Numerical Simulation of Natural Convection Between Two Parallel Plates

by

Aishah binti Rosli

Dissertation submitted in partial fulfilment of
the requirements for the
Bachelor of Engineering (Hons)
(Chemical Engineering)

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CERTIFICATION OF APPROVAL

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Approved by,

(Dr. Rajashekhar Pendyala)

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

September 2011

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.

AISHAH BINTI ROSLI

ABSTRACT

Free convection flow is present and significant in various engineering circumstances, such as the design of heat exchangers, nuclear reactors, and many chemical processes. It is also important in the application of cooling equipment. In this project, a two-dimensional numerical model is formulated to simulate the natural convection of air between two parallel plates using a computational fluid dynamics (CFD) code. Several different situations are modeled to observe the effects on the natural convection flow.

ACKNOWLEDGEMENT

First and foremost, I would like to thank my supervisor of this project, Dr. Rajashekhar Pendyala for the valuable guidance and advice. He has inspired me greatly to work on this project, and his willingness to help and motivate me has contributed greatly to this project.

I would also like to thank my family and friends for their understanding and support throughout the completion of the project. Words are inadequate in offering the thanks to all the other related parties for giving their full support and efforts in completing the project from the starting stage of the project to the end of the semester.

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CHAPTER 1

INTRODUCTION

1.1 Background of Study

Natural convection, also known as free convection, is caused by density changes that are caused by a heating process that results in the movement of the fluid. The heating process causes the density of the fluid close to the heat-transfer surface to decrease, while cooling causes the density of the fluid to increase, which then causes the fluid to move in free convection due to the buoyancy forces that is imposed on the fluid. These forces, which give rise to the free-convection currents are called body forces, and would not be present if some external force field did not act upon the fluid (Holman, 2001).

In free convection, at the heated body, which is the wall of the plate in this case, the velocity is zero according to the no-slip boundary condition. The velocity then increases very quickly in a thin boundary layer adjacent to the body until it reaches a maximum value and then, far away from the body, it becomes zero again. At first, the boundary-layer development is linear, but, depending on the properties of the fluid and the difference in temperature between the two plates, turbulent eddies are formed and transition to a turbulent boundary layer begins at a distance away from the leading edge (Pitts, 1998).

The differential equation of motion for the boundary layer should be obtained in order to analyze the heat-transfer problem. Similar to forced-convection problems, some natural convection problems require for the relations for heat transfer in different situations to be obtained from experimental measurements. Usually, these circumstances involve those in which it is difficult to predict temperature and velocity profiles analytically (Holman, 2001).

1.2 Problem Statement

Many studies have been done to observe the transient behavior of natural convection between vertical parallel plates in various conditions, some done experimentally while some provides analytical solutions for the problems. Numerical solutions, on the other hand, can be used to solve complex problems, unlike analytical solutions, which are only available for simple problems.

This project is aimed to solve some natural convection problems using Computational Fluid Dynamics (CFD) to generate numerical solutions to the problem and compare these solutions with the analytical solutions and experimental findings available in the literature.

1.3 Objectives and Scope of Study

1.3.1 Objectives

- To identify various boundary conditions in which natural convection between vertical parallel plates can occur.
- Develop a simulation method using CFD to obtain numerical solutions for each condition.

1.3.2 Scope of Study

There are many situations in which natural convection can occur and depending on the situation, the behavior of the flow will vary. Therefore, the scope of study will be limited to:

- Flow between vertical parallel plates
- Flow of incompressible fluid

1.4 Significance of the Study

Transient natural convection flow between vertical parallel surfaces is applied in technological processes such as the early stages of melting as well as in heating of insulating air gaps by inserting heat at the furnaces start-up (Jha, 2010). The effects of both thermal and mass buoyancy forces on free convection flows are also significant in many engineering situations; in the design of heat exchangers, nuclear reactors, solar energy collectors thermo protection systems and other chemical processes (Marneni, 2008). These flows are also applied in the cooling of electric and electronic equipment, nuclear reactor fuel elements and home ventilation (Badr, 2006). Free convection is a preferable heat transfer technique because of its simplicity, reliability and cost effectiveness (Yilmaz, 2007).

1.5 Feasibility of Study

Throughout this study there are several phases that will be done throughout completing the project:

I. Research based on literature review on natural convection from multiple references and sources. This phase involved doing researches within the limit of scopes of the project in order to built strong foundation on the theoretical part before proceeding with the next phases of the project.

II. Identifying related data used for the simulation process. This phase involved collecting all the required data needed before proceeding with the simulation.

III. Testing, comparing and modifying collected data. This data that are previously collected from various trusted sources and references will be used as comparison with the data obtained from the simulation.

CHAPTER 2

LITERATURE REVIEW

2.1 Natural Convection between Parallel Plates

Free convection between vertical walls has been studied extensively due to its importance in various engineering applications. Studies have been done on this problem in various conditions.

2.1.1 Natural Convection with One Plate Isothermally Heated and the other Thermally Insulated

Jha and Ajibade (2010) (Jha, 2010) studied the natural convection flow between two infinite vertical parallel plates with one plate isothermally heated and the other one thermally insulated. From the study, solutions were obtained for velocity and temperature fields in the form of convergent infinite series for two cases; Prandtl number $\neq 1$ and Prandtl number = 1.

Case I: Pr ≠ 1

$$\begin{aligned} u(y,t) &= \frac{t}{(\mathrm{Pr}-1)}\sum_{n=0}^{\infty}(\{F_1(b_3;1.0)-F_1(b_4;1.0) \\ &-(-1)^n[F_1(b_3;\mathrm{Pr})+F_1(b_4;\mathrm{Pr})]\} \\ &+2\sum_{m=0}^{\infty}(-1)^n[F_1(b_1;1.0)-F_1(b_2;1.0)]) \end{aligned}$$

$$T(y,t)=\sum_{n=0}^{\infty}(-1)^n[erfc(\frac{a_3}{2}\sqrt{\frac{Pr}{t}})+erfc(\frac{a_4}{2}\sqrt{\frac{Pr}{t}})]$$

Case II: Pr = 1

$$u(y,t) = \sum_{n=0}^{\infty} \frac{(-1)^n}{2}(1-y)[F_3(a_3;t) - F_3(a_4;t)]$$

$$T(y,t) = \sum_{n=0}^{\infty}(-1)^n \left[erfc\left(\frac{a_3}{2\sqrt{t}}\right) + erfc\left(\frac{a_4}{2\sqrt{t}}\right)\right]$$

The functional a_1, a_2, a_3, a_4, b_1, b_2, b_3, b_4, F_1, F_2 and F_3 used in the above equations are defined in Appendix II.

2.1.2 Natural Convection with Constant Temperature and Mass Diffusion at One Boundary

Marneni (2008) has used the Laplace-transform technique to solve the transient natural convection flow between two vertical parallel plates with one boundary having constant temperature and mass diffusion. Velocity profiles are obtained from the study for two cases; Schmidt number $\neq 1$ and Schmidt number =1. In addition, the effect of different parameters such as the buoyancy ratio, Schmidt number, Prandtl number and time are studied.

2.1.3 Turbulent Natural Flow with Rayleigh Number Ranging from 10^5 to 10^7

Badr, Habib, Anwar, Ben-Mansour and Said (2006) investigated buoyancy driven turbulent natural convection in vertical parallel plate channels that covers Rayleigh numbers ranging from 10^5 to 10^7 and focuses on the effect of the channel geometry and the Nusselt number. This problem was investigated for two cases; isothermal and isoflux heating conditions. From this study, two new correlations were obtained for the average Nusselt number in terms of the Rayleigh number and channel aspect ratio.

2.1.4 Natural Convection with Walls Heated Asymmetrically

Natural convection in a vertical parallel plate channel with asymmetric heating was studied experimentally and numerically by Yilmaz and Frazer (2007) using a laser Doppler anemometer (LDA) in the experiment. The investigation was done with one wall maintained at uniform temperature and the opposing wall made of glass. Three different LRN k-ε turbulence models were used in the numerical calculations, and correlation equations were developed for average heat transfer and induced flow rate using the numerical results.

Other than Yilmaz *et al.* (2007), Singh and Paul (2006) have also studied natural convection between two parallel plates with asymmetric heating, but focuses on the effect of the Prandtl number and buoyancy force distribution parameter. This study used the Laplace transform method to find the solutions for the velocity and temperature fields. From this study, it has been found that the symmetric/asymmetric nature of the flow formation can be obtained by giving a suitable value to the buoyancy force distribution parameter.

2.1.5 Natural Convection Flows for Low-Prandtl Fluids

While many of the studies in the literature deals with air and water as the fluid, Campo, Manca and Morrone (2006) dealt with the natural convection of metallic fluids, with Prandtl number of less than 1 between vertical parallel plates with isoflux heating. Correlation equations for the induced flow rate, maximum wall temperatures, and average Nusselt numbers were obtained as functions of the main thermal Grashof or Rayleigh numbers and geometrical parameters.

2.1.6 Turbulent Natural Convection with Symmetric and Asymmetric Heating

Habib, Said, Ahmed and Asghar (2002) investigated the velocity characteristics of free convection in both symmetrically and asymmetrically heated vertical channels. This study, done experimentally, showed that for the symmetrical flow case, the measurements indicate high velocity gradient at the shear layer close to the hot wall and a region of reversed flow at the center of the channel close to the channel exit as illustrated in Figure 2.1 below. For asymmetrical flow, the results showed a large vortex with flow upward the hot wall and down the cold wall with a wider boundary layer close to the hot wall compared to the cold wall boundary layer as presented in Figure 2.2.

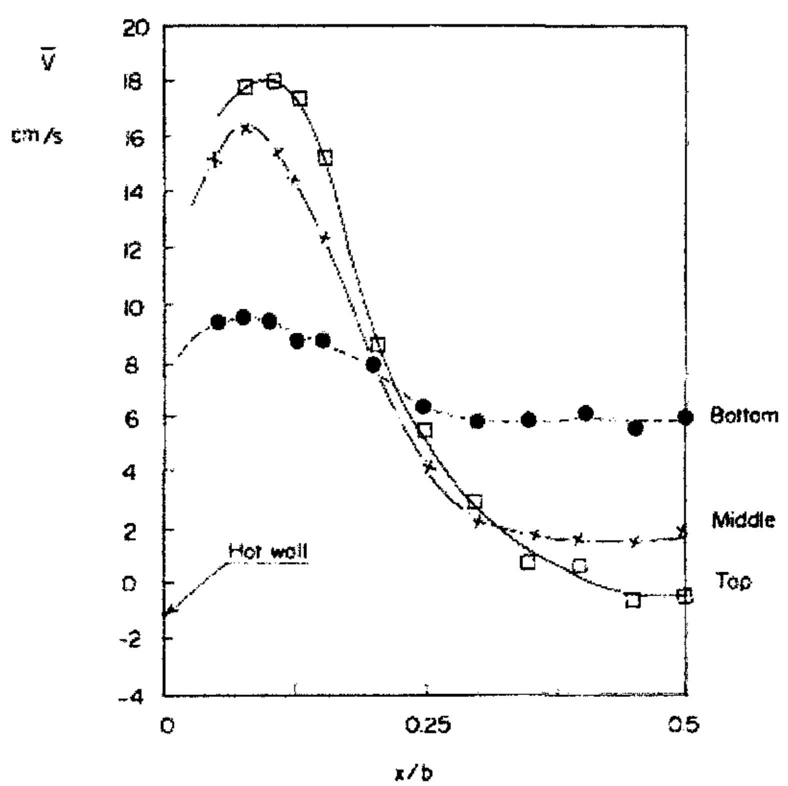

Figure 1: Profiles of the mean vertical velocity component of the symmetrical flow case, *Ra* = *4.0 x 10*^{6}: □ *y/L = 0.98; x, y/L = 0.55;* •, *y/L = 0.11.*

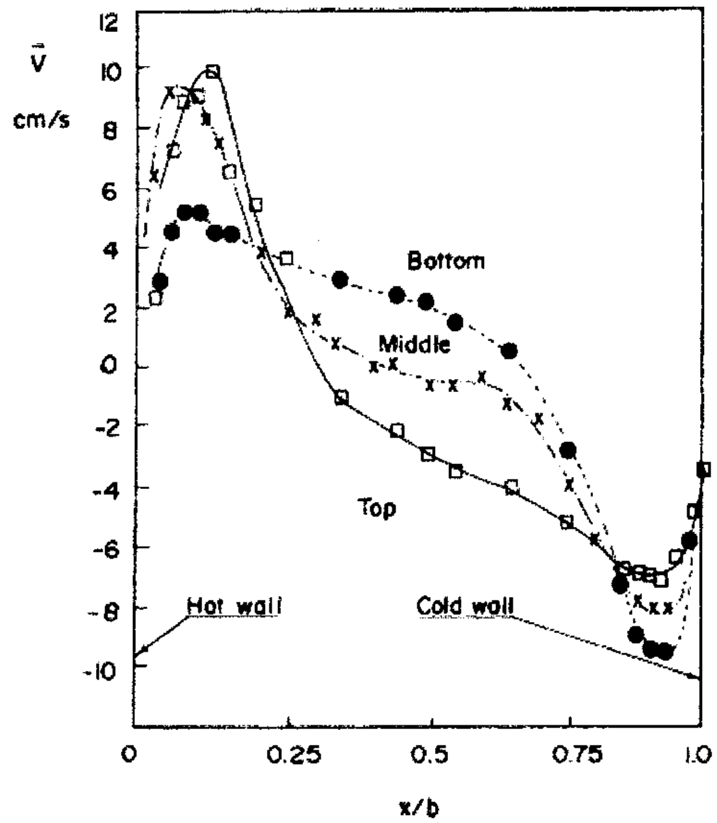

Figure 2: Profiles of the mean vertical velocity component of the asymmetrical flow case, $Ra = 2.0 \times 10^6$: □ $y/L = 0.98$; x, $y/L = 0.55$; •, $y/L = 0.11$.

Another study on natural convection in an asymmetrically heated vertical parallel plate channel was done by Fedorov and Viskanta (1997). Scaling relations for the Reynolds and average Nusselt numbers have been developed in terms of relevant dimensionless parameters in this study.

2.1.7 Laminar Flow with UWT/UWC and UHF/UMF

Earlier, Lee (1999) studied natural convection heat and mass transfer between vertical parallel plates with both unheated entry and unheated exit. Both boundary conditions of uniform wall temperature/ uniform wall concentration (UWT/UWC) and uniform heat flux/ uniform mass flux (UHF/UMF) were considered in this study. Analytical solutions of the dimensionless volume flow rate, the average Nusselt number and the average Sherwood number were derived under fully developed conditions for the boundary conditions considered.

2.1.8 Natural Convection with Asymmetric Heating Coupled with Thermal Radiation

Cheng and Müller (1997) also studied natural convection with asymmetric heating, but with thermal radiation considered as well. Based on the experimental and numerical results of this study, a semi-empirical correlation was developed to describe the heat transfer of turbulent natural convection coupled with thermal radiation in a channel with one-sided heated wall.

CHAPTER 3

METHODOLOGY

3.1 Research and Project Activities

3.1.1 Research

Comprehensive literature study will be done to identify situations and conditions in which natural convection can occur. Many studies provide analytical solutions, which are only available for simple problems, while numerical solutions can be used to solve more complex situations. These analytical solutions and experimental results are to be used later for comparison with the numerical solutions obtained in this project to estimate the deviation between the solutions. Preparing literature review is vital at the very early phase of the project since it can create a good foundation and improves understanding towards completing the project.

3.1.2 Simulation of Problems

All required data are collected from several sources during the early phase of the project. After identifying and choosing the conditions for the problems, a simulation using computational fluid dynamics (CFD) code, FLUENT will be done for each problem to obtain numerical solutions for the problems and to get related profiles such as velocity and temperature profiles.

3.1.3 Comparison and Verification

Outputs gained from the simulation run are analysed and compared to those output data presented in several sources collected before from the previous phase of the project. The input data are modified accordingly with respect to the objective of the project that is to study the effect of operating conditions on the efficiency of the process. After the numerical solutions and the appropriate profiles are obtained, these solutions are then compared to the analytical solutions found in the literature as well as experimental results if it is available.

3.1.4 Conclusion and Report Writing

A conclusion is justified based on the output data collected from the simulation runs. All these data and analysis involved throughout the completion of the project are summarized into a complete documentation of final report thesis writing.

3.2 Gantt Chart and Key Milestones

Table 1: Gantt Chart and Key Milestones for FYP I

No.	Detail/ Week	1	2	3	4	5	6		7	8	9	10	11	12	13	14
1	Selection of Project Topic: Simulation of Natural Convection Between Parallel Plates Using CFD							Mid - Semester Break								
2	Preliminary Research Work															
3	Submission of Extended Proposal															
4	Proposal Defence															
5	Project Work Continues															
6	Submission of Interim Draft Report to Supervisor															
7	Submission of Final Report to Coordinator															

Process

Key Milestone

Table 2: Gantt Chart and Key Milestones for FYP II

No.	Detail/ Week	1	2	3	4	5	6	7		8	9	10	11	12	13	14	15
1	Project Work Continues																
2	Submission of Progress Report									●							
3	Project Work Continues																
4	Pre-SEDEX												●				
5	Submission of Draft Report								Mid - Semester Break					●			
6	Submission of Dissertation (Soft Bound)														●		
7	Submission of Technical Paper														●		
8	Oral Presentation															●	
9	Submission of Project Dissertation (Hard Bound)																●

Process

● Key Milestone

CHAPTER 4

RESULT AND DISCUSSION

4.1 Model Equations

Consider a vertical, parallel-plate channel shown schematically in Figure 3. Air enters the channel at temperature T_o. In this case, the thermal buoyancy force is considered to be driving the fluid motion. The flow is considered incompressible and radiation heat transfer is neglected. The Navier-Stokes equations of motion for two-dimensional incompressible flow in a vertical channel can be written as:

Conservation of mass (continuity)

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

Conservation of x-momentum

$$\frac{\partial}{\partial x}(\rho uu) + \frac{\partial}{\partial y}(\rho uv) = -\frac{\partial p}{\partial x} + \frac{\partial}{\partial x}\left[(\mu + \mu_t)\frac{\partial u}{\partial x}\right]$$

$$+\frac{\partial}{\partial y}\left[(\mu + \mu_t)\frac{\partial u}{\partial y}\right] - \frac{2}{3}\rho\frac{\partial k}{\partial x}$$

Conservation of y-momentum

$$\frac{\partial}{\partial x}(\rho uv) + \frac{\partial}{\partial y}(\rho vv) = -\frac{\partial p}{\partial y} + \frac{\partial}{\partial x}\left[(\mu + \mu_t)\frac{\partial v}{\partial x}\right]$$

$$+\frac{\partial}{\partial y}\left[(\mu + \mu_t)\frac{\partial v}{\partial y}\right] - \frac{2}{3}\rho\frac{\partial k}{\partial y}$$

$$+ (\rho - \rho_o)g$$

Conservation of energy equation

$$\frac{\partial}{\partial x}(\rho uT) + \frac{\partial}{\partial y}(\rho vT) = \frac{\partial}{\partial x}\left[\left(\frac{k}{c_p} + \frac{\mu_t}{Pr_t}\right)\frac{\partial T}{\partial x}\right] + \frac{\partial}{\partial y}\left[\left(\frac{k}{c_p} + \frac{\mu_t}{Pr_t}\right)\frac{\partial T}{\partial y}\right]$$

4.2 Mesh

A mesh, or grid, to model the problem has been formulated in order to simulate the problem using ANSYS Workbench 12.1. The geometry of the model is as shown in Figure 3.

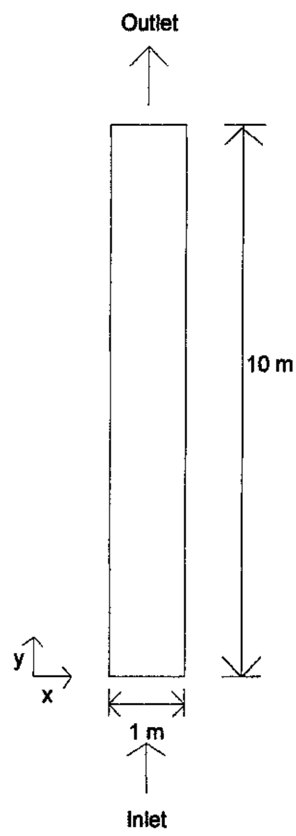

Figure 3: Geometry of the Model

4.2 Simulation

Different boundary conditions are imposed on the plates, and simulations were done in steady state and transient conditions. At steady state, the boundary conditions used were: both plates heated at different temperatures, the left plate heated while the right one is set at zero heat flux, and the right plate heated while the left one is set at zero heat flux. For transient condition simulation, one plate was isothermally heated while the other was adiabatic. For all the conditions, the fluid is set as air, and the inlet temperature at 300 K.

4.2.1 Case 1: Both Plates Set at Different Temperatures (Steady State)

In this case, the temperature of the left plate is set at 1000 K while the temperature of the right plate is set at 2000 K. The profiles obtained from the simulation are as shown below.

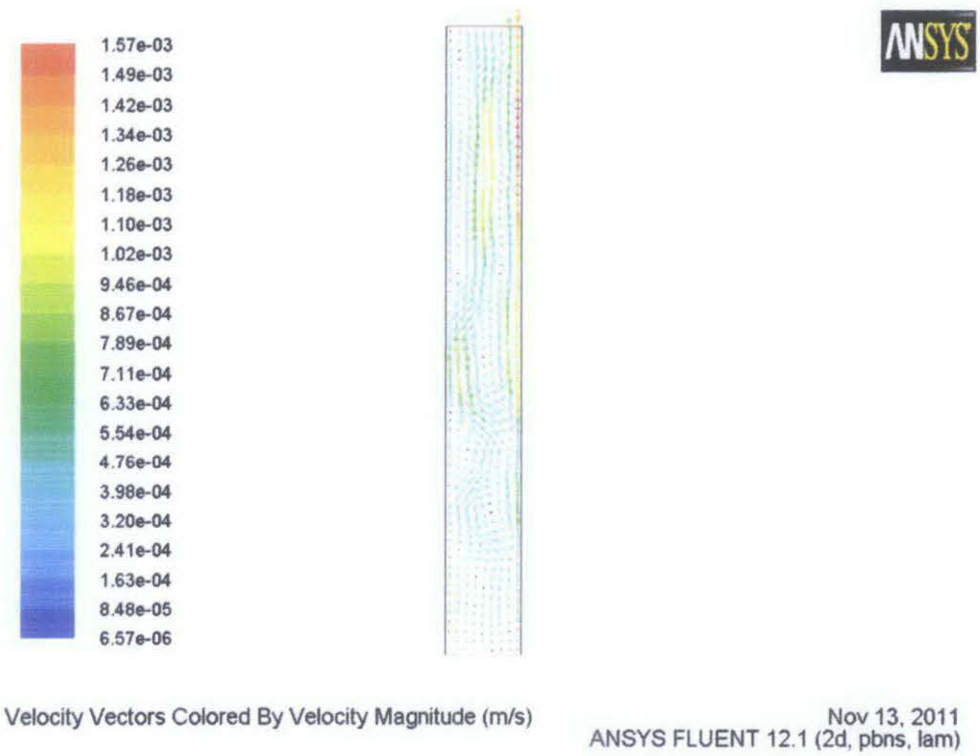

Figure 4: Velocity Vector for Case 1

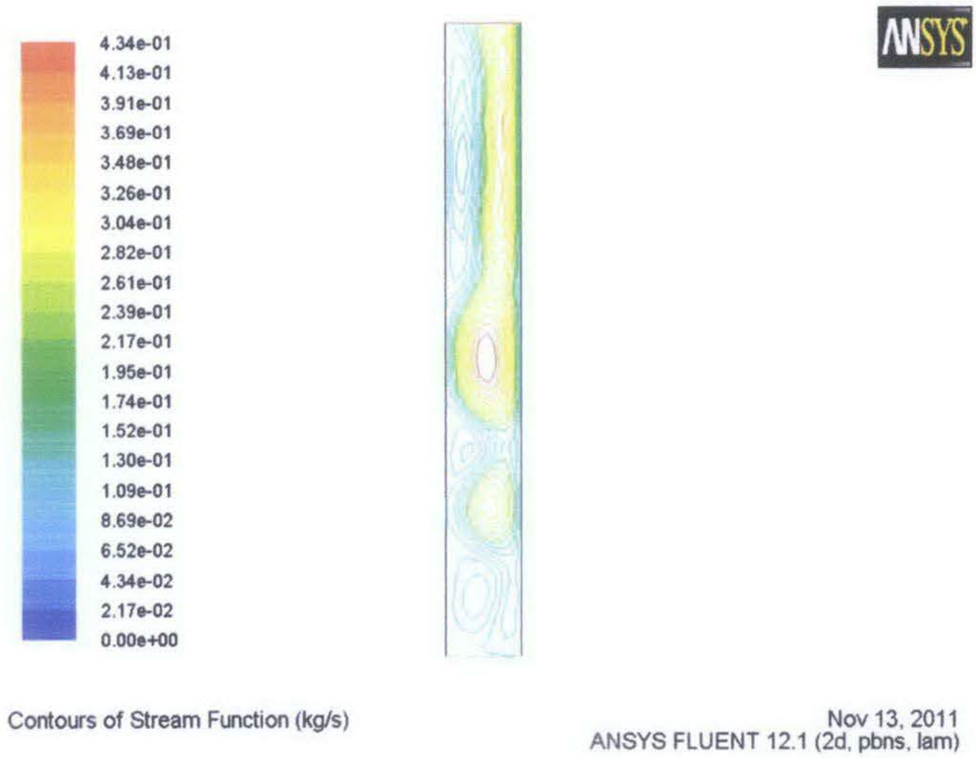

Figure 5: Stream Function for Case 1

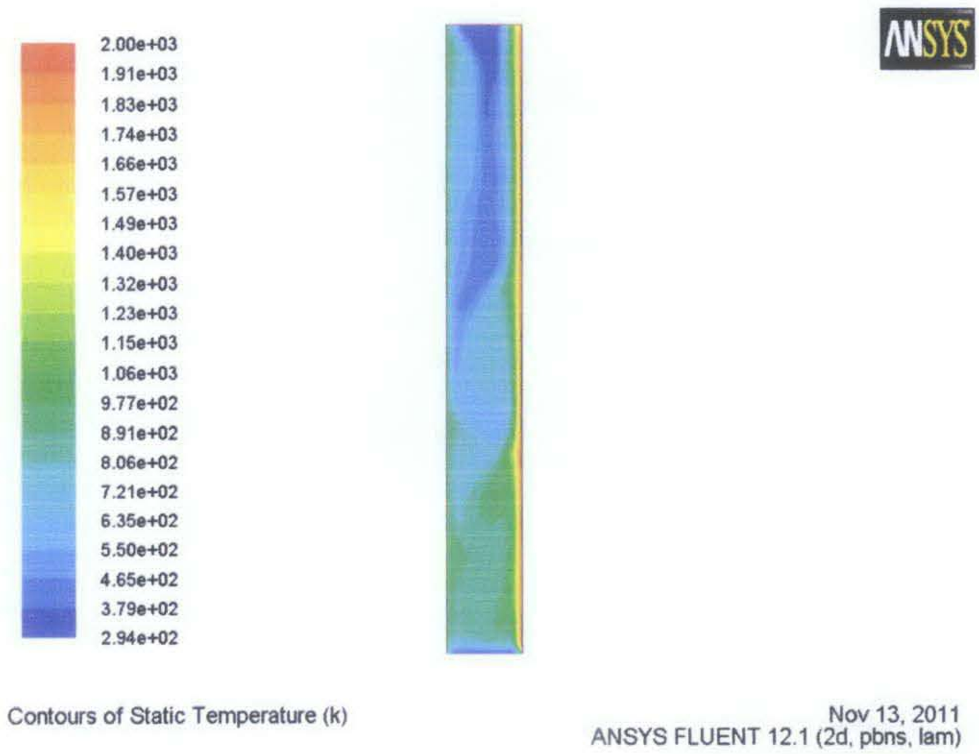

Figure 6: Temperature Profile for Case 1

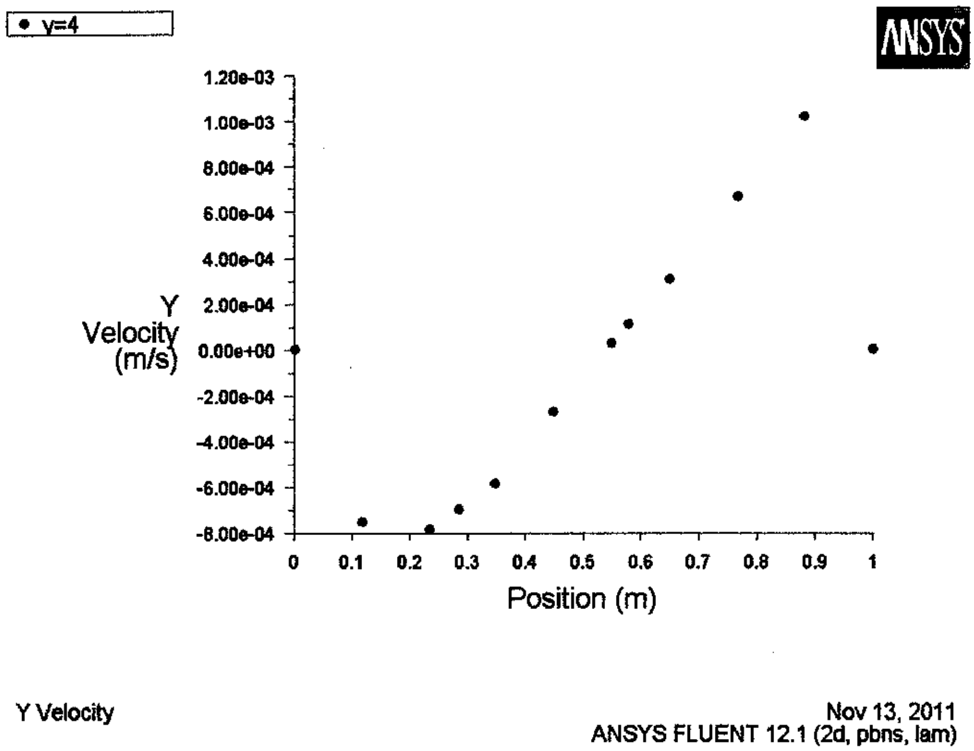

Figure 7: Velocity XY Plot for Case 1

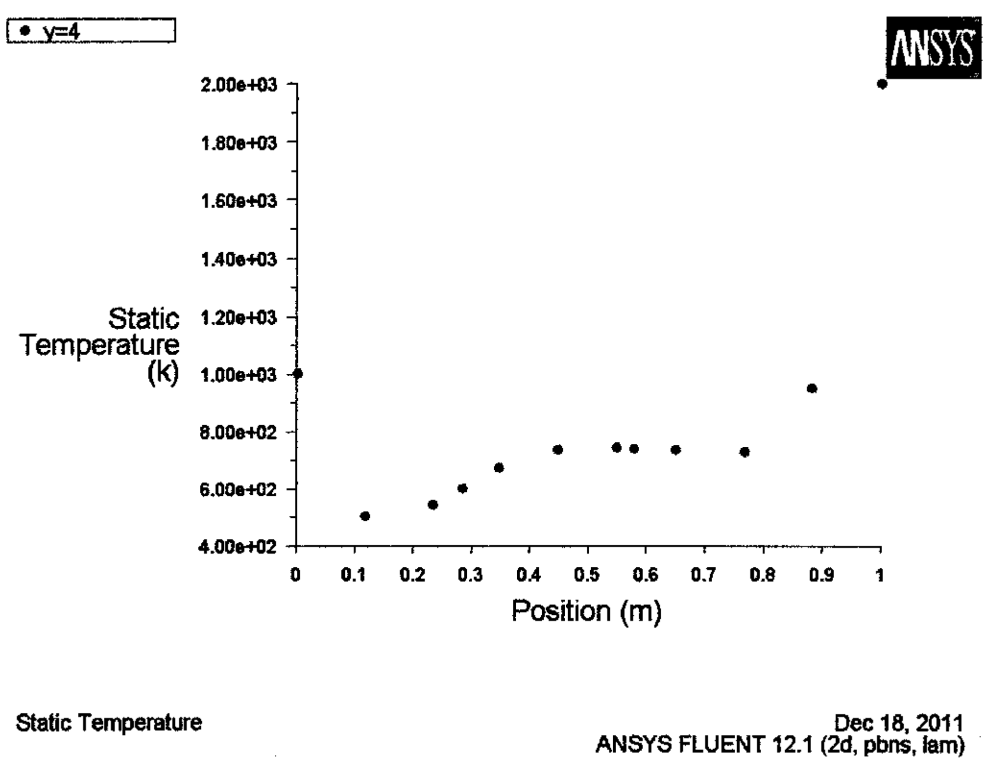

Figure 8: Temperature XY Plot for Case 1

From the velocity vector, it can be seen that the velocity of air is higher near the right plate, which is heated at a higher temperature, and that the velocity of air increases as it goes up the plate from the inlet to the outlet. The recirculatory patterns observed in the stream function are due to the natural convection between the plates. The temperature profile predictably shows a higher temperature near the right plate, which is heated at 2000 K as opposed to the left plate, which is heated at 1000 K. The temperature profile also shows a decrease in the temperature of air around the middle of the plates as it approaches the outlet. This may be caused by the reversed flow that occurs between the plates.

The velocity XY plot shows the velocity of air in around the middle of the plates, and it can be observed that for x = 0.55 to x = 1, reverse flow does not occur whereas it occurs for x = 0 to x = 0.55, due to the lower temperature of the left plate. The magnitude of the upward flow increases near the right plate while the magnitude of the downward flow decreases near the left wall as time increases. Finally at steady state, the upward flow was formed near the right wall and the downward flow near the left wall. Exactly at the right and left plates, the velocity is zero, which is true according to the no-slip boundary condition. The temperature XY plot at the same position shows a decrease of temperature as it moves away from the left plate, then starts to increase at approximately x = 0.45 as it approaches the right plate.

4.2.2 Case 2: Right Plate Heated, Left Plate at Zero Heat Flux (Steady State)

In this case, the temperature of the right plate is set at 1000 K while the left plate is set at zero heat flux. The profiles obtained from the simulation are as shown below.

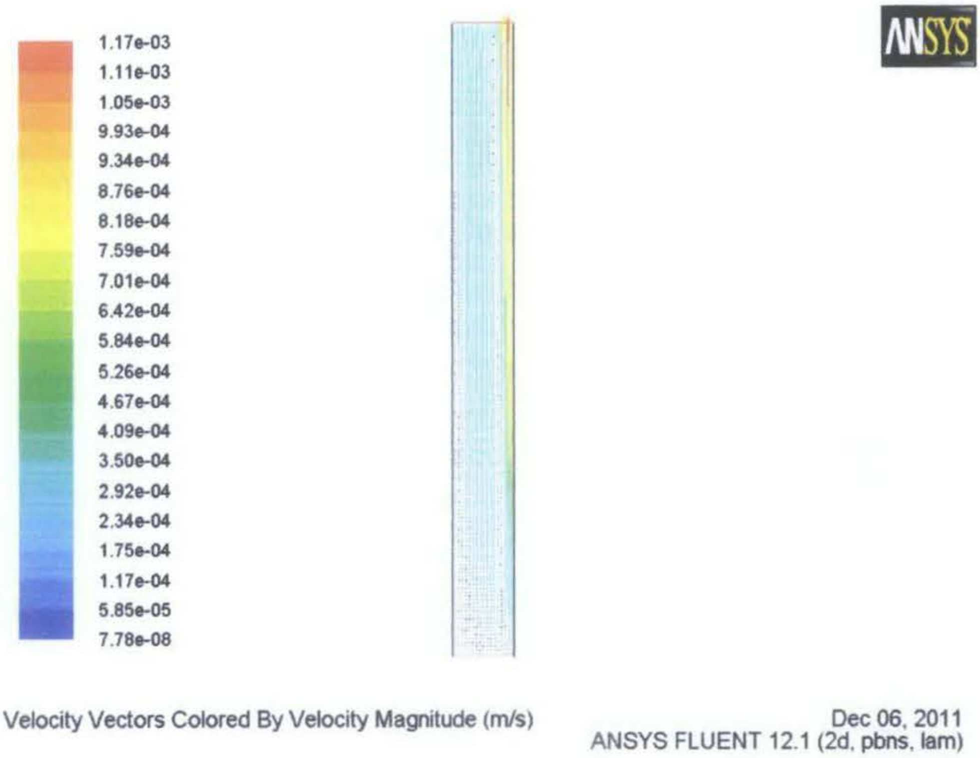

Figure 9: Velocity Vector for Case 2

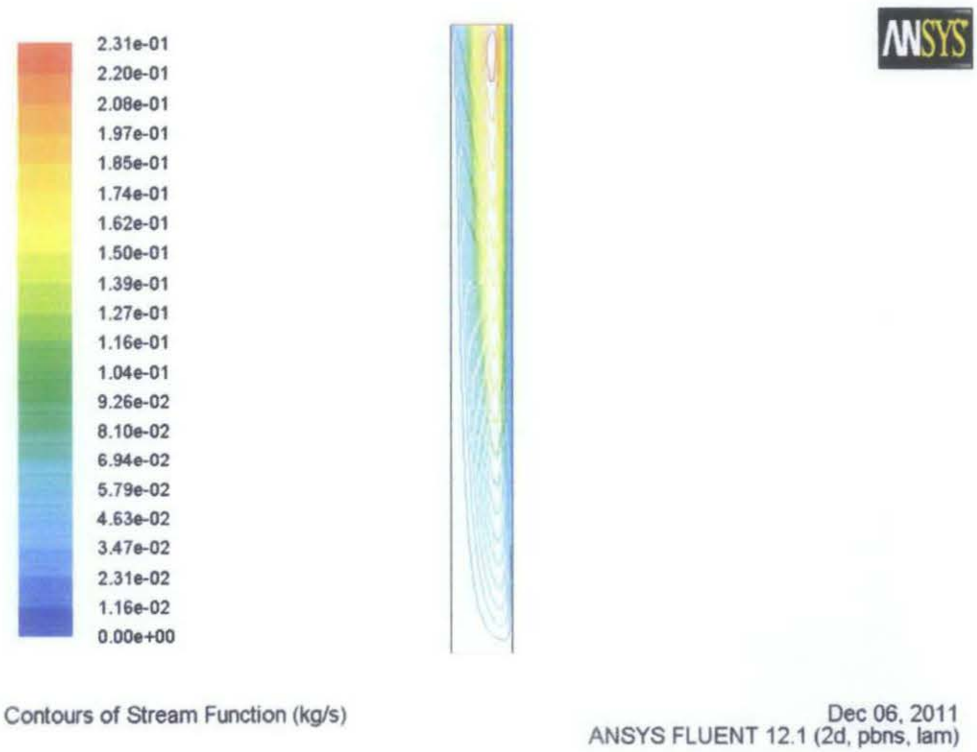

Figure 10: Stream Function for Case 2

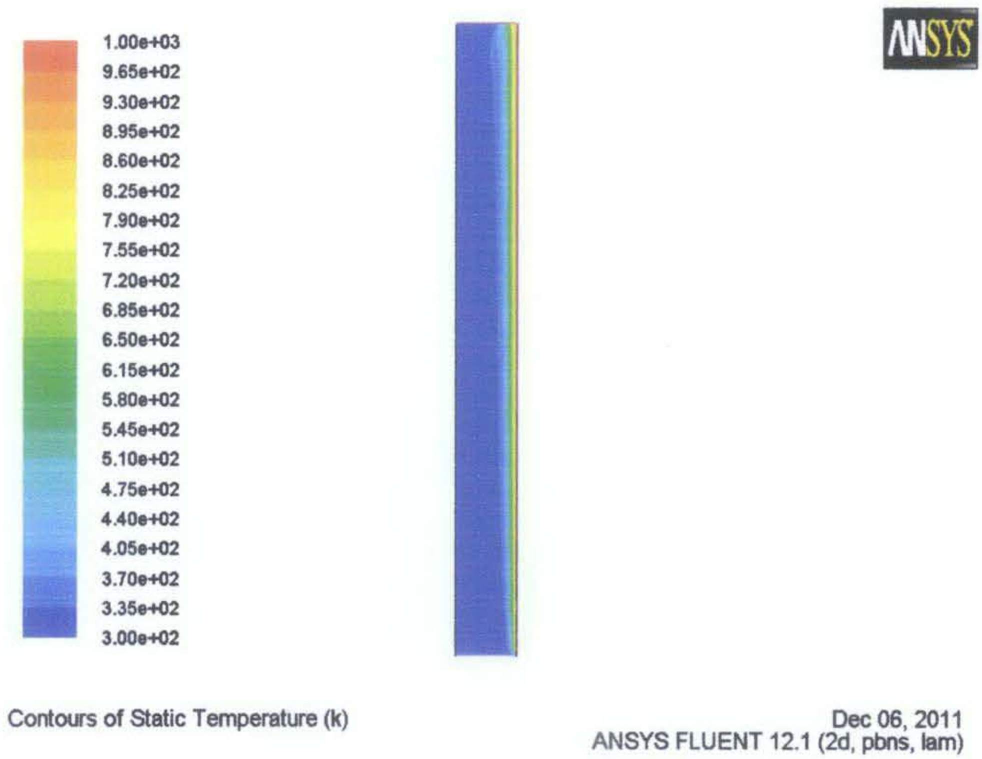

Figure 11: Temperature Profile for Case 2

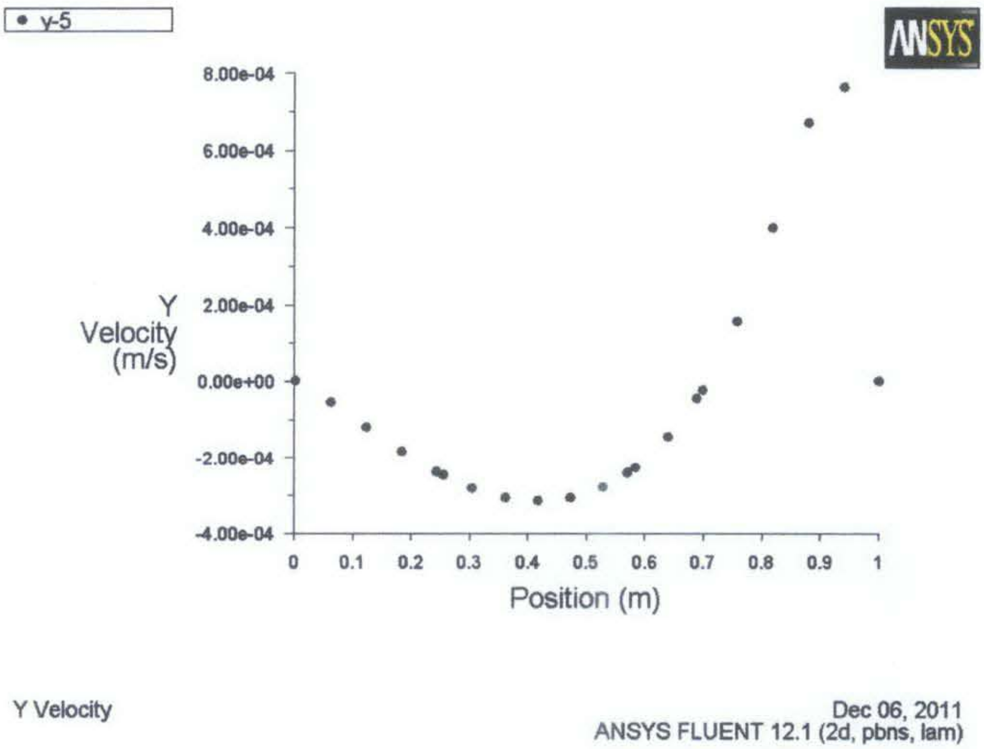

Figure 12: Velocity XY Plot for Case 2

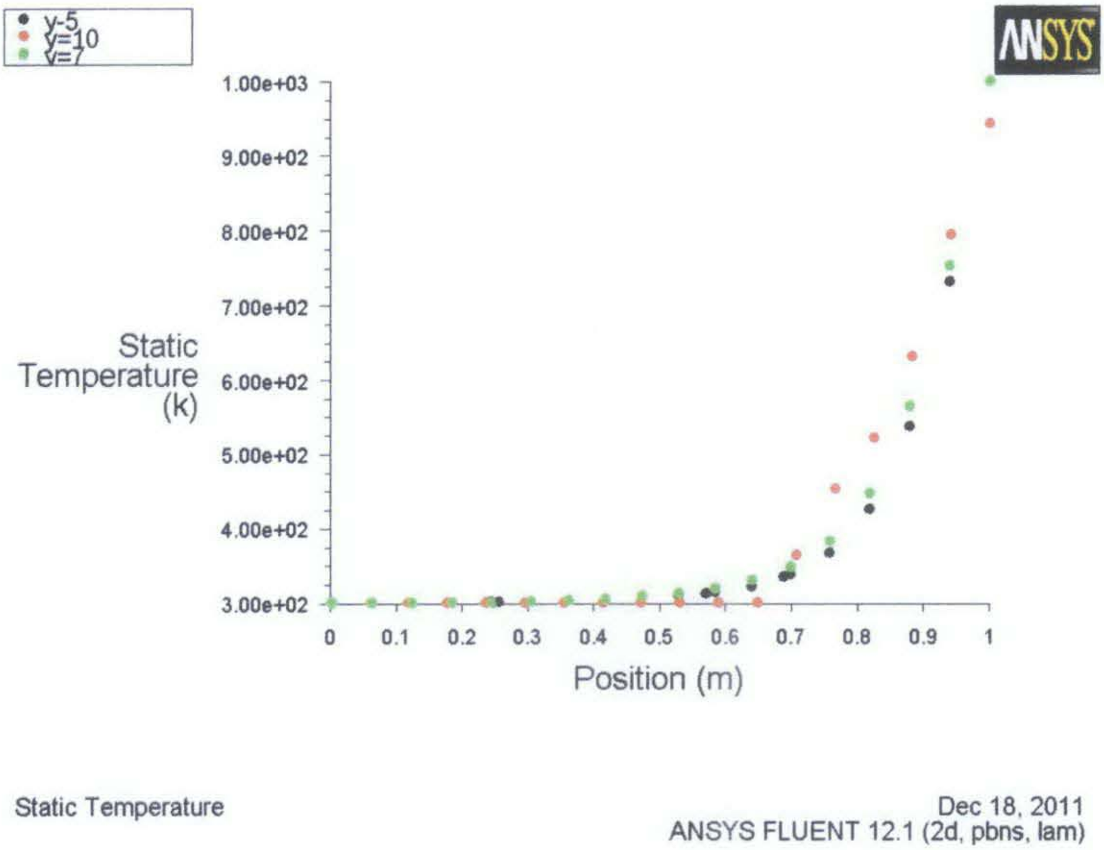

Figure 13: Temperature XY Plot for Case 2

From the velocity vector, it can be seen that the velocity of air is higher near the heated right plate, and that the velocity of air increases as it goes up the plate from the inlet to the outlet. The stream function shows the mass flow rate near the inlet to be high and concentrated at the heated plate, while as the flow goes toward the outlet it moves toward the adiabatic left plate as well, even as it increases near the heated right plate. The temperature profile predictably shows a higher temperature near the right plate, which is heated while near the left plate, the temperature of air remains at 300 K, which is the initial temperature of air when it enters the channel. The temperature profile also shows that near the heated plate, as the air approaches the outlet, the region of heated air near the plate gets larger, further from the plate.

The velocity XY plot shows the velocity of air in around the middle of the plates, and it can be observed that for x = 0.7 to x = 1, reverse flow does not occur whereas it occurs for x = 0 to x = 0.7, as a result of more cooling of the air near the left plate. The magnitude of the upward flow increases near the right plate while the magnitude of the downward flow decreases near the left wall as time increases. Finally at steady state, the upward flow was formed near the right wall and the downward flow near the left wall. Exactly at the right and left plates, the velocity is zero, which is true according to the no-slip boundary condition.

The temperature XY plot at the same position shows an increase of temperature as it approaches the right plate, while close to the left plate, the temperature of air is 300 K, the initial temperature of air at the inlet. However, it can be observed from the graph that at y = 7, the temperature of the air increases faster as it approaches the right plate compared to air at y = 5 and y = 10, even though at y = 10, the air is at the outlet. Theoretically, at y = 10, the temperature of air should increase faster as it approaches the heated right plate compared to both at y = 5 and y = 7. This may be due to the reversed flow that exists between the two plates.

4.2.3 Case 3: Left Plate Heated, Right Plate at Zero Heat Flux (Steady State)

In this case, the temperature of the left plate is set at 1000 K while the right plate is set at zero heat flux. The profiles obtained from the simulation are as shown below.

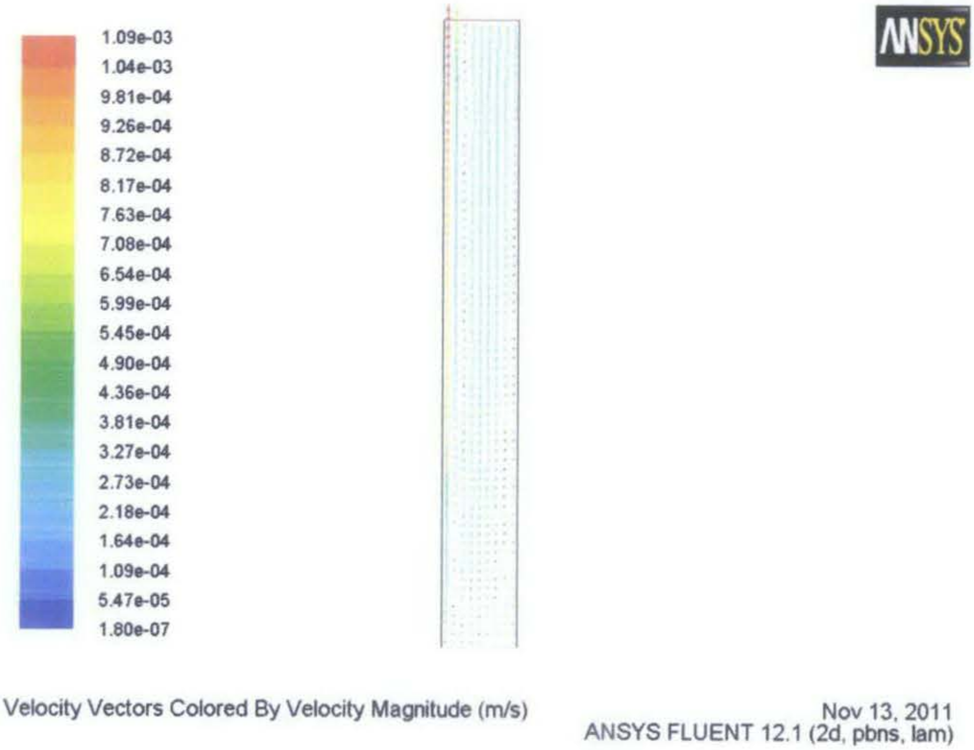

Figure 14: Velocity Vector for Case 3

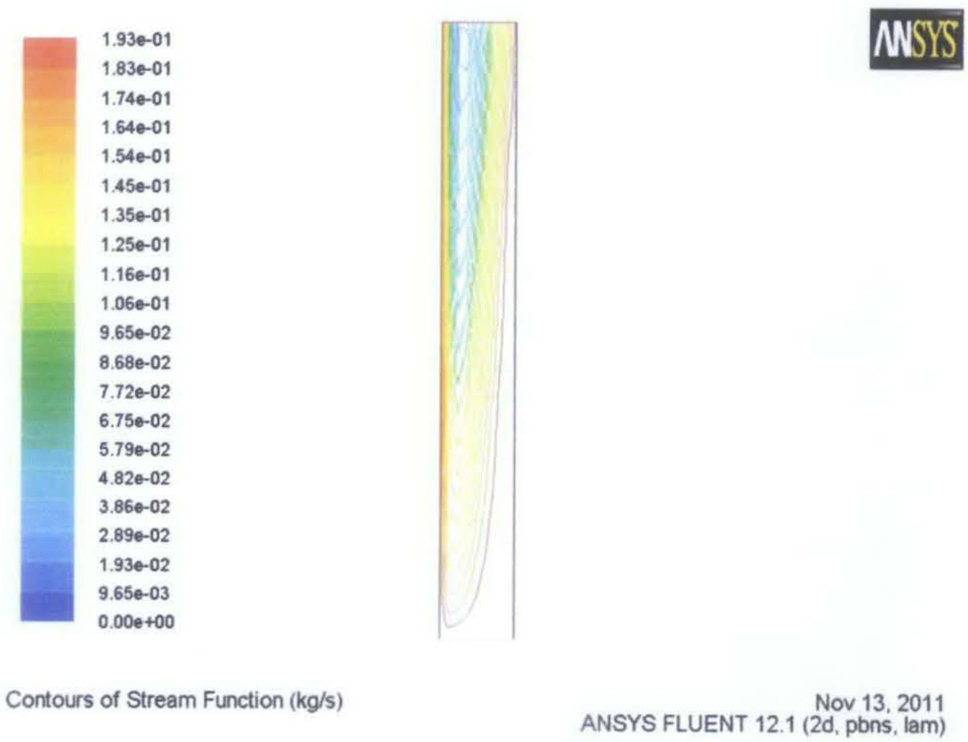

Figure 15: Stream Function for Case 3

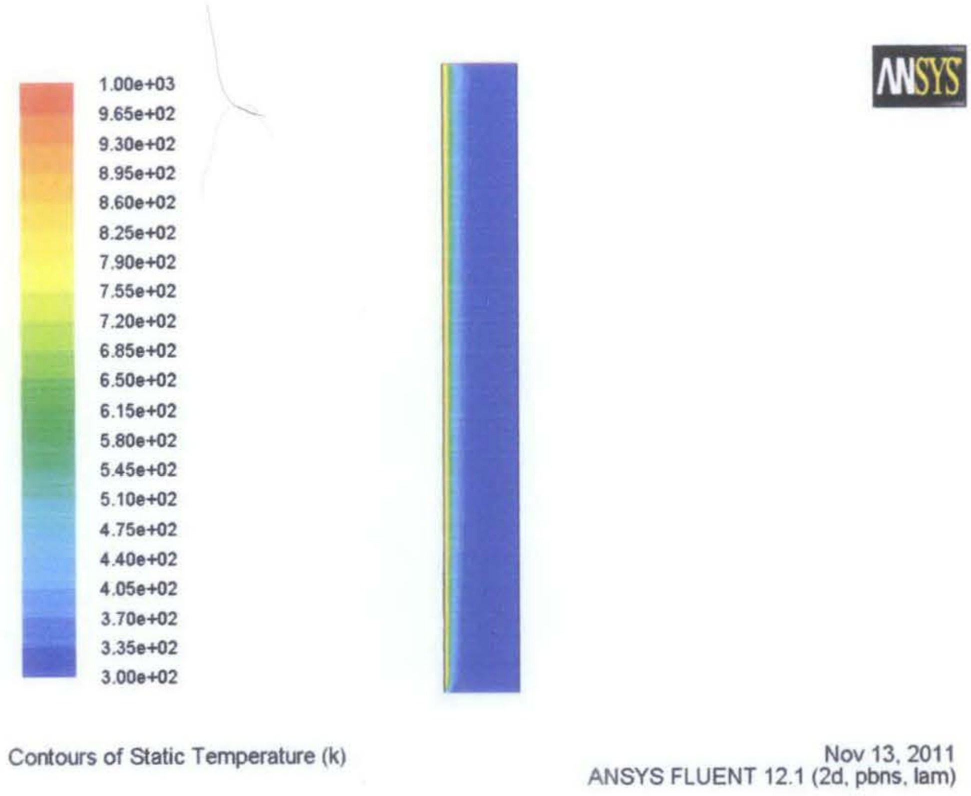

Figure 16: Temperature Profile for Case 3

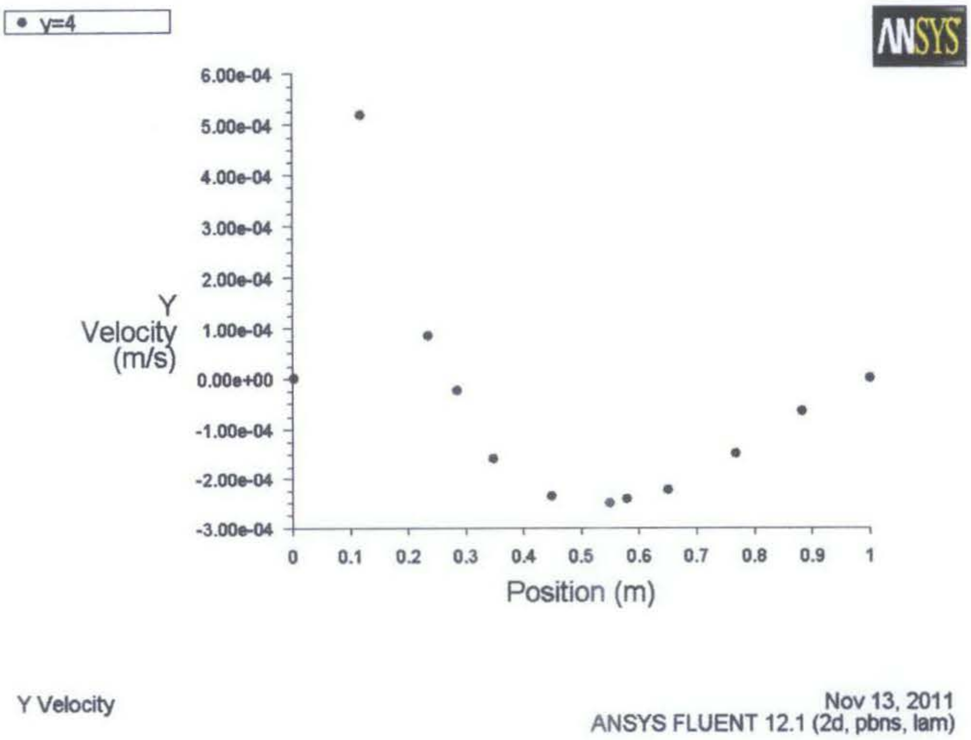

Figure 17: Velocity XY Plot for Case 3

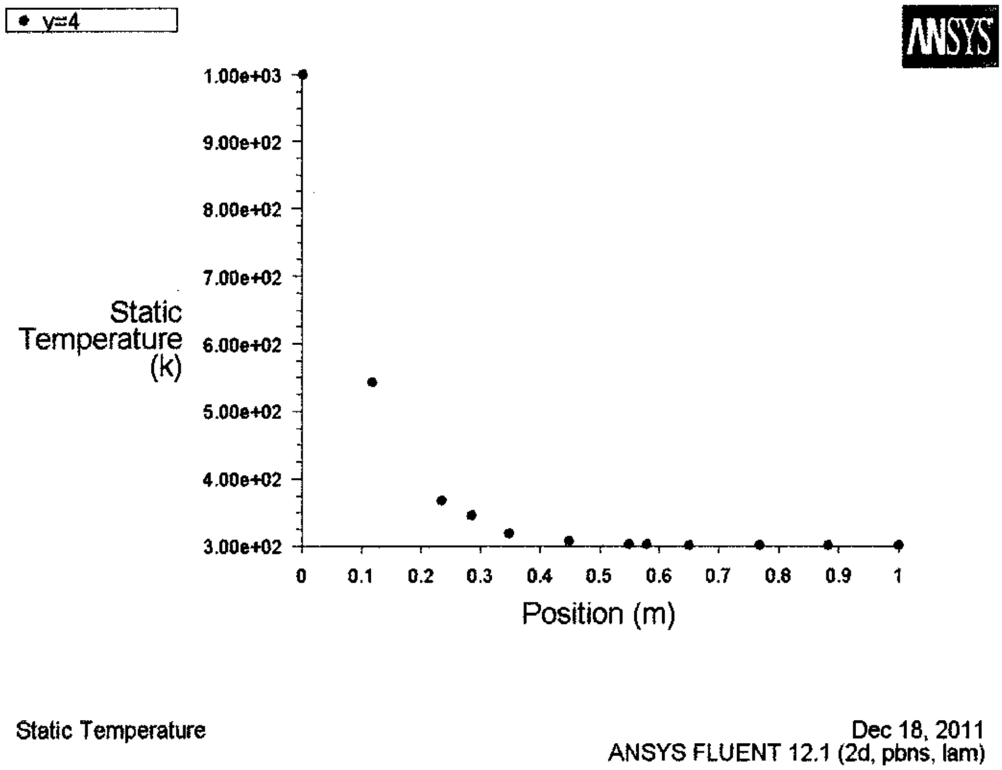

Figure 18: Temperature XY Plot for Case 3

From the velocity vector, it can be seen that the velocity of air is higher near the heated left plate, and that the velocity of air increases as it goes up the plate from the inlet to the outlet. The stream function shows the mass flow rate near the inlet to be high and concentrated at the heated plate, while as the flow goes toward the outlet it moves toward the adiabatic right plate as well, even as it increases near the heated left plate.

The temperature profile predictably shows a higher temperature near the left plate, which is heated while near the right plate, the temperature of air remains at 300 K, which is the initial temperature of air when it enters the channel. The temperature profile also shows that near the heated plate, as the air approaches the outlet, the region of heated air near the plate gets larger, further from the plate. However, it can be observed from the graph that at the outlet, the region of heated air is smaller than that neat the outlet. Theoretically, the outlet, the region of heated air should be larger compared to further inside the plates. This may be due to the reversed flow that exists between the two plates.

The velocity XY plot shows the velocity of air in around the middle of the plates, and it can be observed that for x = 0 to x = 0.3, reverse flow does not occur whereas it occurs for x = 0.3 to x = 1, as a result of more cooling of the air near the right plate. The magnitude of the upward flow increases near the left plate while the magnitude of the downward flow decreases near the right wall as time increases. Finally at steady state, the upward flow was formed near the left wall and the downward flow near the right wall. Exactly at the right and left plates, the velocity is zero, which is true according to the no-slip boundary condition. The temperature XY plot at the same position shows an increase of temperature as it approaches the left plate, while close to the right plate, the temperature of air is 300 K, the initial temperature of air at the inlet.

4.2.4 Case 4: Right Plate Heated, Left Plate at Zero Heat Flux (Transient)

In this case, the transient simulation was done with the temperature of the right plate set at 1000 K while the left plate set at zero heat flux. The profiles obtained from the simulation are as shown below.

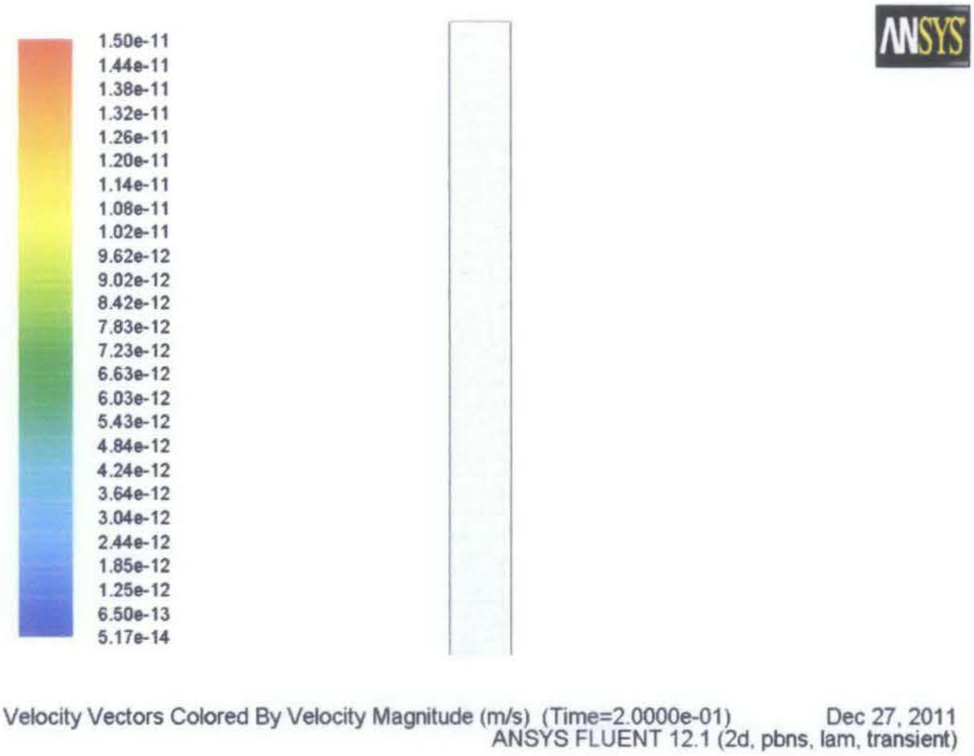

Figure 19: Velocity Vector at the Start of the Simulation

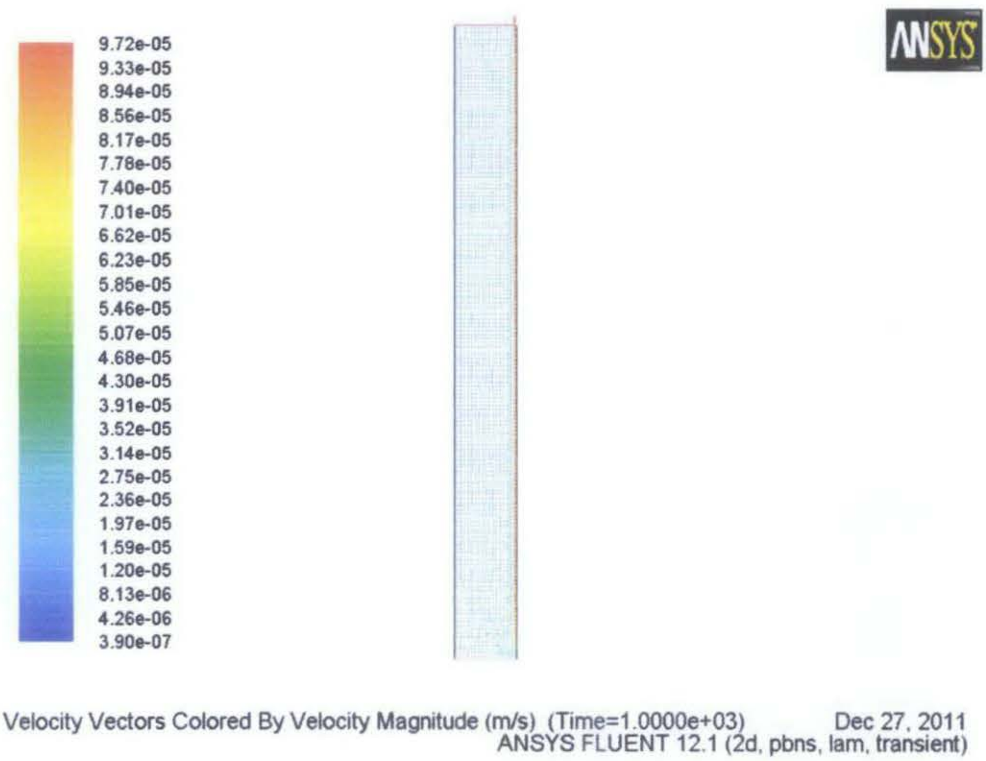

Figure 20: Velocity Vector After Time has Passed

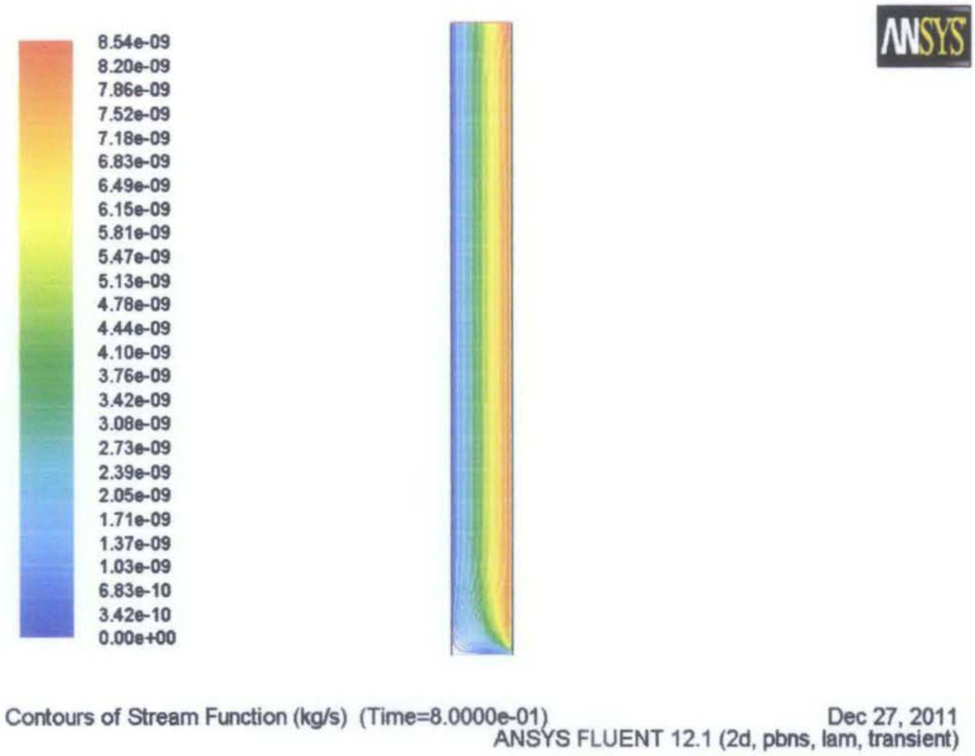

Figure 21: Stream Function at the Start of the Simulation

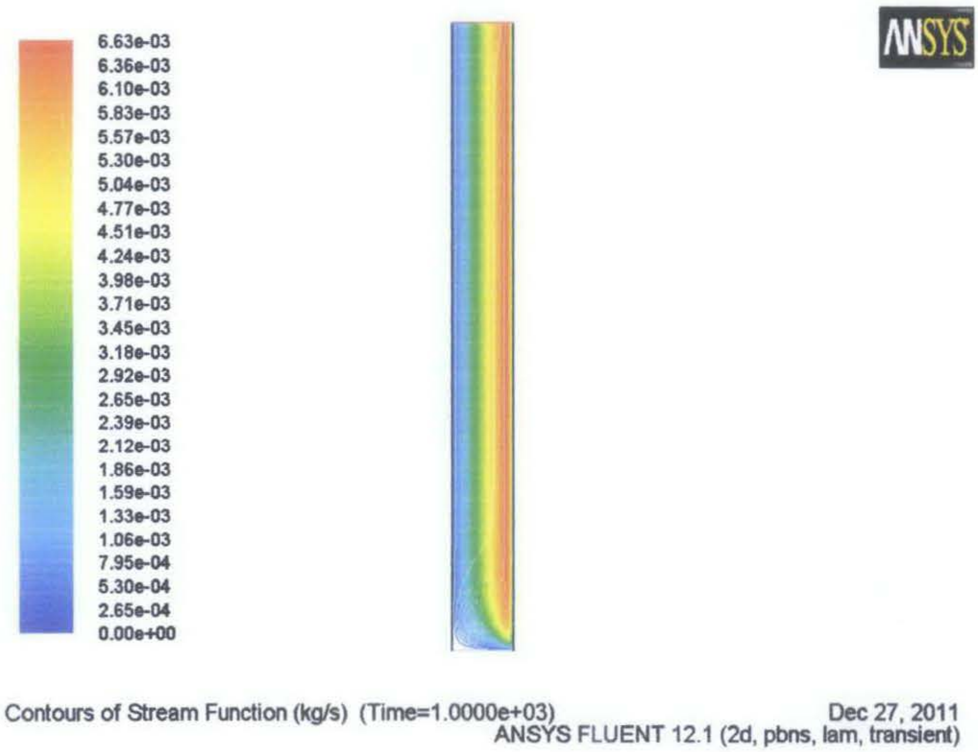

Figure 22: Stream Function After Time has Passed

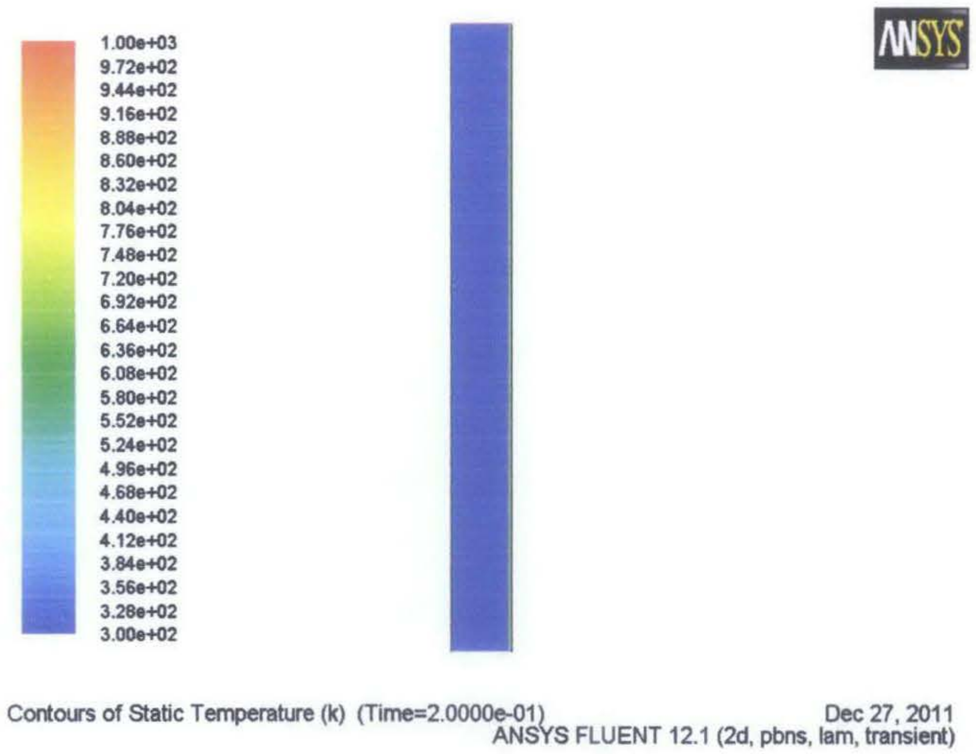

Figure 23: Temperature Profile at the Start of the Simulation

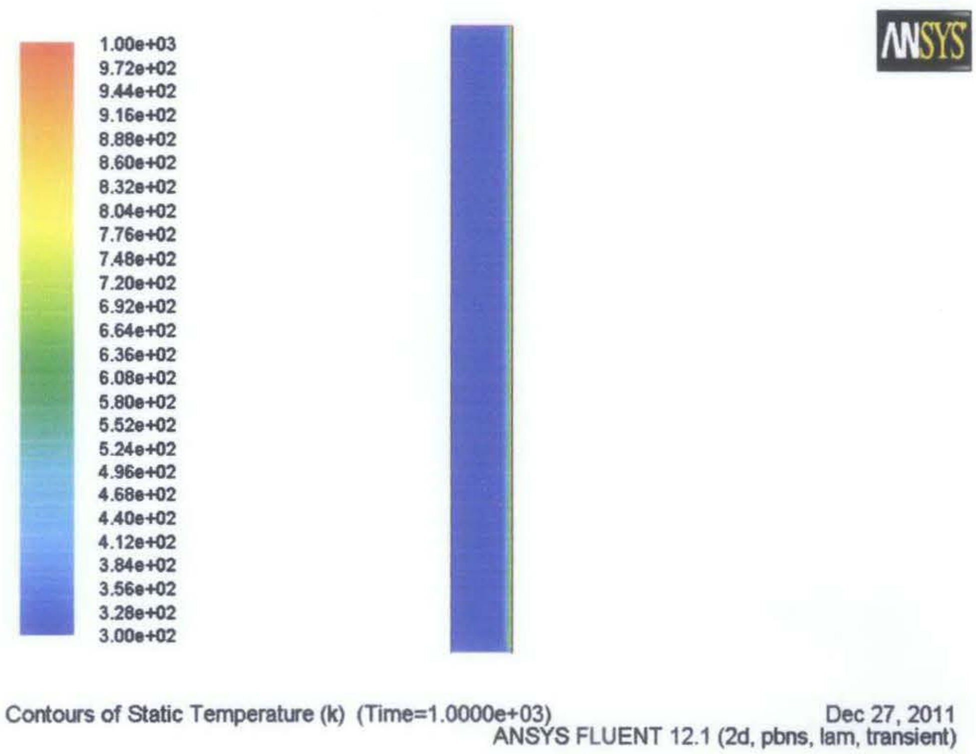

Figure 24: Temperature Profile After Time has Passed

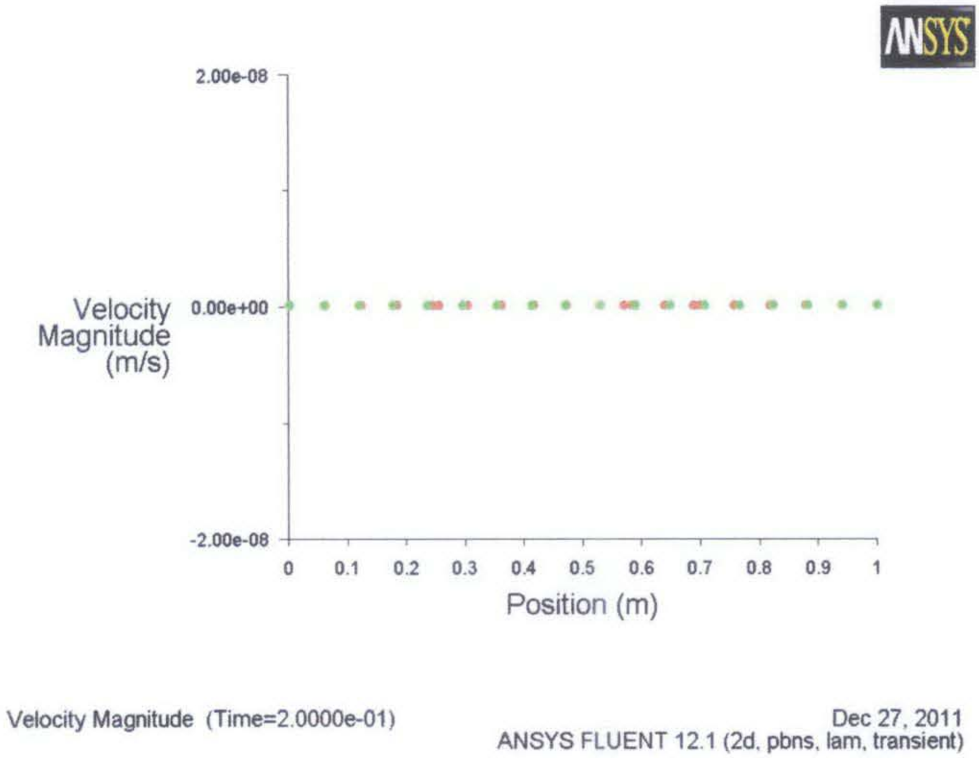

Figure 25: Velocity XY Plot at the Start of the Simulation

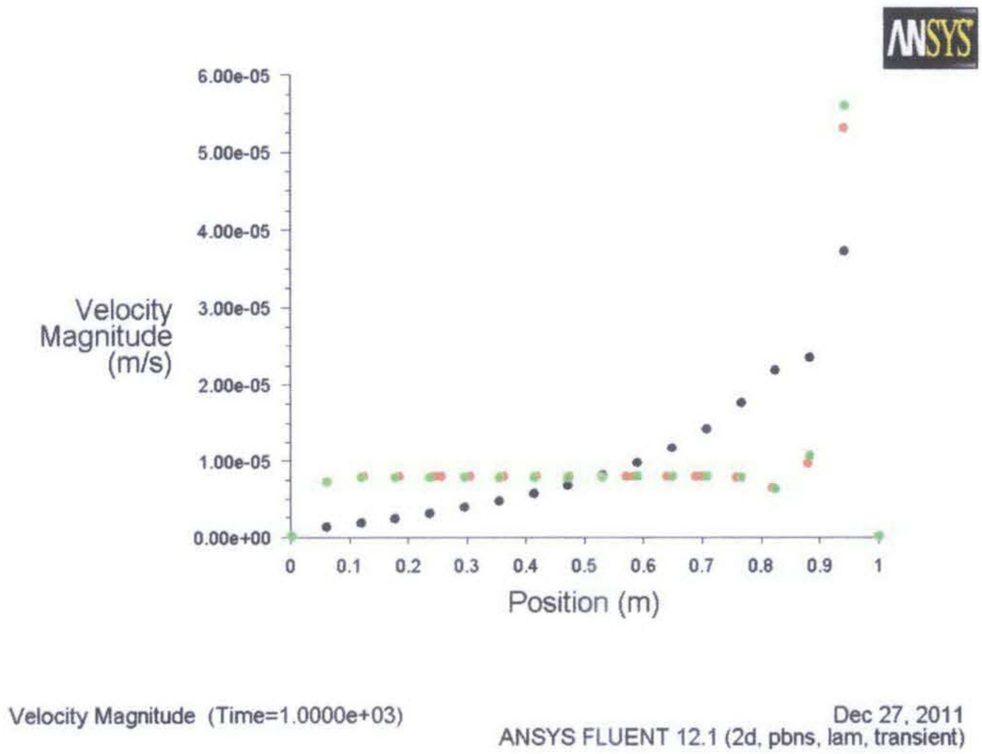

Figure 26: Velocity XY Plot After Time has Passed

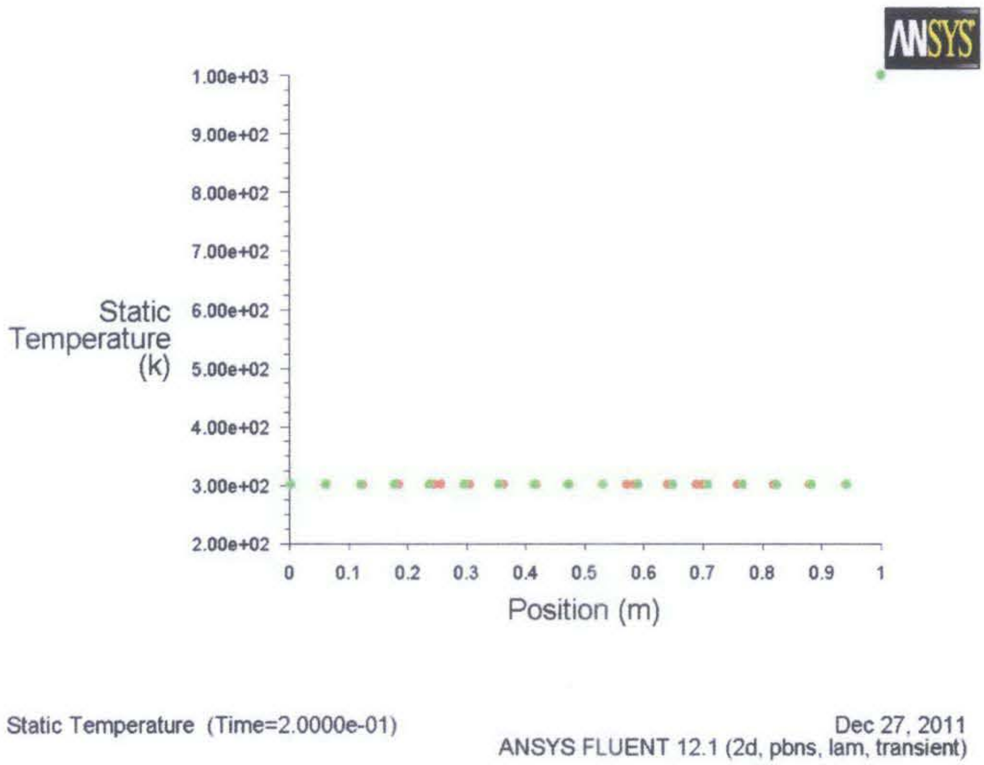

Figure 27: Temperature XY Plot at the Start of the Simulation

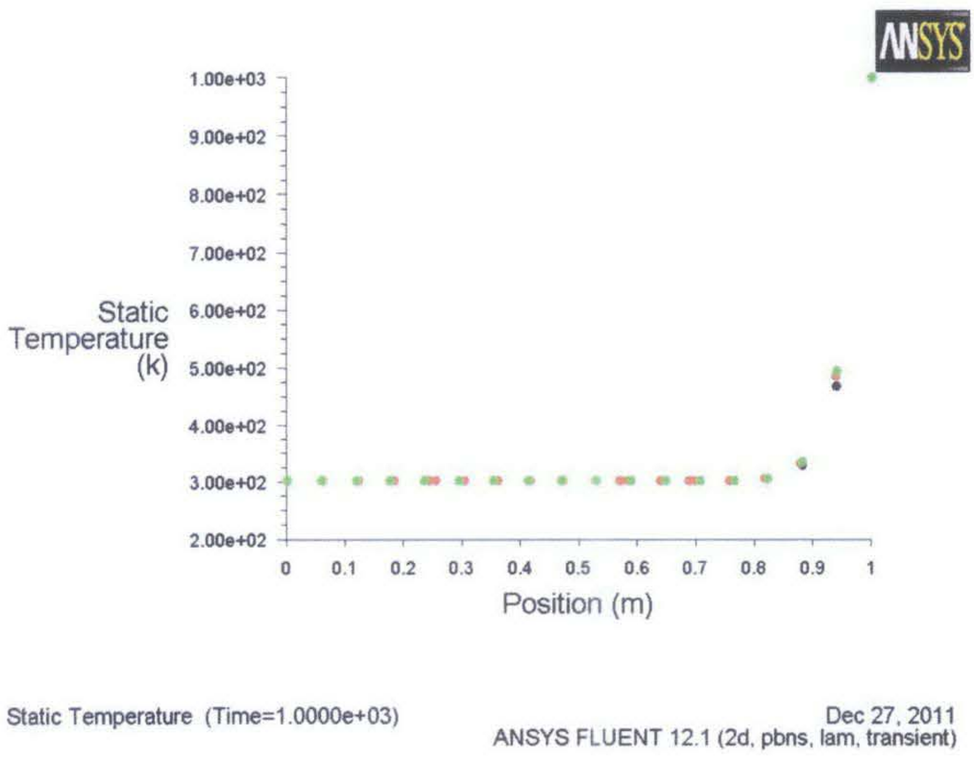

Figure 28: Temperature XY Plot After Time has Passed

The stream function contours show that the flow rate of air increases as the simulation goes on, and that the mass flow rate near the heated plate is always higher compared to the flow rate near the adiabatic plate. This agrees with the other simulations done in the steady state condition. The stream function shows that the mass flow rate of air increases with time. The temperature profile, on the other hand, does not show any change in the temperature of air, even as time passes.

The XY plots show the value of the parameters at different positions along the y-axis: (1) black at y=0.1 near the inlet, (2) red at y=5 in the middle and (3) green at y=9.9 near the outlet.

The velocity XY plot in the at the beginning shows the same velocity, at 0 m/s, and then, as time passes, the velocity of the air increases, with the air close to the heated right plate at approximately x = 0.95 relatively higher than the velocity of air away from the heated plate. Near the entrance of the plate, the velocity increases rapidly from the cold left wall to the heated right wall, while away from the inlet, the velocity of air that is not close to the heated plate is approximately uniform. This is because at the inlet the temperature gradient of air near the cold wall and the right wall is great, while this gradient decreases as air moves toward the outlet. There is a slight decrease in velocity near the heated right plate, due to the reverse flow. Even as the velocity of air increases with time, exactly at the plates, the velocity is zero, which adheres to the no-slip boundary condition.

Although from the temperature profile, no apparent change in the temperature can be observed even as time passes, looking at the temperature XY plot, it can be seen that there is a change in temperature that happens slowly. At the beginning of the simulation, the temperature is uniform across the plate at 300 K, which is the inlet temperature of air, except at x = 1, due to the right plate being heated at 1000 K. As time passes, the temperature of air close to the heated right plate at approximately $x = 0.95$ increases slightly, even though the temperatures further away from the heated plate remains the same. As even more time passes, the temperature of the air further from the heated plate starts to increase as well even as the temperature of air near the heated right plate increases, and this goes on until steady state is reached.

CHAPTER 5

CONCLUSION AND RECOMMENDATIONS

A mesh for the specified geometry of two vertical parallel plates with height of 10 m has been created for the simulation purposes of this study. Simulations have been done for four different situations using the model for both steady state and transient conditions, using air as the working fluid. From the profiles obtained from the CFD code, it can be observed that the velocity of air increases near a heated plate, but becomes exactly zero at the plate itself due to the no-slip boundary condition. In some cases, the velocity at certain parts between the plates is negative, due to the reversed flow that occurred between the plates. The mass flow rate is higher close to the heated plate, as does the temperature. For velocity, mass flow rate and temperature of the air, all increases as the air moves from the inlet to the outlet, but in certain cases near the outlet the parameter decreases slightly due to reverse flow. According to the results from the transient simulation, the velocity, mass flow rate and temperature increases with time.

Further study can be done for different working fluids, instead of air. In these simulations, radiation was not taken into account. The CFD code provides several radiation models, which can be used so that radiation can be taken into account in the simulation. Simulations can be done with radiation taken into account, and the most suitable radiation model for the simulation should be determined. Other operating conditions and geometries should be investigated to observe their effects on natural convection.

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APPENDIX I

u	Horizontal velocity component of the fluid in the x-direction
v	Vertical velocity component of the fluid in the y-direction
t	Time component
Pr	Prandtl number
T	Temperature of the fluid
T_o	Initial temperature
ρ	Density
k	Kinetic energy of turbulence or thermal conductivity
g	Acceleration due to gravity
c_p	Specific heat

APPENDIX II

The functions below are used in the equations from a journal described in Section 2.1.1.

$$F_1(a;b) = [(1 + a^2 b) erfc(a\sqrt{b}) - \frac{a\sqrt{b}}{\sqrt{\pi}} \exp(-a^2 b)]$$

$$F_2(a;b;c) = ab\, erfc\left(\frac{a}{2}\sqrt{\frac{b}{c}}\right) - 2\sqrt{\frac{bc}{\pi}} \exp\left(-\frac{a^2 b}{4c}\right)$$

$$F_3(a;b) = 2\sqrt{\frac{b}{\pi}} \exp\left(-\frac{a^2}{4b}\right) - a\, erfc\left(\frac{a}{2\sqrt{b}}\right)$$

$$erfc(\eta) = \frac{2}{\sqrt{\pi}} \int_{\eta}^{\infty} \exp(-x^2)\, dx$$

$$a_1 = (2n + 1)\sqrt{Pr} + 2m + y$$

$$a_2 = (2n + 1)\sqrt{Pr} + (2m + 2) - y$$

$$a_3 = 2n + 1 - y$$

$$a_4 = 2n + 1 + y$$

$$b_1 = \frac{a_1}{2\sqrt{t}}$$

$$b_2 = \frac{a_2}{2\sqrt{t}}$$

$$b_3 = \frac{a_3}{2\sqrt{t}}$$

$$b_4 = \frac{a_4}{2\sqrt{t}}$$