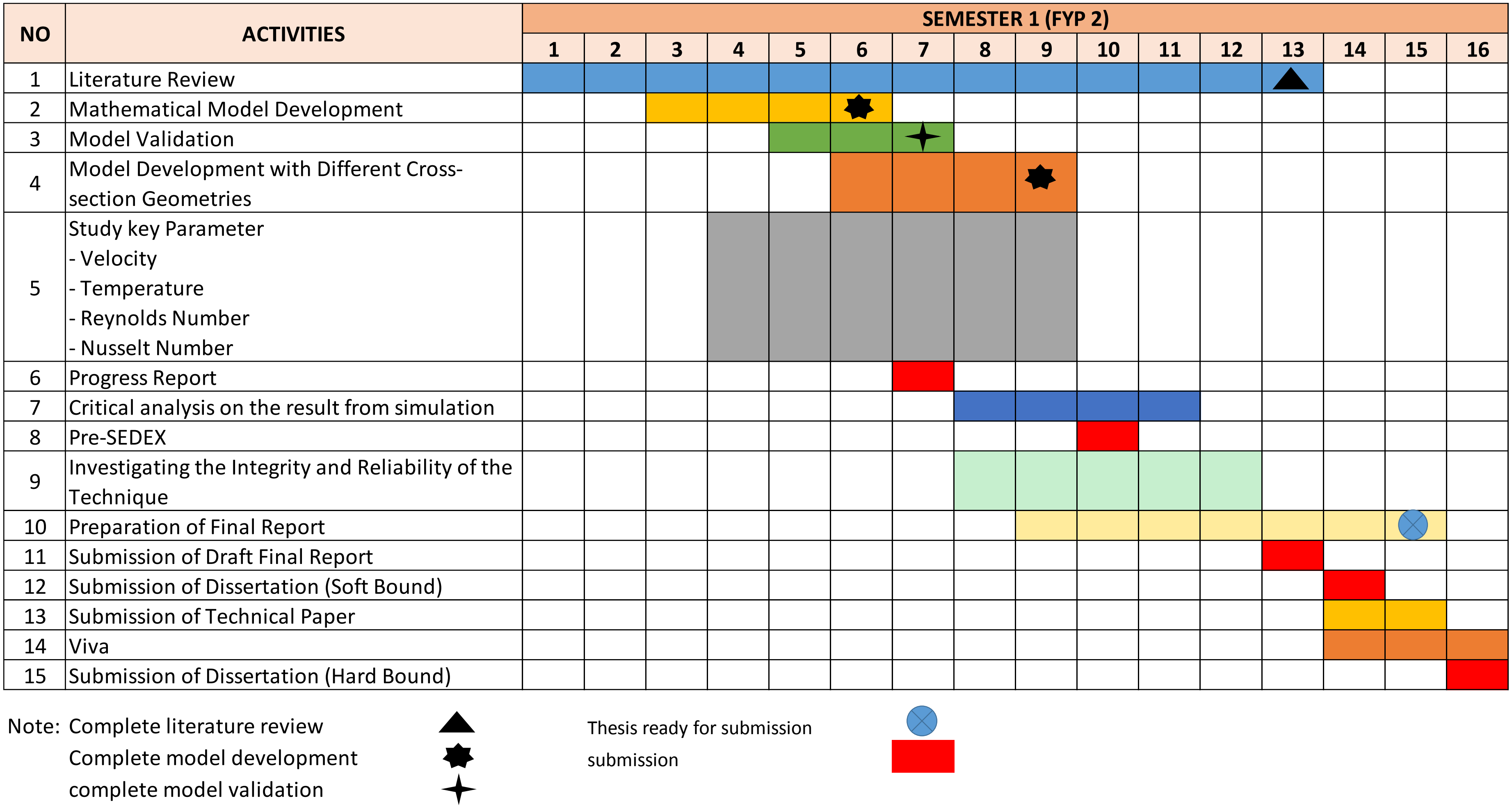

Figure 3.5: Gantt chart for semester 2 (FYP 2)

3.3 Key Milestone

There are the key milestone for this project which from Final Year Project (FYP) 1 and 2. It has been highlight the important date to be fulfill the required task in the FYP period. Below show the key milestone for this project.

Table 3.5: Key milestones

Key Milestone	**Complete**
Complete literature review	Week 14
Complete validation	Week 9
Complete geometry study	Week 10
Complete analysis on the result from simulation	Week 13
Complete final report	Week 15

CHAPTER 4: RESULTS AND DISCUSSION

In his chapter, it will be focusing on the results obtained from the simulation that have been running in ANSYS FLUENT. The validation has been done and discuss in this chapter.

4.1 Model Validation

The numerical method is used to investigate the model in this works as mentioned before, the numerical method is not suitable enough to get trusted results. Hence, before starting the model proposed with different cross-section geometries, the model should been developed in AutoCAD and Gambit according to the method and then continue running on ANSYS Fluent. It should be validated by some past work that already done.

For this purpose, the work from Ranjbar et al. (2014) and Rennie et al. (2005) act as references which experimental literature data that have been done as validation purposes. The comparison between the experimental works would be the best way to validate the results from the numerical method. The geometry is circular tube-in-tube helical coiled heat exchanger as figure 4.0 show the mesh plots. Unfortunately, the total volume of mesh is not dense enough for the mathematical model developed due to limitation numbers of meshed that allow in Fluent for student version. It is limited to 512000 total mesh volume.

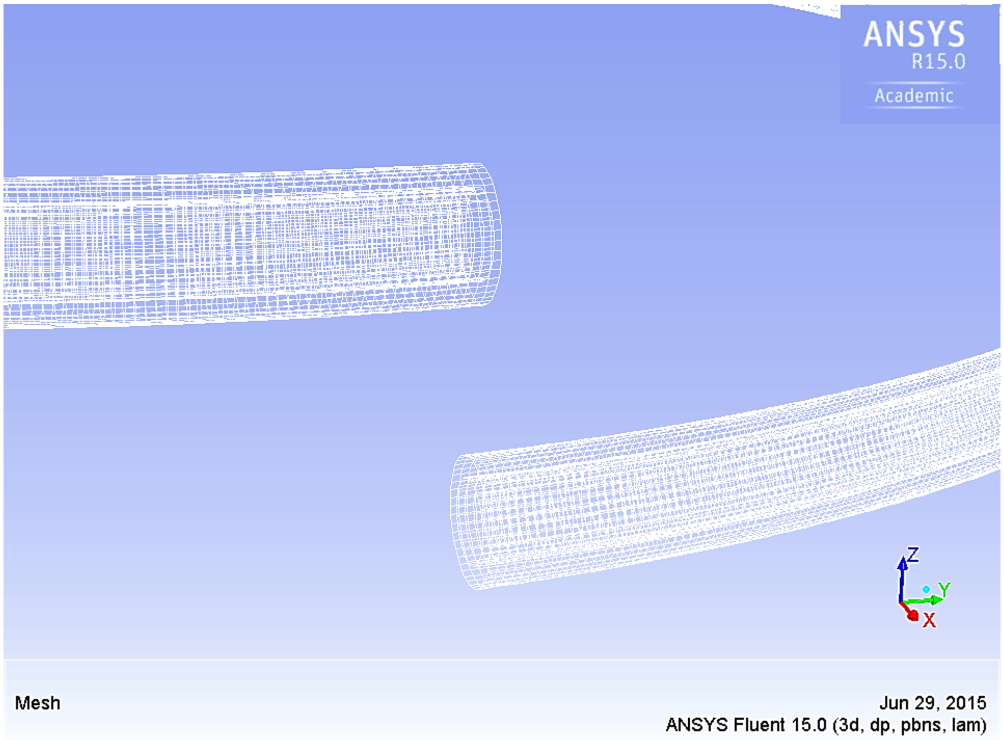

Figure 4.0: Mesh plot for circular tube-in-tube helical coiled

The overall heat transfer coefficient was calculated and compared with the overall heat transfer coefficient form the work done by Rennie et al. (2005). It has been compared with different flow in annulus region which are 300 cm^3/min, 500 cm^3/min and 700 cm^3/min while in the inner tube flow rates with different Reynolds number. The result shows that the overall heat transfer coefficient from calculated is slightly different from the present data. Overall heat transfer coefficient for parallel flow are presented in figure 4.1 which the overall heat transfer coefficient is plotted against the Inner Dean number.

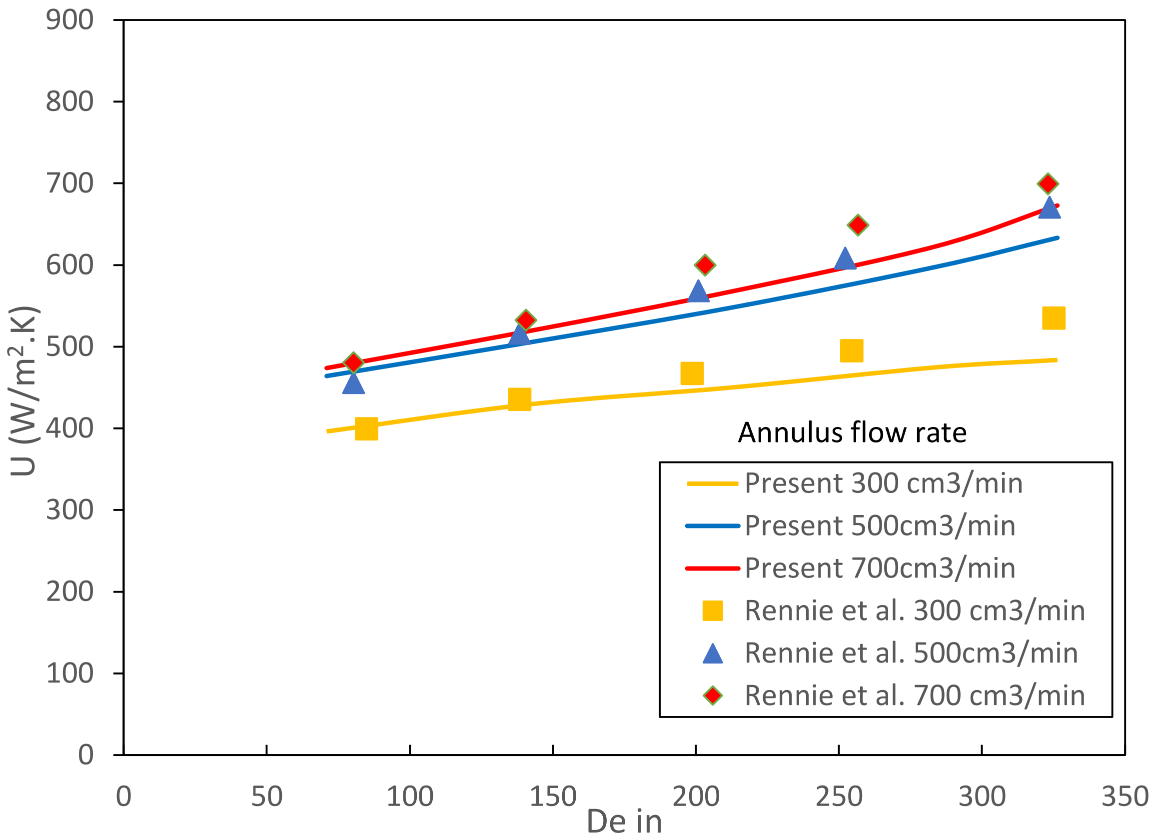

Figure 4.1: Overall heat transfer coefficient comparison in present predictions with the results of Rennie et al. (2005).

From the figure 4.1, its shows the present prediction is slightly higher compare to Rennie et al. (2005) at the lower inner Dean number but become lower varies with increasing inner Dean number. It happen may due to the error which has been set with different values. However, the overall results shows with higher Inner Dean Number will increase the overall heat transfer coefficient. In this case of the calculated results matches properly with the paper that has done and it’s fairly agrees with the predicted results.

4.1.1 Velocity Fields

From the execution of the model developed, the main factors that contributed the heat transfer rate is velocity. Based on the research before, the higher the velocity gives higher heat transfer rate.

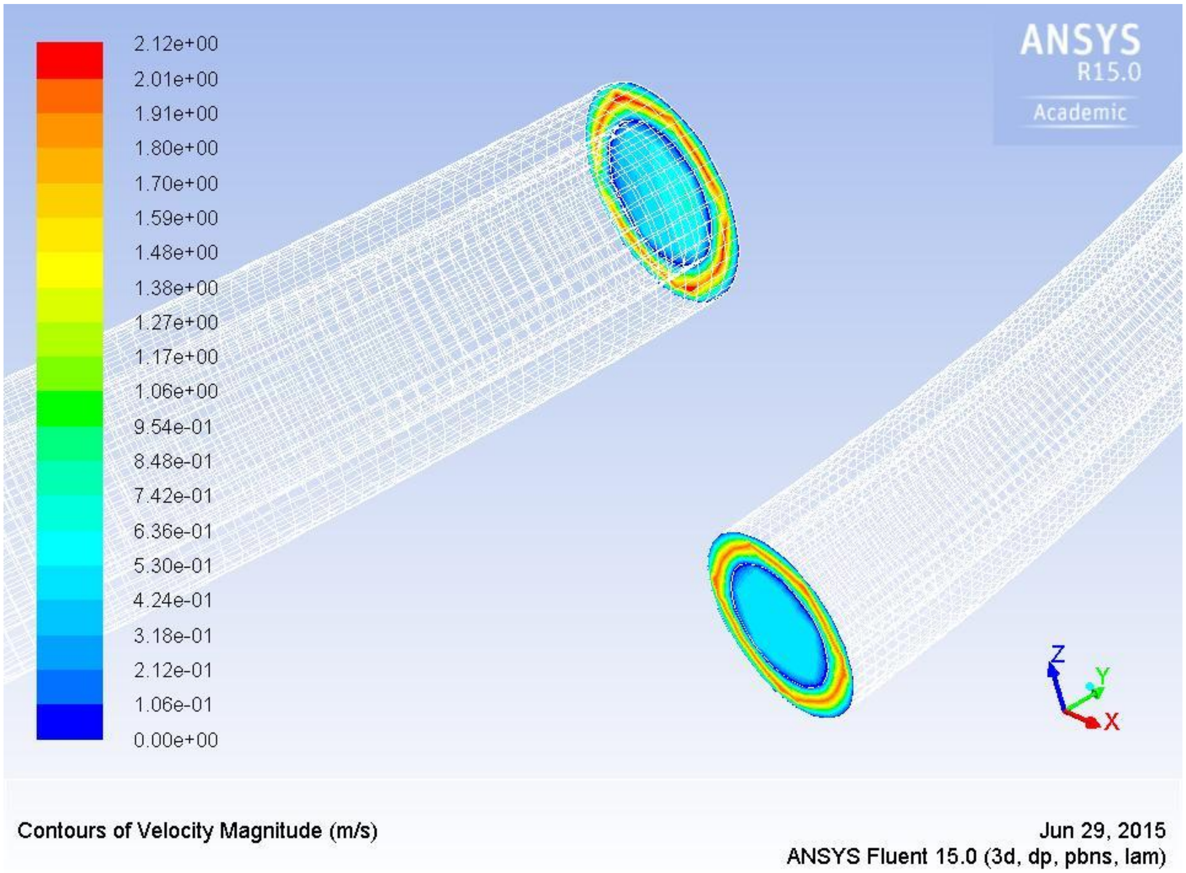

Figure 4.2: Contours of Velocity Magnitude at the Inlet and Outlet of Circular Concentric Helical Coiled

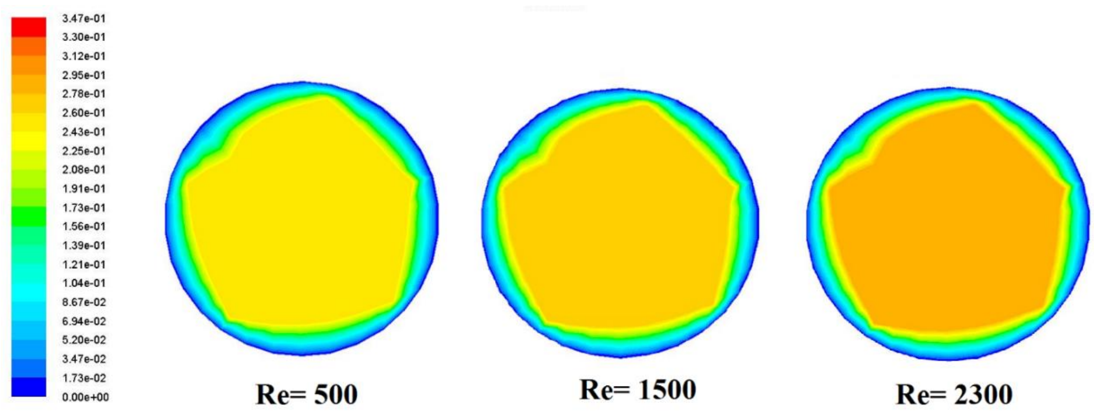

Figure 4.3: The velocity contours at the inlet of the inner tube with different Reynolds number

The figure 4.3 shows three different Reynolds number for the inner tube, as shows in figure above the velocity contours shape for inlet inner tube is look like the same even in different Reynolds number but the velocity sign shows with different velocity at inlet. The contours has same results because at the starting point the flow doesn't affect with the curvature tubes.

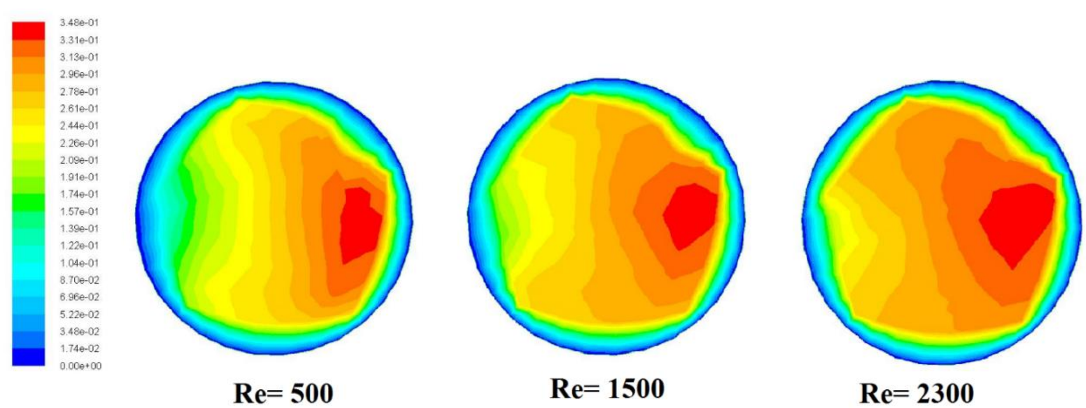

Figure 4.4: The velocity contours at the 180^o angle of length cross-section with the different Reynolds number

The higher Reynolds numbers at the 180^o angle of length cross-section shows the velocity at outer wall inner tube is higher compare to less Reynolds number. For inner side of inner tube, due to the centrifugal force of curvature tube make it has low velocity profile.

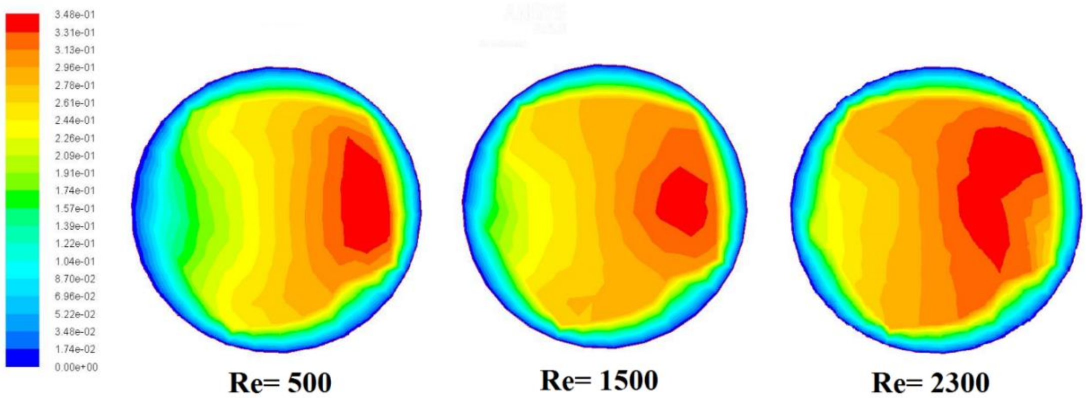

Figure 4.5: The velocity contours at the outlet of the tube with different Reynolds number

It is same with the previous result that shows the high Reynolds number give the velocity at outlet inner tube higher compare to others.

4.1.2 Temperature Fields

Temperature contours at 300 cm3/min annulus flow rates

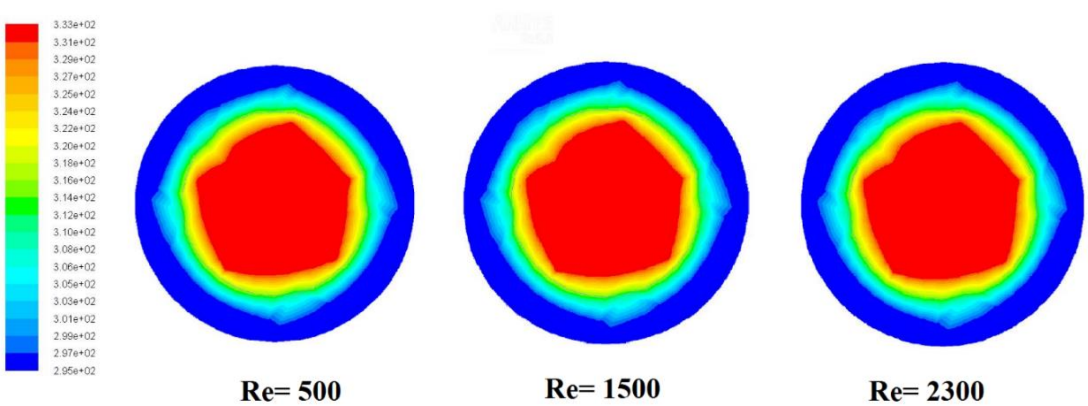

Figure 4.6: The temperature contours at the inlet

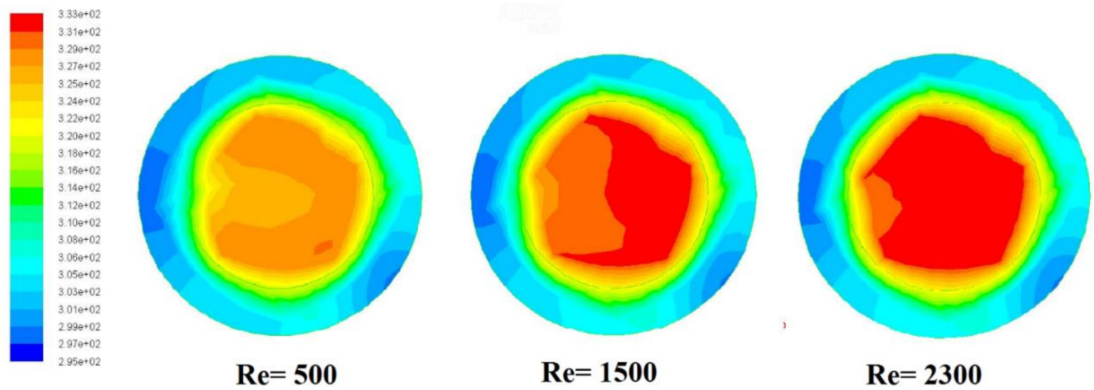

Figure 4.7: The temperature contours for tube at 180°angle of length cross-section with different Reynold number

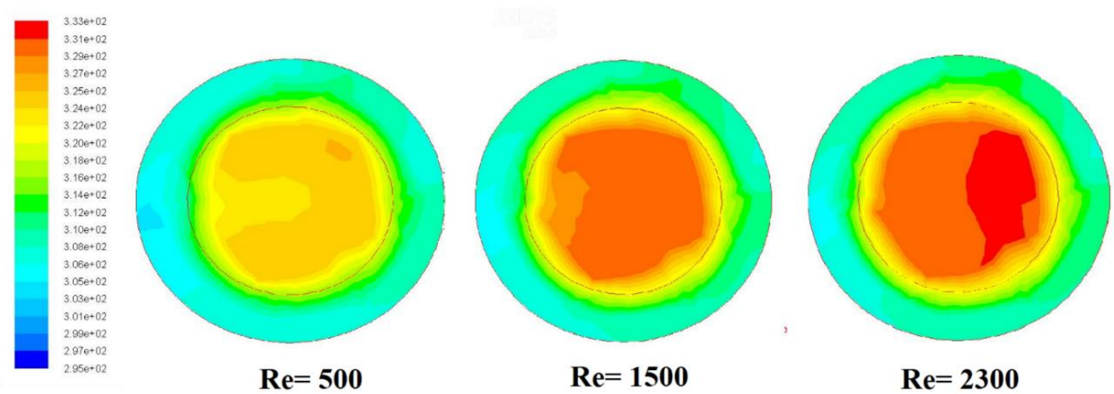

Figure 4.8: The temperature contours at outlet with different Reynolds numbers

As the figures above, its shows the temperature contours for different length of cross-section which at inlet, 180^0 length and outlet with different in Reynolds number. The patterns of the temperature shows in figure 4.7, the temperature zones at 180^0 length is shift to the outer side of tube because of the centrifugal force that occurs in helical coiled while in figure 4.8, its shows the temperature has more transfers in the annulus region with greater flow rates of fluid.

Temperature contours at 500 cm3/min annulus flow rates

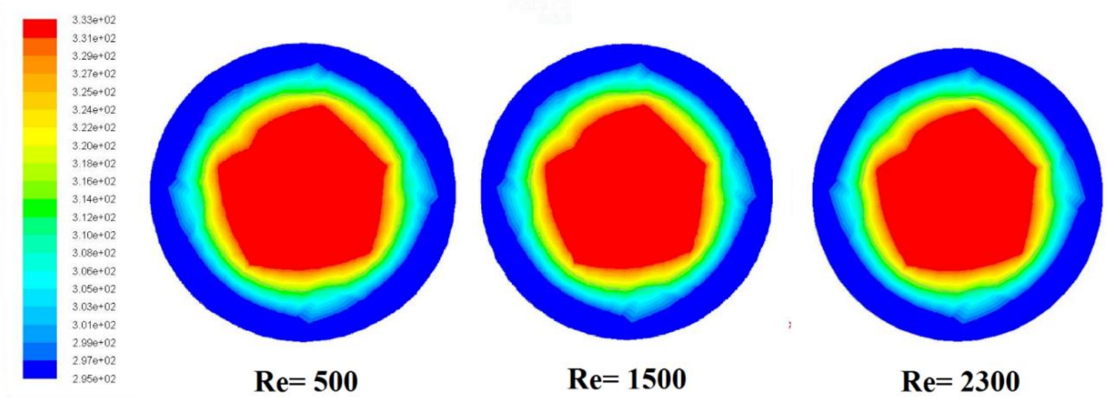

Figure 4.9: The temperature contours at the inlet

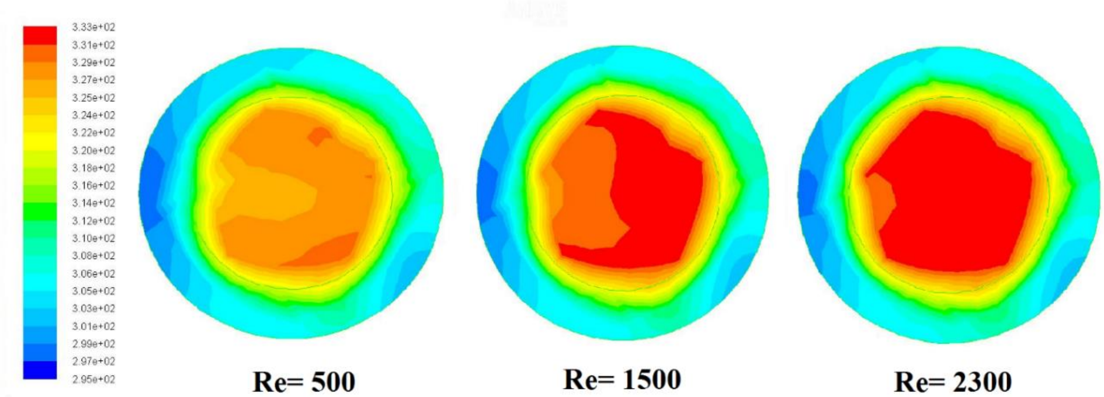

Figure 4.10: The temperature contours for tube at 180^{o}angle of length cross-section with different Reynold number

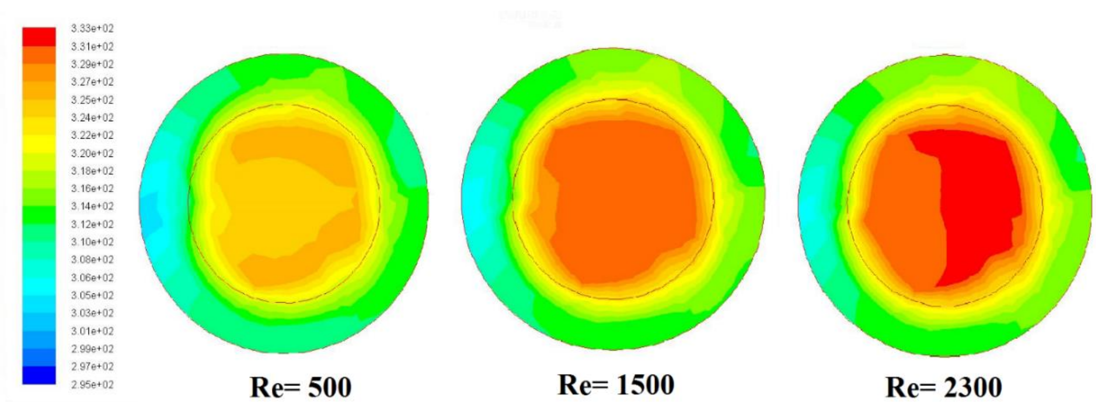

Figure 4.11: The temperature contours at outlet with different Reynolds numbers

Temperature contours at 700 cm3/min annulus flow rates

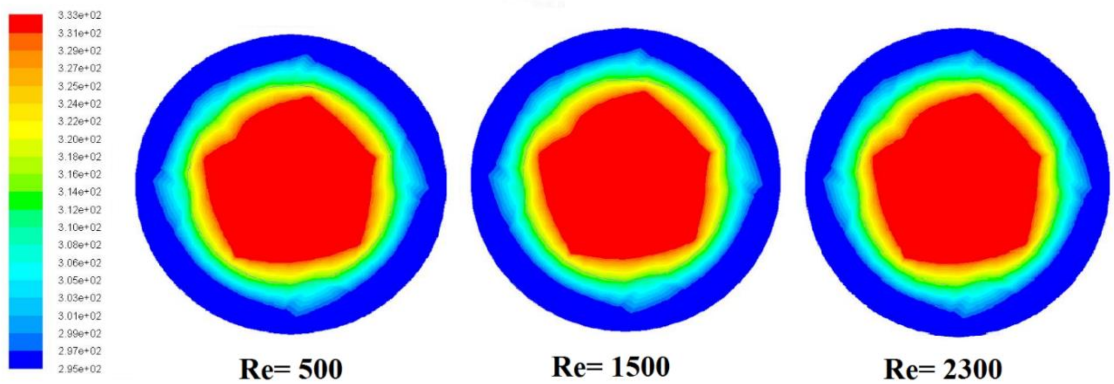

Figure 4.12: The temperature contours at the inlet

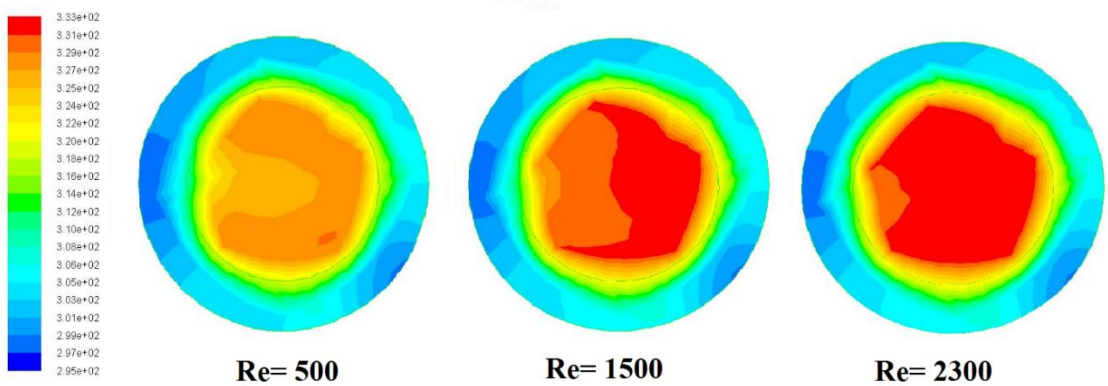

Figure 4.13: The temperature contours for tube at 180°angle of length cross-section with different Reynold number

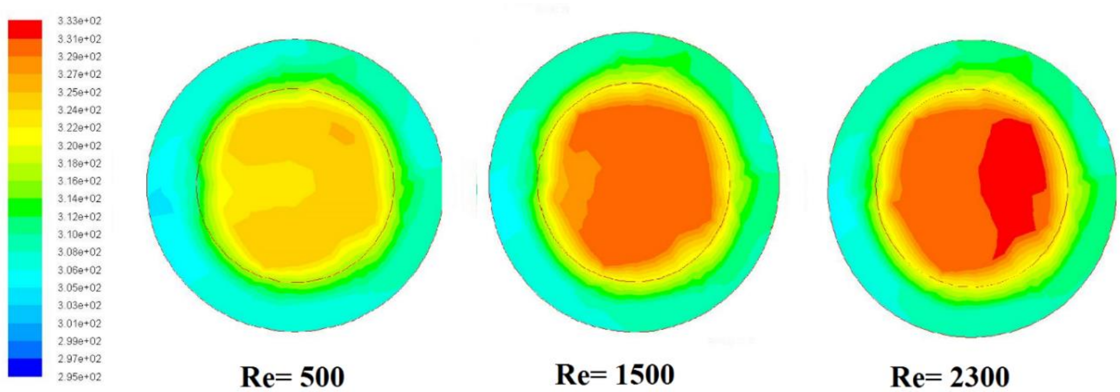

Figure 4.14: The temperature contours at outlet with different Reynolds numbers

The principle behind it is same like the others with different flow rates at annulus region but same at inner tubes region. It can be refers to the temperature contours between figure 4.9 until figure 4.14. The higher the flow rates in annulus give better heat transfers for tube-in-tube helical coiled heat exchanger.

4.2 Effects of Geometry

In this project, it is emphasizing on the evaluation of performance for various cross section of helical coil such as circular concentric, circular eccentric, square concentric and square eccentric cross section. The results shows a comparison between concentric and eccentric overall heat transfer coefficient against Dean Number inside the inner tubes of helical coil. For every model developed, simulation has been done with differences flow rates according to the fixed Reynolds number which are 500, 1000, 1500, 2000 and 2300. So, the figures 4.15 & 4.16 shows the result from calculation that was done in order to find overall heat transfer coefficient.

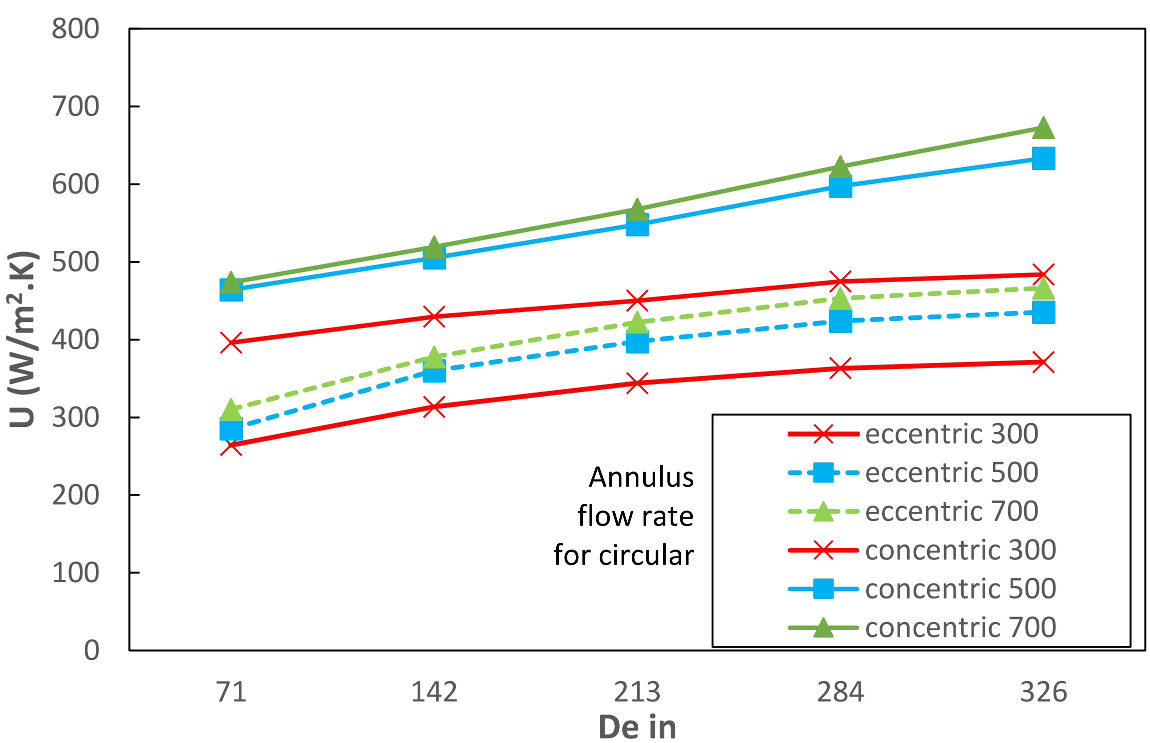

Figure 4.15: Comparison of circular tubes overall heat transfer coefficient against Dean Number in inner tubes between concentric and eccentric helical coil

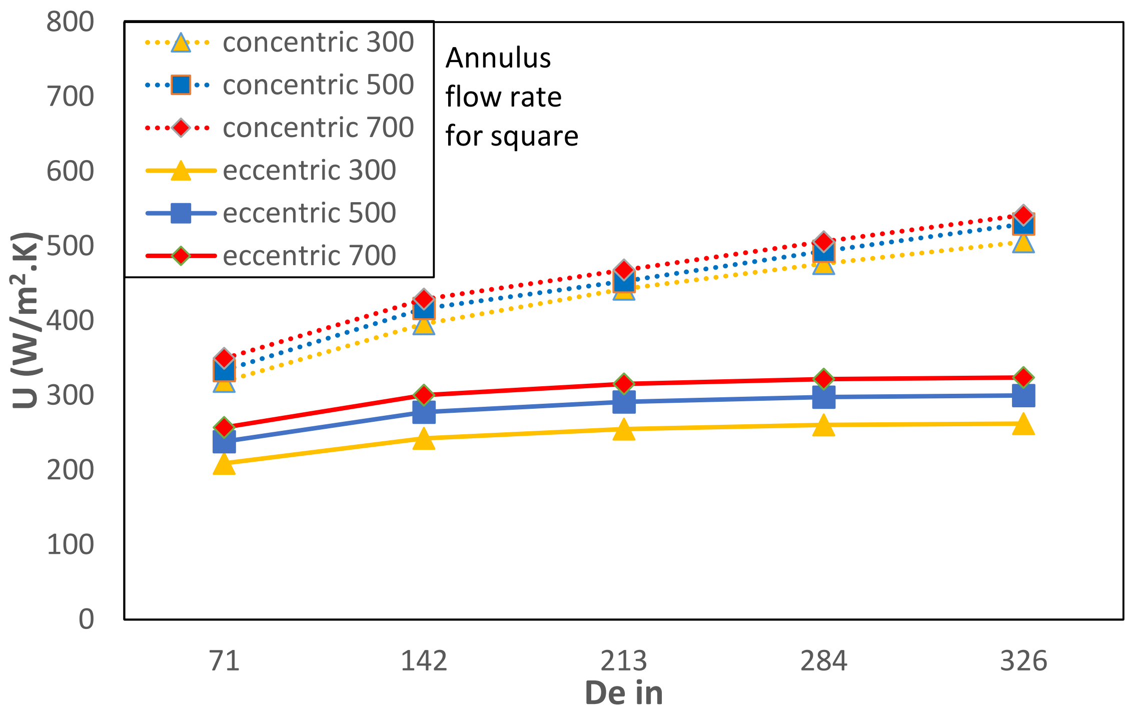

Figure 4.16: Comparison of square ducts overall heat transfer coefficient against Dean Number in inner ducts between concentric and eccentric helical coil

Figure 4.15 illustrates that the overall heat transfer coefficient increases with increasing the Dean numbers in inner tubes of circular helical coil heat exchanger. It is also shows that with increasing the annulus flow rates give higher values of overall heat transfer coefficient. But when compared between concentric and eccentric, it indicates that the circular concentric has higher overall heat transfer coefficient rather than circular eccentric helical coil geometry. Same goes to square ducts helical coil geometry whereas in figure 4.16, square concentric shows higher values when compared to square eccentric in terms of overall heat transfer coefficient. It is same like circular tubes in terms of annulus flow rates and Dean Numbers in inner tubes that will also increases overall heat transfer coefficient. The reason is possibly that the eccentricity causes the temperature at the wall becoming not consistence, and heat transfer turns worse in the lowering wall temperature region compared with the eccentric one (Zhang et al., n.d.).

4.2.1 The influence of geometry on heat transfer

Influence of heat transfer between circular concentric and square concentric cross section is graphed in this section.

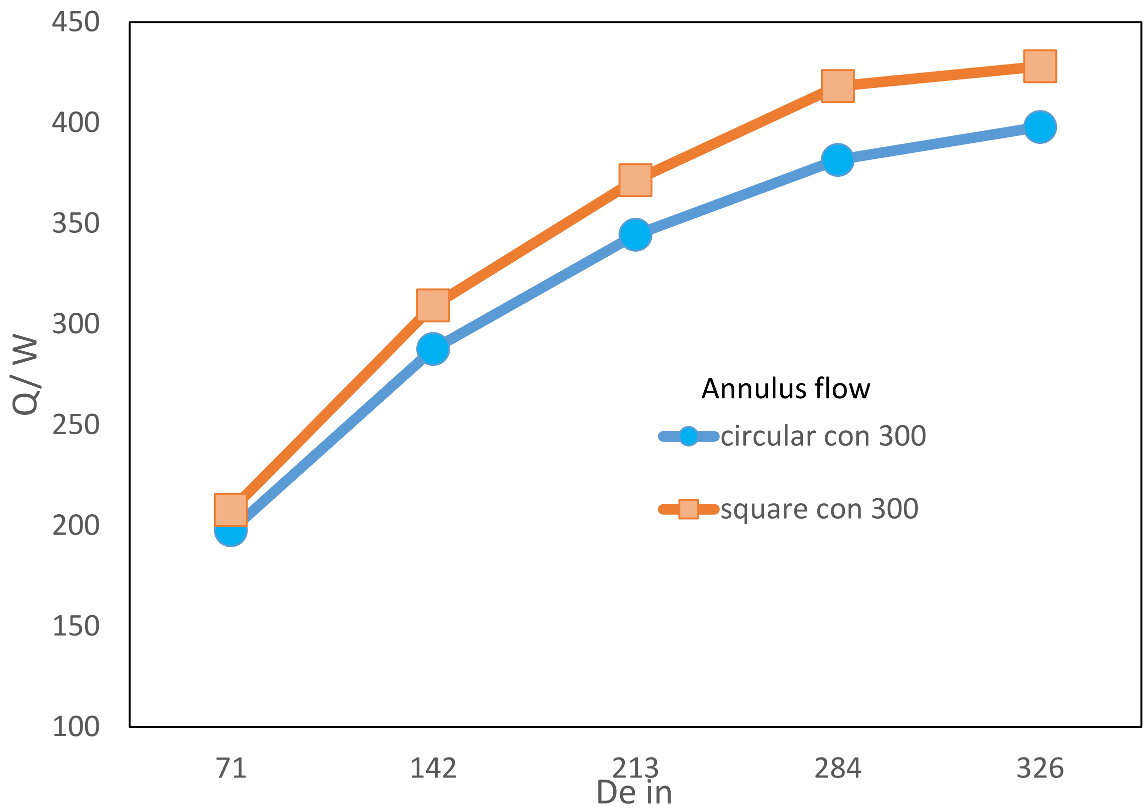

Figure 4.17: Comparison of heat transfer performance of different cross section between square and circular

It is show that the square concentric cross section has higher heat transfer when compare to the circular concentric cross section. It is probably because of the square cross section has bigger contact of surface area when compare to circular one even though the cross section was fixed. Study from Kurnia et al. (2012) that evaluate the heat transfer performance between several cross-sections that lead of square cross-section was higher heat transfer when compared to circular cross-section. However, for that study the higher heat transfer was triangular and rectangular cross-sections than others.

4.2.2 Flow characteristics

The flow characteristics in the inner tubes of various cross-sections geometries has shown in this section.

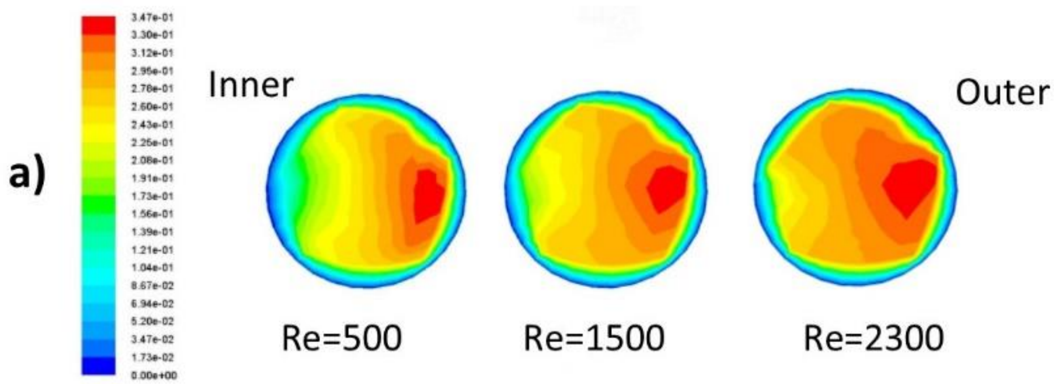

Figure 4.18: Velocity contours in inner tubes of circular concentric cross-section

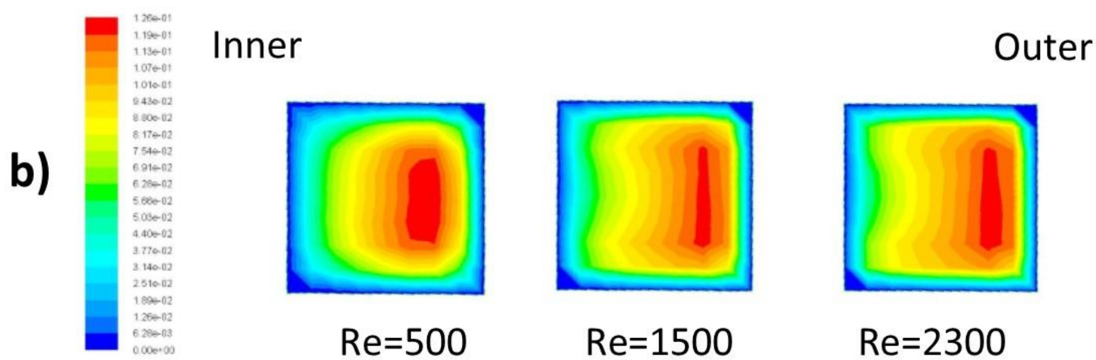

Figure 4.19: Velocity contours in inner ducts of square concentric cross-section

The numerical study for flow and thermal fields were conducted on parallel flow of helical heat exchanger with several cross-sections geometries such as circular and square concentric, circular and square eccentric. The developments of axial velocity and temperature fields were carried out for one turn of the heat exchanger as shown in figure above. For axial velocity the results indicate the flow at the 180^0 length of curvature for different cross-sections geometry of double pipe helical heat exchanger. Due to the effect of centrifugal forces, the flow in the core of the pipe begins to be forced to the outer bend, (Kumar et al., 2006). As the axial angle increases, the unbalanced centrifugal force of the main flow result in the change of the maximum velocity fields to the outer wall.

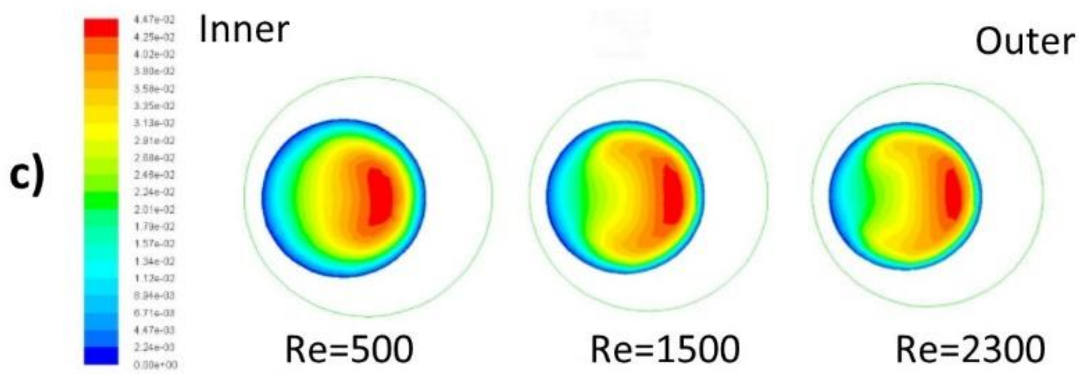

Figure 4.20: Velocity contours in inner tubes of circular eccentric cross-section

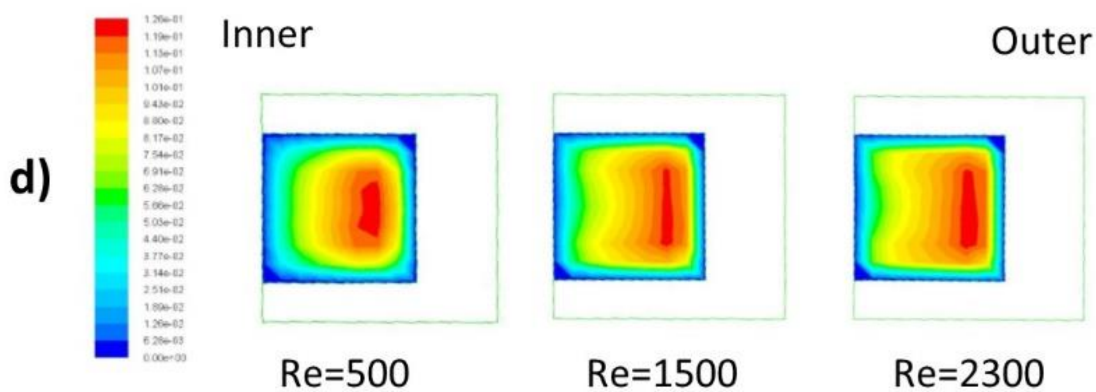

Figure 4.21: Velocity contours in inner ducts of square eccentric cross-section

The comparison between different cross-sections geometry with fixed cross section area has been done and the results show that the secondary flow is higher at square cross section than circular cross section. One possible reason to this phenomena where same Reynolds number were carried out is the cross sections that have lower hydraulic diameter will lead to higher mass flow rate and hence higher velocity, (Sasmito et al., 2012).

CHAPTER 5: CONCLUSION

This project is to evaluate the performance of helical coil heat exchanger. Therefore, the objective is to study flow characteristics and heat transfer phenomena in helical heat exchanger. So the numerical study of tube-in-tube helical coil was performed with various cross section geometries like circular concentric, square concentric, circular eccentric and square eccentric double pipes helical heat exchanger. It was found that along the curvature region, the location of maximum axial flow moves from the center of the curved pipes towards the outer wall. It is due to the centrifugal forces from curvature that enhance the secondary flow occurred. Even though there were different in cross section geometries, but in the inner tubes it’s indicates same flow characteristics.

Other than that, the objective of this project is to evaluate the several parameter affecting the performance of the helical heat exchanger. The flow rates in the inner tube and annulus region were both varied and the co-flow configuration is tested. The numerically simulate obtained overall heat transfer coefficient (U) for different values of flow rate in the inner coiled tube and annulus region. It is reported that the overall heat transfer coefficient increases with increase in the inner coiled tube Dean number for constant flow rate in the annulus region. Similar trends in the variation of overall heat transfer coefficient were observed for different flow rate in annulus region with same flow rate in the inner-coiled tube. It is applied to all cross sections geometries that was studied in this project. However, when compared the overall heat transfer coefficient between concentric and eccentric, the result shows that whether it is square or circular, the concentric is higher compared to eccentric. It is because the flow in the eccentric tubes were not uniformly obtained. Besides, for different cross sections between square ducts and circular tubes, the result show that square ducts offer higher heat transfer compared to circular tube double pipe heat exchanger.

5.1 Recommendations

This project may have many things to be consider about, unfortunately because of time constrain which become limitation to fulfill the project requirement. Therefore, there are several recommendation that can be done in future studies to ensure this project is well done enough. First of all is the mesh independent test. It is good work to identify the exact number of meshes needed for each geometries to ensure for getting the right result. It will become inconsistence result if this method is neglect. In addition, this project only done for co-flow in the helical heat exchanger. It will be good comparison if we can do at the same time with counter-flow. We can see the correlation of performance between this two types of flow.

Furthermore, based on this study for helical heat exchanger square cross-section, the experimental rig should be conducted in order to verify it. So, we can find whether the result from simulation is perfectly fit or not with experimental.

We also can improved this project by comparing with differences cross sections. For instance, we can used side-by-side cross section and square with double triangular inside of square cross section. It is because the triangular cross section has higher heat transfer compared to other.

Hopefully with this recommendation will lead this project into the best contribution in engineering fields.

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