DOKUZ EYLÜL UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

HEAT TRANSFER AND FLUID FLOW IN A REFRIGERATOR THROUGH DOOR OPENING

by
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İZMİR

HEAT TRANSFER AND FLUID FLOW IN A REFRIGERATOR THROUGH DOOR OPENING

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by
Melih ODABAŞI

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İZMİR

M.Sc THESIS EXAMINATION RESULT FORM

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Melih ODABAŞI

HEAT TRANSFER AND FLUID FLOW IN A REFRIGERATOR THROUGH DOOR OPENING

ABSTRACT

In this study, natural convection within closed and open cavities are numerically investigated. For the closed cavity, side walls were kept at a constant temperature of T_{cold} and bottom wall was kept at a constant temperature of T_{hot} while the top wall was assumed adiabatic. The conservation equations of mass, momentum and energy were being solved by using boussinesq approaches. Analyses were carried out for the numerical values of Rayleigh number between 10^3 and 10^5. For open cavity analysis, the wall which is facing to the open surface was kept constant T_{hot} under assumption of the cavity's top and bottom surface areas are insulated. Calculations were done for the interval of 10^5 and 10^7 of Rayleigh number and dimensionless differences of the temperature, ε, values between 0.03 and 1.3 of the wall and liquid. The analyses made are compared with other literature studies done. As a conclusion of the research, it was observed that the heat transfer coefficient and Nusselt number are increasing with respect to increase in Raleigh number. Moreover, hydrodynamic boundary layer and thermal boundary layer are observed to get thinner as per increase of Rayleigh number.

As a conclusion for this study conducted, after detailed analysis and comparison with the literature, the work done is observed to produce similar results that are used as an indicative measure for reliability of the data and fridge is modeled in 2D and 3D. During 2D modeling, the investigation is grouped depending on the temperature that starts from 2, 4 and 6 Celsius and the number of the shelves 1, 2 or none. For the 3D model, 3 or 4 shelve situations, different shelves' height with same considerations including 2 and 4 Celsius temperature are investigated. Numerical analysis done shows that higher shelves tend to get warmer than the lower ones for the same duration exposed, while gain of energy tends to be lower inversely proportional to increased number of shelves and initial temperatures given.

Moreover, reducing the height of the shelves above while increasing the height of the following shelves were observed to decreases the energy gain.

Keywords: Refrigerated cavity, natural convection, computational fluid dynamics

KAPININ AÇIK KALDIGI SÜRECE BUZDOLABINA OLAN ISI TRANSFERİ VE AKIŞIN MODELLENMESİ

ÖZ

Çalışmada öncelikle kapalı bir hacim içerisindeki ve yalıtılmış duvarlar tarafından sınırlandırılmış açık bir hacimdeki doğal taşınım sayısal olarak incelenmiştir. Kapalı hacimdeki yan duvarlar sabit $T_{soğuk}$ sıcaklığında alt duvar sabit $T_{sıcak}$ sıcaklığında alınmış ve üst duvarın yalıtılmış olduğu varsayılmıştır. Kütle, momentum ve enerji korunum denklemleri boussinesq yaklaşımı kullanarak çözülmüştür. Analizler Rayleigh sayısının 10^3 ile 10^5 arasındaki değerleri için gerçekleştirilmiştir. Açık hacim için yapılan analizlerde ise açık yüzeye bakan duvarın sıcaklığı sabit T_h olarak alınmış ve hacmin üst ve alt yüzeylerinin yalıtılmış olduğu varsayılmıştır. Hesaplamalar Rayleigh sayısının 10^5 ile 10^7 arasındaki değerleri için ve duvar ile akışkan arasındaki boyutsuz sıcaklık farkının, ε, 0.03 ile 1.3 arasındaki değerleri için yapılmıştır.

Elde edilen sonuçlar literatürde yapılmış olan çalışmalar ile karşılaştırılmıştır. Yapılan çalışma sonucunda ısı transfer katsayısının ve Nusselt sayısının Rayleigh sayısının artışı ile arttığı görülmüştür. Hidrodinamik sınır tabaka ve ısıl sınır tabakanın ise Rayleigh sayısının artışı ile inceldiği görülmüştür. Yapmış olduğumuz çalışmalar ile literatürdeki çalışmaların örtüştüğü kanısına varıldıktan sonra buzdolabı iki boyutlu ve üç boyutlu olarak modellenmiştir. İki boyutlu yapılan modellemelerde sistemin buzdolabı başlangıç sıcaklığının 2,4 ve 6 derece olması durumları ile rafsız, 1 raflı ve 2 raflı durum incelenmiştir. 3D modelde ise 3 ve 4 raflı durumlar, farklı raf yükseklikleri için yine aynı durumlar ve, 2 ve 4 derece başlangıç sıcaklığı durumları incelenmiştir. Yapılan nümerik analizler sonucunda soğutucunun en hızlı şekilde en üst rafların ısındığı, soğutucunun raf sayılarının ve başlangıç sıcaklıkları arttıkça enerji kazanımlarının azaldığı görülmüştür. Ayrıca en üst rafların yüksekliklerinin azaltılıp alt rafların yüksekliklerinin arttırılmasının enerji kazanımını azalttığı görülmüştür.

Anahtar Kelimeler: Soğutulmuş hacim, doğal taşınım, hesaplamalı akışkanlar dinamiği

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CHAPTER ONE

INTRODUCTION

1.1 General Introduction

Many existing and ongoing research that are about refrigerators are built up based on reducing the energy consumption. In the early 90s, chillers that are ozone type gas are widely used but with the time due to increased importance of reduced energy consumption and which is also expected to be the case in near future, usage of new type of cooler gas became more common.

As common practice from past to nowadays, fridges are expected to achieve two fundamental functionality which are to keep the nutrient in acceptable conditions for a long time while achieving low energy consumption. Both of these functionalities are determined by temperature and humidity. The reason for the temperature and humidity are not constant is due to air that is more humid enters to the fridge while opening and closing the fridge door. When the humid air interacts with the cold surface which is lower than the freezing temperature firstly it condenses than it will cause snow by solidifying on evaporator surfaces.

Frosting can be observed at first where the air interacts with the evaporator. Frosting that occurs on evaporator surface is an unwanted condition which results in reduced performance and energy efficiency. The main reason can be explained as the increased thermal resistance caused by frosted surface. The second reason is it causes decreased distance between the evaporator fins which increases the pressure drop that results in increased power need for the same cooling effect for the compressor and fan. Another reason is the more often repetitive usage of defrost operation itself to prevent frosting in the fridges meaning increased energy consumption for the same resistance value operating during defrost operation.

1.2 Literature Review

1.2.1 Natural Convection Studies on Closed and Open Cavities

Refrigerators can be seen as application of closed (when the doors are closed) or open cavities (when the doors are open). Today, many study on open and closed cavities done and maintained. These studies have been made can be examined under two main headings.

1.2.1.1 Literature Review of Natural Convection Studies on Closed Cavity

Basak, Roy & Balakrishnan (2006) had a numerical study to research stable and regular natural convection which was created with the bottom wall heated regularly or irregularly while the upper wall was insulated in a quadratic cavity. To solve mass, momentum and energy equations, bi-quadric oblong elements and finite elements methods were used. Even though average Nusselt numbers show low heat speed for irregular heating, irregular heating at the bottom wall showed more heat transfer for the all Rayleigh numbers. The critical Rayleigh numbers were obtained for the heat transfer cases of which transmission was dominant, also, for the cases of which transfer was dominant, power law correlation between the Rayleigh number and the average nusselt number was defined.

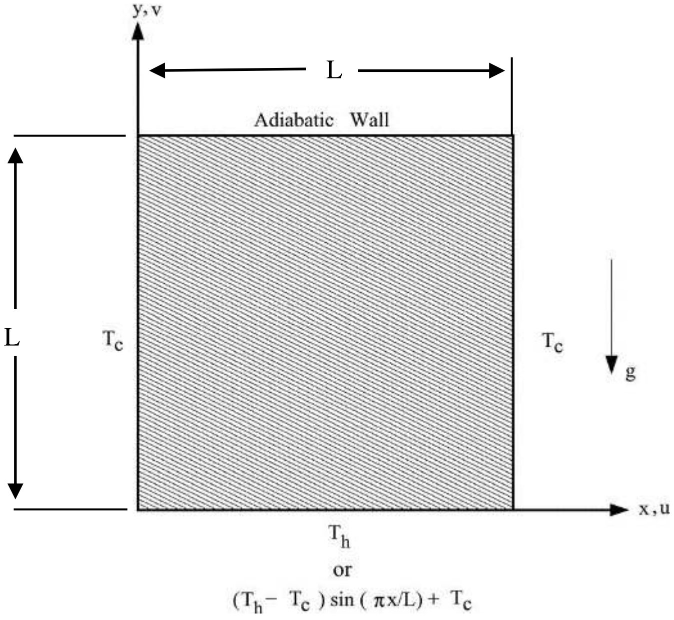

Figure 1.1 Schematic of the physical model of Basak (2006)

1.2.1.2 Literature Review of Natural Convection Studies on Open Cavity

Polat & Bilgen (2002) examined the natural convection which exists with stable and regular flow in a numerical way. The calculations have been done by considering the thin wall by the open surface has been exposed to stable heat flow, the other sides as insulated, and the open surface as in touch with the liquid in a stable heat and pressure. Mass- momentum and energy equations were solved by stable functions, boussineq approach and considering an approximate boundary condition. The lines of isotherm and flow was produced, the transfers of mass and heat were calculated for the estimations between the Rayleigh numbers 10^3 and 10^{10}, aspect ratio A=H/L, 1 and 0.125. The results showed that Rayleigh number of heat transfer and flow were changed by length-width coefficient and inclination angle.

Polat & Bilgen (2003) numerically examined the heat transfer on open shallow cavities. The thin wall by the open side was exposed to the heat flow, the other sides were considered as insulated and the calculations were done by considering the open surface was connected to the liquid under stable heat and pressure. The heat transfer with transmission and convection was analysis with solving mass, momentum and energy bounds numerically. In the study parameters were chosen as Rayleigh numbers between 10^3 and 10^{10} , the conductivity coefficient between 1 and 60, aspect ratio between A/H=L, 1 to 0.125 , the inclination angle between 0° and 45° from the horizontal surface. Isotherms and stream lines were produced, mass and heat transfer were calculated. It was observed that flow was increased by A, L, u, Ra, yet decreased by k_r. On the other hand; the heat transfer increased by L, Ra, yet decreased by k.

Nouanegue, Müftüoğlu & Bilgen (2008) studied on natural transfer, transmission and heating situations of the volume of which had an open side and exposed to a regular heating. The preservation equations were solved with the finite volume method. The parameters used on the study were taken as Rayleigh numbers between 10^9 and 10^{12}, aspect ratio, between A=L/H, 0.4 and 1, the thickness of the wall, between l/H 0.02 and 0.08, transmission parameter, between k_r 1 and 50, the surface emission coefficient between, ε, 0 and 1, and Prandtl number was taken as 0.7 apodictically. In the study, it was seen that surface heating was effective on the flow and temperature areas. The heating flow by the heating increased with the emission coefficient, yet because of this increase, heating flow caused by natural transfer and transmission decreased. On the other hand, transfer and heating for the conductivity coefficient with the nusselt numbers increased as the function of the surface emission rate. The combined nusselt number and the Rayleigh numbers high on volume flow rate increased as the function of the surface emission. Transfer and heating nusselt numbers were in the shape of the function of decreasing the wall conductivity and the increasing aspect ratio.

Juárez, Hinojosa, Xamán & Tello (2011) calculated flow within an open volume considering fluid features connected to natural transfer and heating. To solve the general duration, momentum and energy equations, finite elements method was used. The calculations were done for the Rayleigh numbers values between Ra, 10^4 and 10^7. The heat difference between the wall and the liquid was altered between (ΔT) 10 and 500 K and the temperature was made non-dimensional to generalize the observed tendencies. The results showed that the thickness of the border layer by the warm wall increased by the increase of the ε value from 0.03 to 1.6. The average nusselt number showed an increase for ε=0.03 807% and for ε=1.6 1105% by the increase of the Rayleigh number from 10^4 to 10^7. At the same time, the average nusselt number showed a decrease for Ra=10^4 39% and for Ra=10^7 18% with the increase of ε value from 0.003 to 1.6.

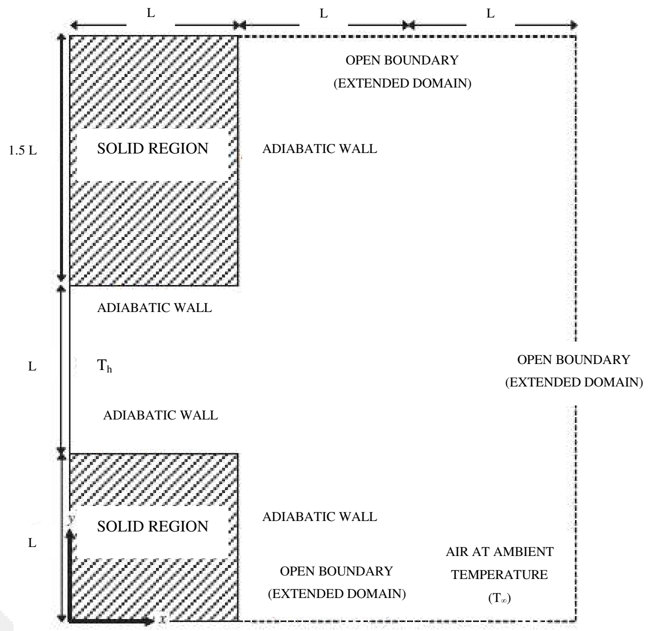

Figure 1.2 Schematic of the physical model of Juárez (2011)

1.2.2 Heat and Mass Transfer into Refrigerator and Flow Modeling Studies

The studies on refrigerators include two situations and these are the situations when the refrigerators are either open or closed.

1.2.2.1 The Studies Carried Out for the Situation of Closed Door

Laguerre, Amara, Charrier-Mojtabi, Lartigue & Flick (2008) used a refrigerator in which heat was carried by the natural convection of the heat on his experimental studies. This transfer is taking place between cold wall of the refrigerator and other walls. In the experiment, air speed calculation was shown with particle vision anemometer and an air circulation was observed from the cold wall in the refrigerator and for the other walls it was observed as from the bottom to the top. The maximum air speed (0.2 m/s) was observed in the area close under the wall, and the least air speed (<0.04 m/s) was observed in the center and above of it. Besides, the air profile of the boundary layer in the empty refrigerator was studied. The effect of the heat of the cold surface and the cold wall space on the air speed, and it was understood that the heat of the cold surface had more effect on the air speed than the space.

For the aim of seeing the effect of the obstacles on the air speed, the refrigerator was stuffed with four empty nodular blocks. Result of the air speed of the full refrigerator was compared to the empty refrigerator results, and it was observed that air speed was less almost everywhere inside of the refrigerator and the blocks homogenized the speed.

Laguerre, Benamara, Remy & Flick (2009) experimented the natural convection with heat, humidity transfer and air circulation in a rectangular space including line cylinder. On the symmetry platform, temperature, speed and humidity areas were moisturized, and calculated for the non- moisturized areas, and these results were used for comparing them to the CFD simulation. In the empty volume circular air flow was seen. It seems that moistening the cavity is cause to increase air velocity at the bottom layer. It was seen radiation effect was important close to the cold and warm walls.

Laguerre, Benamara & Flick (2010) have a clearer idea then before about vaporization and condensation situations by natural convection. At first, a refrigerator stuffed with moistened cylinder was simulated. At the same time, CFD simulation air flow was provided with the heat transfer and the mass transfer (vaporization and condensation). The numerical results were compared to the experiment. Even though simulation didn't include the experimental results, the areas where vaporization and condensation occurred were estimated correctly. After that, air conditioning in a real refrigerator was carried out. The same results were observed on the real refrigerator as the previous model one (condensation for the products close to the evaporator, vaporization for the products close to the door and the top areas).

1.2.2.2 The Studies Carried Out for the Situation of Open Door

Knackstedt, Newell & Clausing (1995) studied on the heat transfer parameters inside of the refrigerator and its surfaces in the open door situations. The flow visualization and the air change studies were carried on to provide the whole picture of the transfer and heat gaining. As a result, it was found out for 3 shelves situation, the surface heat flux was 0.551 W/m^2 (on the bottom), and 5.208 W/m^2 (on the top-front plaques). These parameters were directly affected by lines of the cooling area flow as a function of the amount of present shelves. The simple temperature calculations were done before the door was open, when the door was open and after. It was seen that the air temperature deterioration after the door was closed and the exponential deterioration curve were almost in a perfect harmony in 50 seconds.

Inan, Newell & Eğrican (2000) had an experiment on temperature division is inside of the refrigerator when the door was opened and the flow moves of the air. With the results, they had 3 different flow regime categories as without the shelves, with 1 shelf and with 2 shelves. These categories:

I. The first temporary period; this period starts from opening the door and lasts until 7-10 seconds. In this time, the cold air leaks under the refrigerator as the clean air enter from the upper area. In this period, both speeds of the entrance and exit are very fast.

II. The transitional period; this period includes the time from 7th-10th seconds to 20th second. In this period, both the heat and the speed approach their own half-stable situations in a linear way.

III. The half stable period; it is considered as steady if the temperatures of the walls stay as the same after 20^{th} second. In this time, the speed and the temperature values are stable just as the temperature of the wall.

1.3 Objectives, Motivation and Methodology

In this project, the goal is to investigate and analyses numerically distributed effect of increased moisture and temperature over humidity and temperature inside a working fridge during different periods that the refrigerator's door is kept open.

Project targets to conduct a literature search. Despite the existence of numerical analysis of opening a refrigerator's door, its coverage can be defined as shallow. This literature work is to be investigated in detail to create numerical models, open cavity and closed cavity models and refrigerator model, thermal and hydrodynamic analysis are to be done. Numerical modeling and analysis will be done by using ANSYS FLUENT computable fluid dynamics (CFD) package program while modeling the refrigerator and its working environment on a basic level. The energy exchange and several parameters' affect such as different initial temperatures and the number of shelves will be numerically analyzed.

CHAPTER TWO
NUMERICAL STUDY

2.1 Problem Definition of Natural Convection for Closed Cavity

There is a rectangular cavity in this model. Top Wall of the cavity is insulated, side walls were keeping at a constant temperature of T_c and bottom wall was keeping at a constant temperature of T_h Temperature differences, $\Delta T=T_h-T_c$, causes a vertical change in air density then motion and heat transfer with natural convection.

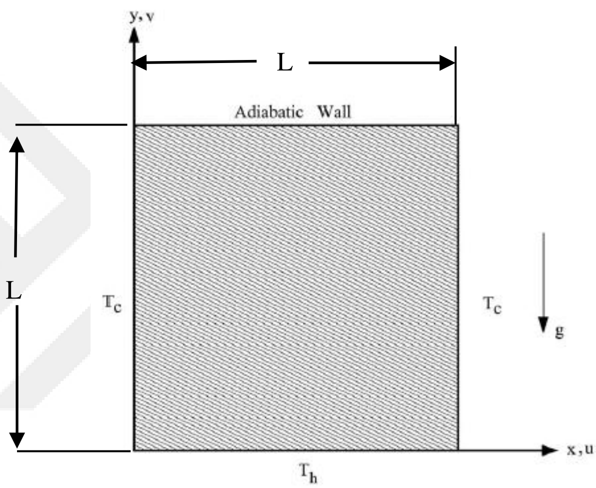

Figure 2.1 Schematic of the physical model of closed cavity

The flow model is based on the assumptions that system in steady state and the fluid is Newtonian and that the properties are constant with the exception of the density in the body force term of the momentum equation. The Boussinesq approximation is invoked for the fluid properties to relate density changes to temperature changes. From above the considerations, viscous chosen as a laminar, pressure solution method chosen as a PRESTO momentum and energy solution methods chosen as a power law method at CFD (Ansys Fluent) program.

The governing equations of the natural convection flow problem can be written as;

Continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{2.1}$$

x momentum equation:

$$\rho\left(u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = -\frac{\partial p}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \tag{2.2}$$

y momentum equation:

$$\rho\left(u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right) = -\frac{\partial p}{\partial y} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) + \rho g_y \tag{2.3}$$

Energy equation:

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \propto\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{2.4}$$

The Non-dimensional equations:

$$Ra = \frac{g\beta(T_h - T_c)L^3 Pr}{v^2} \tag{2.5}$$

$$Pr = \frac{v}{\propto} \tag{2.6}$$

2.2 Problem Definition of Natural Convection for Open Cavity

Rectangular open cavity was limited by insulated walls. The vertical walls was keeping at a constant temperature T_h, and the air, excepted as a Newtonian fluid and ideal gas, was in contact with extended boundary layers at a constant temperature T_∞. Temperature differences, $\Delta T=T_h-T_\infty$, causes a vertical change in air density then motion and heat transfer with natural convection.

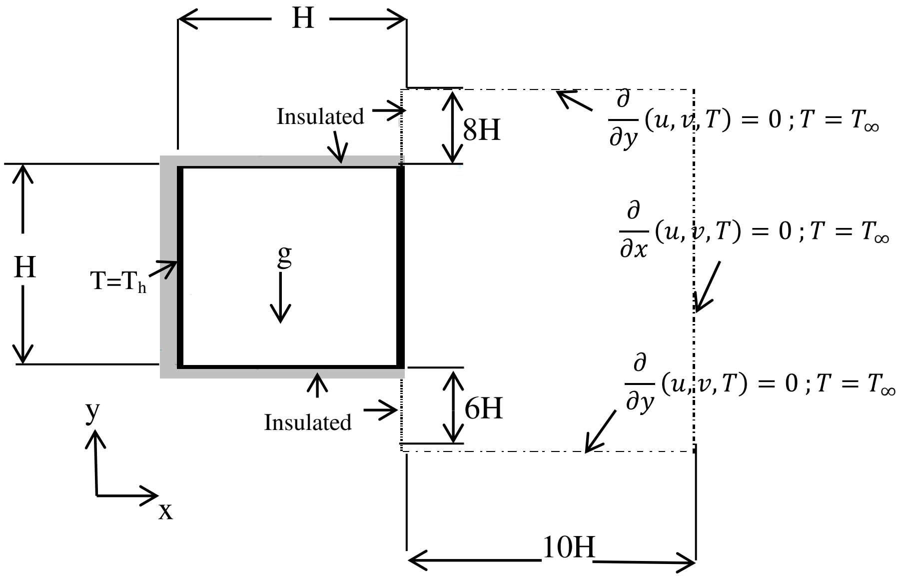

Figure 2.2 Schematic of the physical model of open cavity

The fluid flow is assumed to be laminar and at steady state. The large open space outside the cavity is referred to as the extended domain and was assumed to be bounded by hypothetical boundaries which are far enough from the cavity so as to neglect its influence on the fluid motion inside the cavity. From above the considerations, viscous chosen as a laminar, spatial discretization for pressure chosen as a PRESTO for momentum and energy chosen as a power law method at CFD (Ansys Fluent) program.

The governing equations of the natural convection flow problem can be written as;

Continuity equation:

$$\frac{\partial u}{\partial x}+\frac{\partial v}{\partial y}=0 \qquad (2.7)$$

x-momentum equation:

$$\rho\left(u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=-\frac{\partial p}{\partial x}+\mu\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \qquad (2.8)$$

y- momentum equation:

$$\rho\left(u\frac{\partial v}{\partial x}+v\frac{\partial v}{\partial y}\right)=-\frac{\partial p}{\partial y}+\mu\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right)+\rho g_y \qquad (2.9)$$

Energy equation:

$$u\frac{\partial T}{\partial x}+v\frac{\partial T}{\partial y}=\propto\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}\right) \qquad (2.10)$$

The Non-dimensional equations:

$$Ra=\frac{g\beta(T_h-T_\infty)L^3 Pr}{v^2} \qquad (2.11)$$

$$\epsilon=\frac{T_h-T_\infty}{T_\infty} \qquad (2.12)$$

$$Pr=\frac{v}{\propto} \qquad (2.13)$$

$$Nu=\frac{h.l}{k} \qquad (2.14)$$

2.3 Problem Definition for the 2D Model

2.3.1 Mathematical Model

Rectangular open cavity was limited by insulated walls. There are two shelves at the cavity and the analysis is run for one self, two shelves and no shelf situations. The vertical walls was keeping at a constant temperature T_c, and the air, excepted as a Newtonien fluid and ideal gas, was in contact with extended boundary layers at a constant temperature T_∞. Temperature differences, $\Delta T = T_\infty - T_c$, causes a vertical change in air density then motion and heat transfer with natural convection.

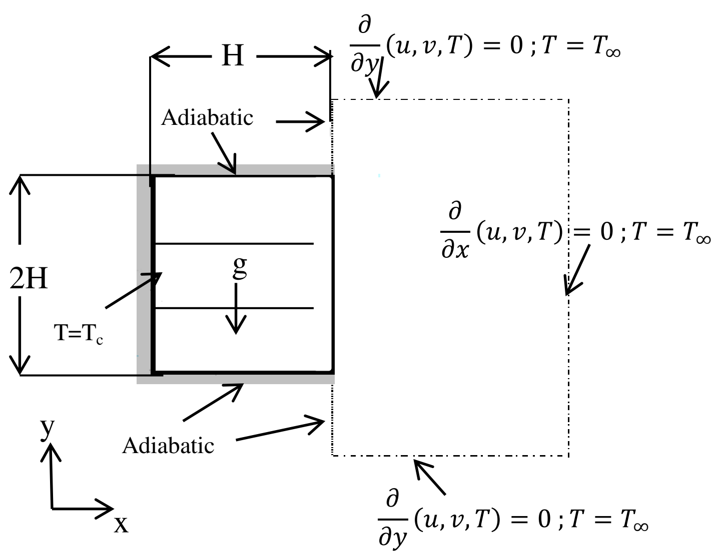

Figure 2.3 Schematic of the 2D model

The fluid flow is assumed to be laminar. The large open space outside the cavity is referred to as the extended domain and was assumed to be bounded by hypothetical boundaries which are far enough from the cavity so as to neglect its influence on the fluid motion inside the cavity. From above the considerations, viscous chosen as a laminar, pressure solution method chosen as a PRESTO, momentum and energy solution methods chosen as a power law method at CFD (Ansys Fluent) program. Also the time step chosen 0.5 sec during the analyze.

The governing equations of the natural convection flow problem can be written as;

Continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{2.15}$$

x- Momentum equation:

$$\rho\left(\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y}\right) = -\frac{\partial p}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2}\right) \tag{2.16}$$

y- Momentum equation:

$$\rho\left(\frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y}\right) = -\frac{\partial p}{\partial y} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2}\right) + \rho g_y \tag{2.17}$$

Energy equation:

$$\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \propto \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2}\right) \tag{2.18}$$

2.3.2 Mesh Structure

Mesh generated at ansys program (Figure 2.4). Quadratic method used on meshing. Also bias used at near of shelves and surfaces for calculate better changes in the temperature and air velocity. Choices we have made in accordance with the minimum mesh size generated as $4.64xe^{-0.03}$ and the maximum mesh size generated as 0.9271.

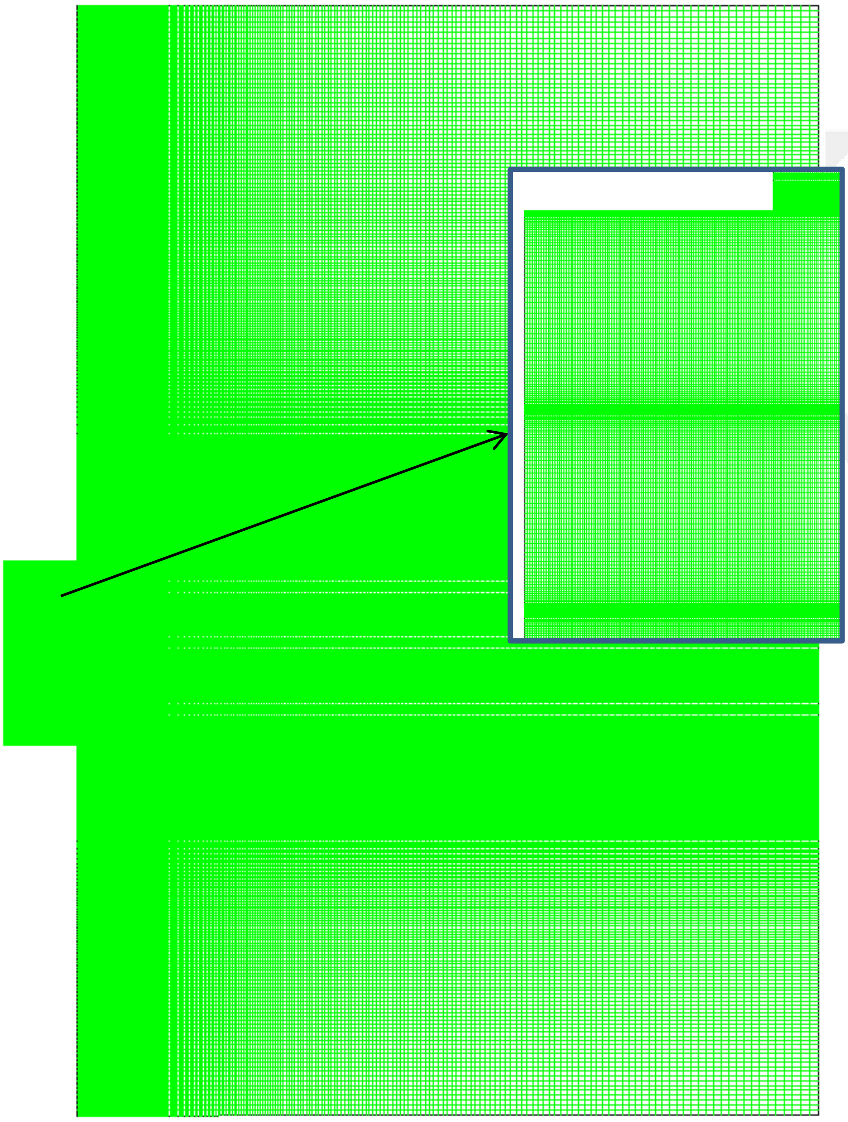

Figure 2.4 Mesh structure of the 2D model for 2 shelf situation

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2.4 Problem Definition for the 3D Model

2.4.1 Mathematical Model

Refrigerator (freezer at the bottom), are have different shelf heights and numbers, was modeled at the open doors situation in the rectangular environment (Figure 2.5). It is assumed as an insulated environment walls, air Newtonian fluid and has been recognized as an ideal gas. The temperature differences between environment temperature and refrigerator volume temperature causes a vertical change in air density then motion and heat transfer with natural convection.

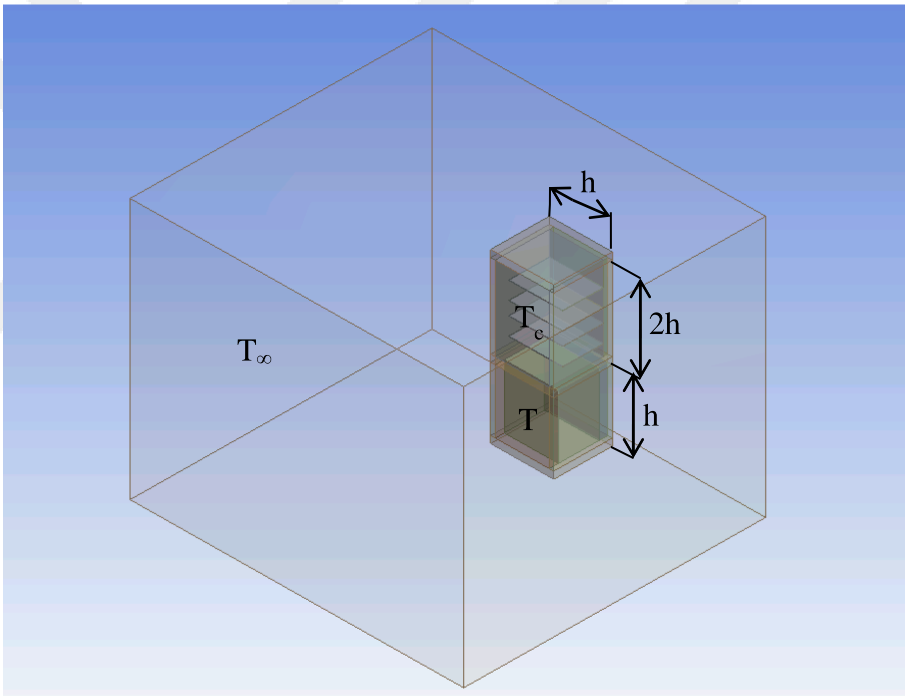


Figure 2.5 Schematic of the 3D model

The fluid flow is assumed to be laminar. The material and the shelf of the refrigerator has been recognized that allow heat transfer. Initially starting temperature of the refrigerator is taken as the system average. From above the considerations, viscous chosen as a laminar, pressure solution method chosen as a PRESTO momentum and energy solution methods chosen as a power law method at CFD (Ansys Fluent) program.

The governing equations of the natural convection flow problem can be written as;

Continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \tag{2.19}$$

x-momentum equation:

$$\rho\left(\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} + w\frac{\partial u}{\partial z}\right) = -\frac{\partial p}{\partial x} + \mu\left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} + \frac{\partial^2 u}{\partial z^2}\right) \tag{2.20}$$

y- momentum equation:

$$\rho\left(\frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} + w\frac{\partial v}{\partial z}\right) = -\frac{\partial p}{\partial y} + \mu\left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} + \frac{\partial^2 v}{\partial z^2}\right) + \rho g_y \tag{2.21}$$

z- momentum equation:

$$\rho\left(\frac{\partial w}{\partial t} + u\frac{\partial w}{\partial x} + v\frac{\partial w}{\partial y} + w\frac{\partial w}{\partial z}\right) = -\frac{\partial p}{\partial z} + \mu\left(\frac{\partial^2 w}{\partial x^2} + \frac{\partial^2 w}{\partial y^2} + \frac{\partial^2 w}{\partial z^2}\right) \tag{2.22}$$

Energy equation:

$$\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} + w\frac{\partial T}{\partial z} = \propto\left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2}\right) \tag{2.23}$$

2.4.2 Mesh Structure

Mesh generated at ansys program (Figure 2.6). Quadratic method used on meshing. Also bias used at near of shelves and surfaces for calculate better changes in the temperature and air velocity.

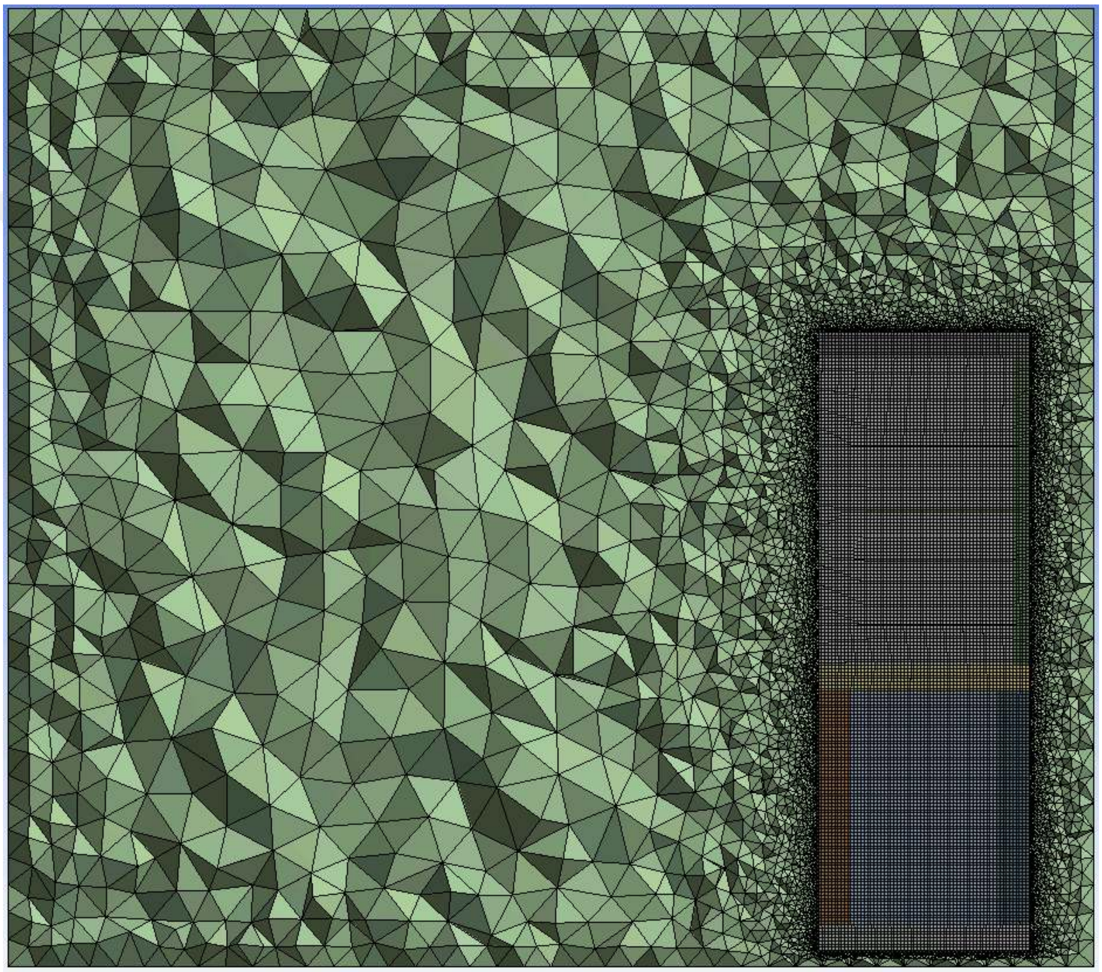


Figure 2.6 Sectional view of mesh structure of the 3D model for 4 shelf situation

CHAPTER THREE

RESULTS AND DISCUSSION

3.1 Compare of Math Model of Closed Cavity

The results which we obtain while solving the equations from our closed cavity model with CFD (Ansys Fluent) program is compared with the Basak et al., 2006's results which they obtain while solving the same model and boundary conditions with their own code (Figure 3.1–3.4).

At the analysis the bottom wall of the closed cavity was kept at a constant temperature T_h, which is warmer than the side walls, the side walls were kept at a constant temperature T_c and the top wall were accepted as adiabatic.

During the comparison of these obtained results it was seem that the air, which is heated by the bottom wall, is moving upward by the middle of the bottom wall and air, which is chilled by the side walls, is moving downward by the side walls. Also as a result of this situation, the middle of the cavity to be the axis of the symmetry, two vortex flow in the system can see clearly from both of graphs.

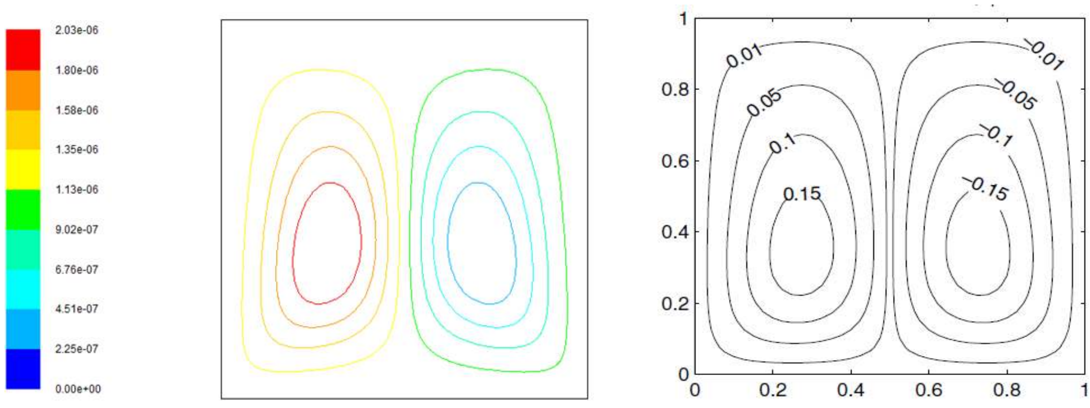


Figure 3.1 Present work (left side) and Basak's (2006) (right side) stream lines for Ra= 10^3 Pr=0.7

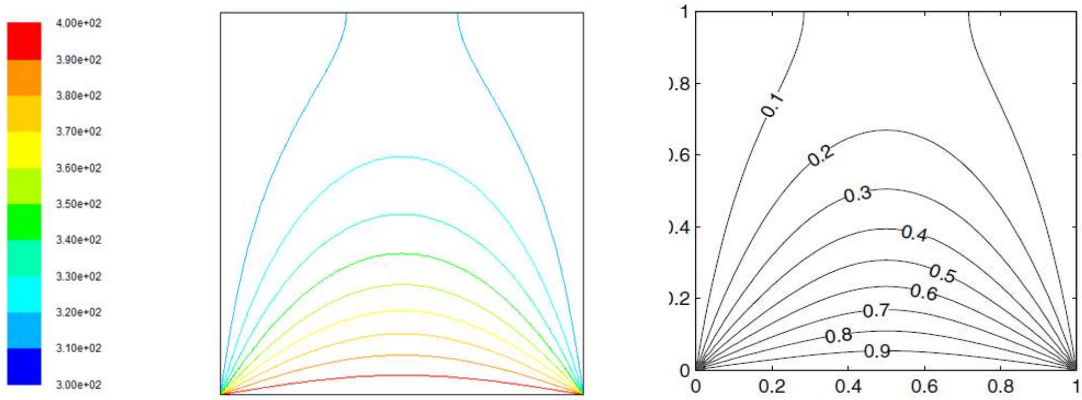


Figure 3.2 Present work (left side) and Basak's (2006) (right side) temperature contours for Ra= 10^3 Pr=0.7

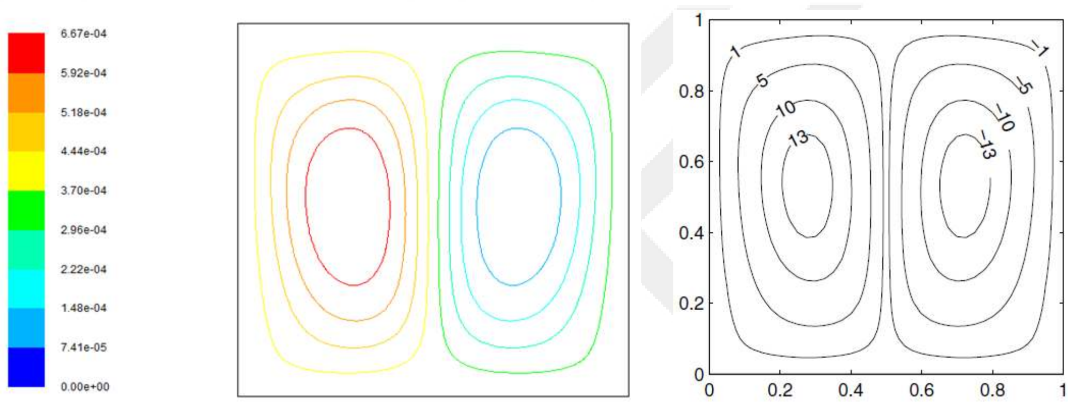


Figure 3.3 Present work (left side) and Basak's (2006) (right side) stream lines for Ra= 10^5 Pr=0.7

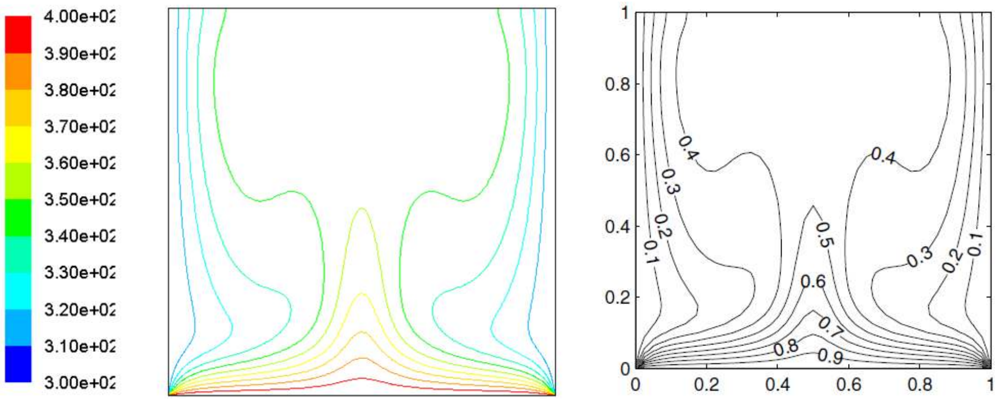


Figure 3.4 Present work (left side) and Basak's (2006) (right side) temperature contours for Ra= 10^5 Pr=0.7

3.2 Compare of Math Model of Open Cavity

The results which we obtain while solving the equations from our second model (same heat and dimension values with Juárez et al., 2011) with CFD (Ansys Fluent) program is compared with the Juárez et al., 2011's results which they obtain while solving the same model and boundary conditions with their own code (Figure 3.5-3.10).

Analyzes were carried out for different Rayleigh number and dimensionless temperature differences. At the analysis, the wall front of the open area was kept at the constant temperature of T_h, which is warmer than open area temperature, and top and bottom walls were accepted as adiabatic. During the analysis, the air, which is heated by the warm wall, is moved towards upward and outside of the cavity. The ambient air, which is colder than the heated one, flows toward downstream and into the volume above.

Figure 3.5 Present work and Juárez (2011) stream lines for $Ra=10^5$ $\epsilon=1$

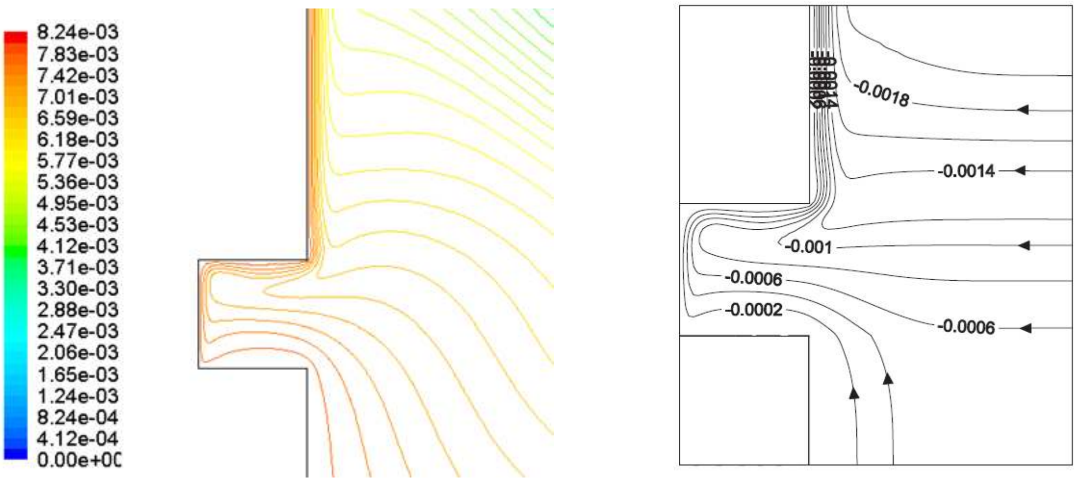


Figure 3.6 Present work and Juárez (2011) stream lines for Ra=10^6 ϵ=1

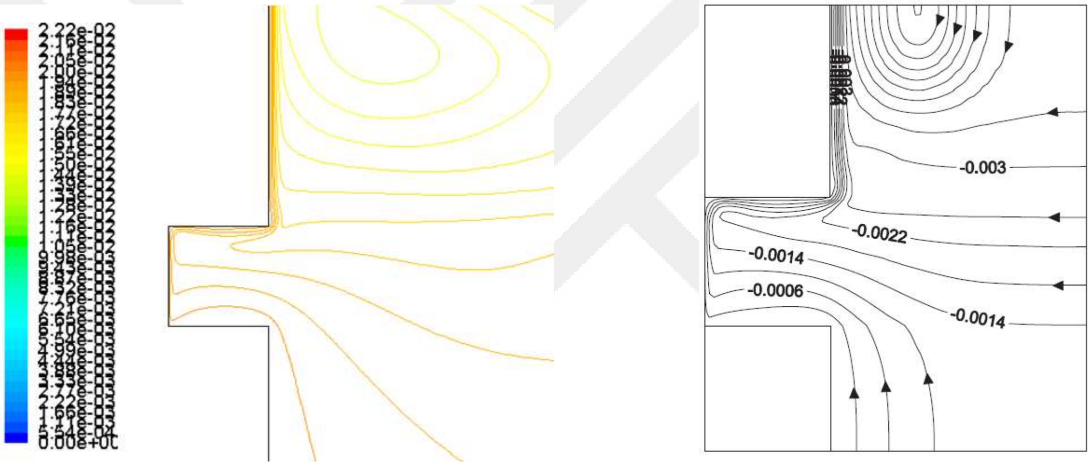


Figure 3.7 Present work and Juárez (2011) stream lines for Ra=10^7 ϵ=1

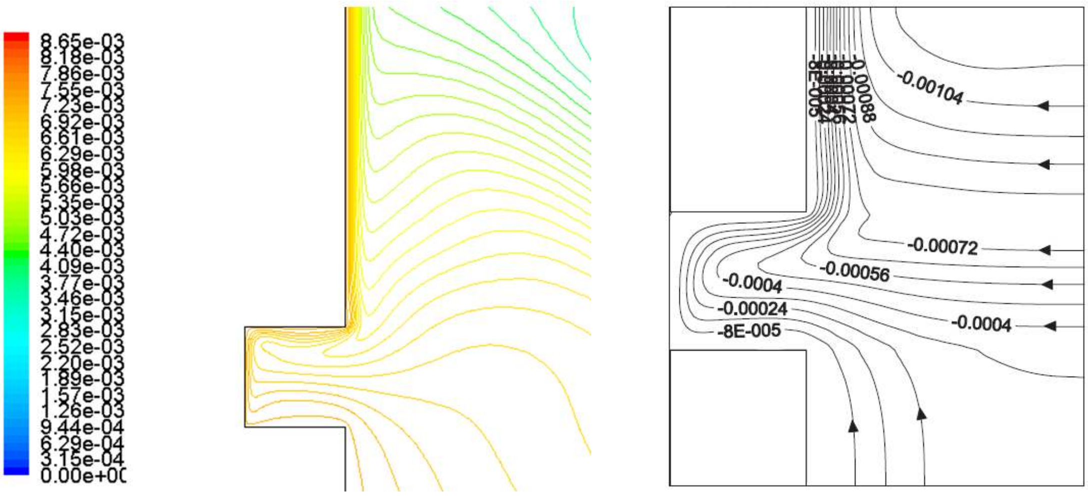


Figure 3.8 Present work and Juárez (2011) stream lines for Ra=10^5 ϵ=1.3

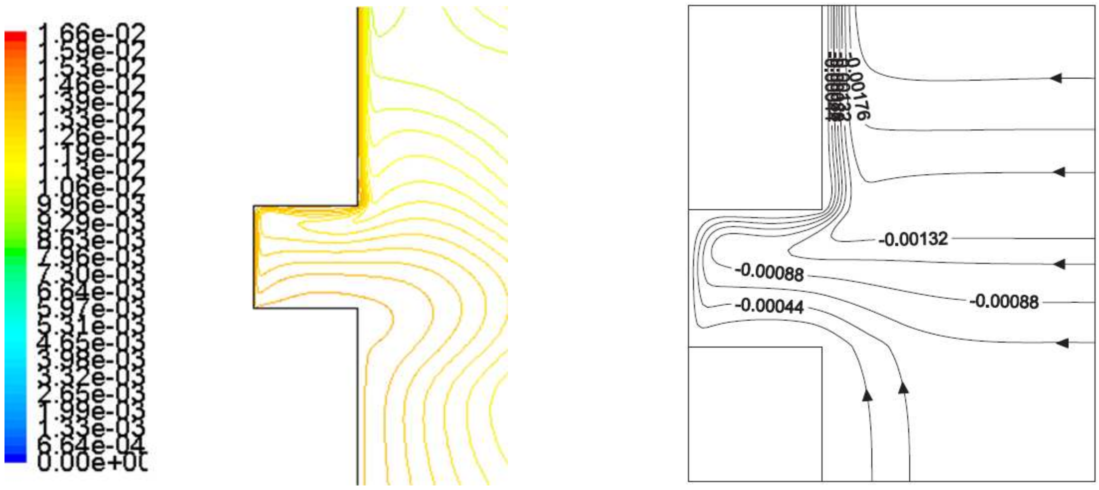


Figure 3.9 Present work and Juárez (2011) stream lines for $Ra=10^6$ $\epsilon=1.3$

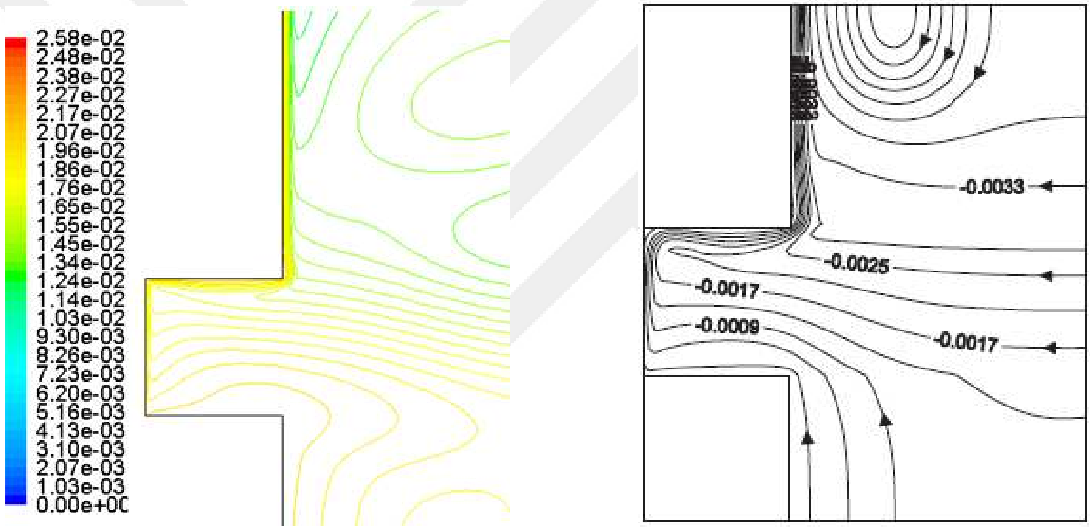


Figure 3.10 Present work and Juárez (2011) stream lines for $Ra=10^7$ $\epsilon=1.3$

As a result of studies made, it seems that the heat transfer coefficient, h, and the nusselt number is increasing with the increase of Rayleigh number (Table 3.1). At the same time it seems that with the increase of Rayleigh number the hydrodynamic boundary layer and thermal boundary layer is become thin.

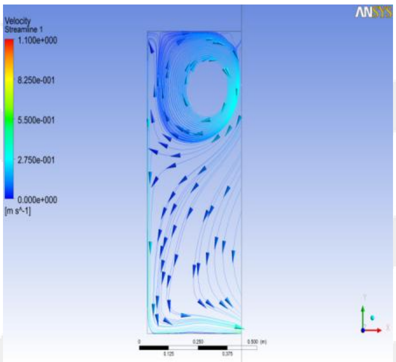
Table 3.1 Nusselt numbers at the ϵ=1.3

Ra	ϵ	Nu (Present work)	Nu (and Juárez (2011))
10^5	1.3	7.21	6.07
10^6	1.3	13.3	12.71
10^7	1.3	26.1	24.7

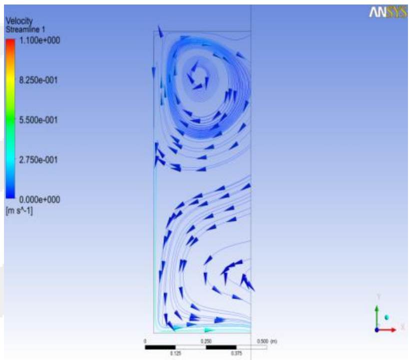
Results what we achieve are so similar to Juárez et al., (2011),'s and Basak et al., (2006),'s results. It was concluded that the small differences at the stream lines in second model are because of the soling with another programs (c++, Fortrun), the distance of the outside walls and mesh structure.

3.3 Results and Discussion of 2D Model

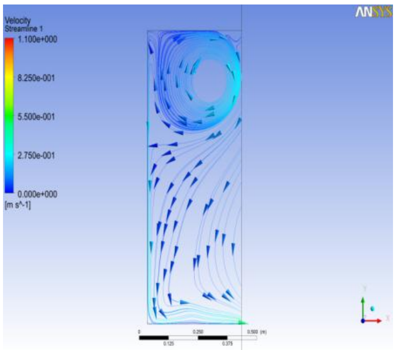
Two dimensional analysis of the refrigerator, the refrigerator door facing surface modeled as 2^0C and 6^0C. At the analysis the internal temperature of the refrigerator is taken the same as the surface temperature. Near of the different operating temperature the analysis done for no shelf, one shelf (at the middle of the refrigerator, y=50 cm), and two shelves (y=40 cm and y=70 cm) situations. The temperature of the room, where is the refrigerator stays, has been fixed at 24^0C during the analysis.



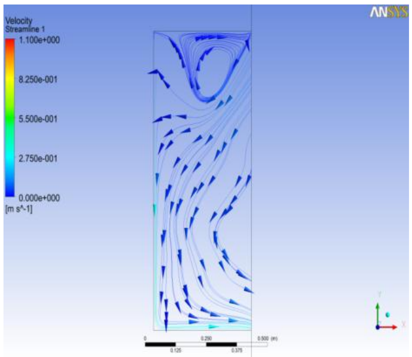
2^0C, 5 sec



2^0C, 120 sec

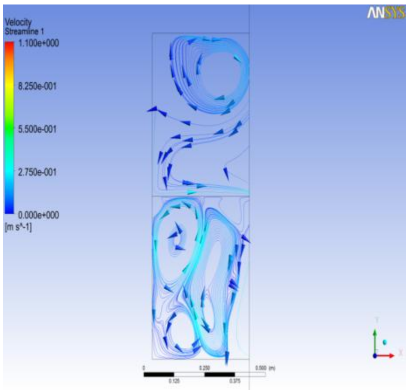


6^0C, 5 sec

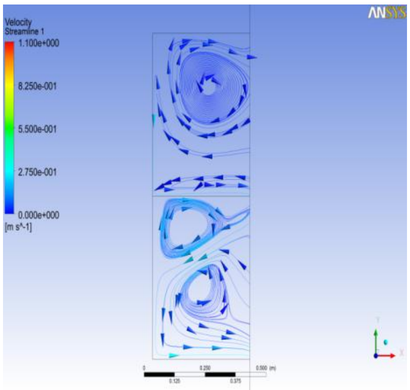


6^0C, 120 sec

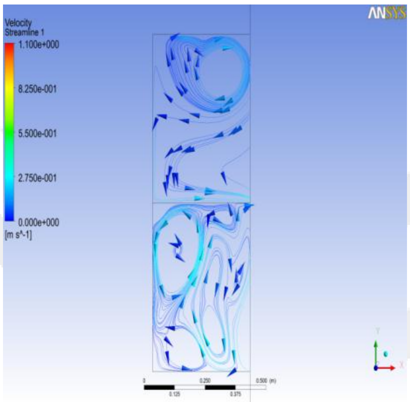
Figure 3.11 Velocity contours at the 5th and 120th seconds at the refrigerator with no shelf at 2^0C and 6^0C starting temperature.



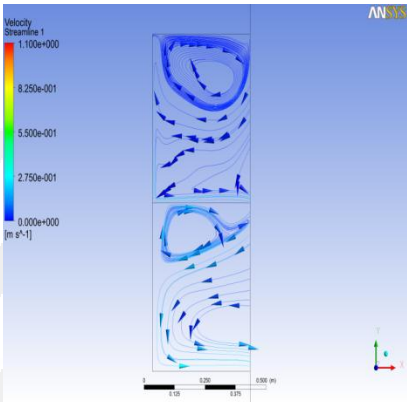
2^{0}C, 5 sec



2^{0}C, 120 sec

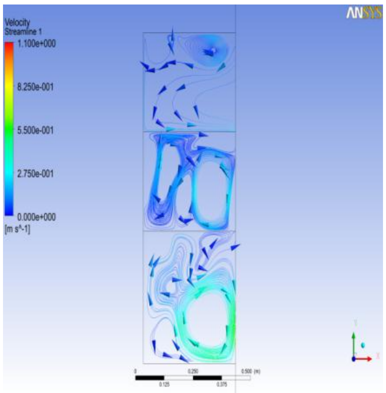


6^{0}C, 5 sec

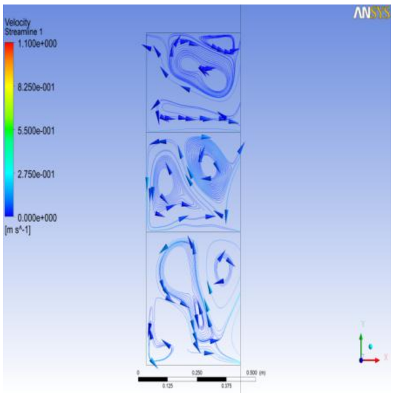


6^{0}C, 120 sec

Figure 3.12 Velocity contours at the 5th and 120th seconds at the refrigerator with 1 shelf at 2^{0}C and 6^{0}C starting temperature.

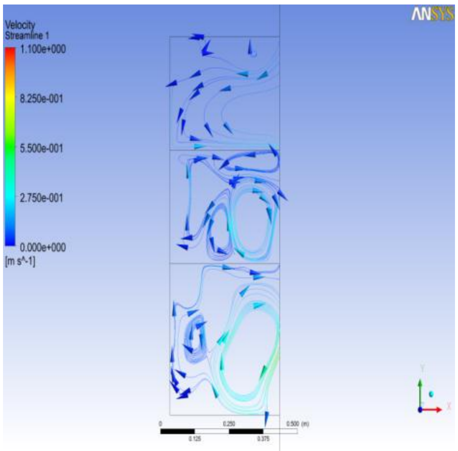


2^{0}C, 5 sec

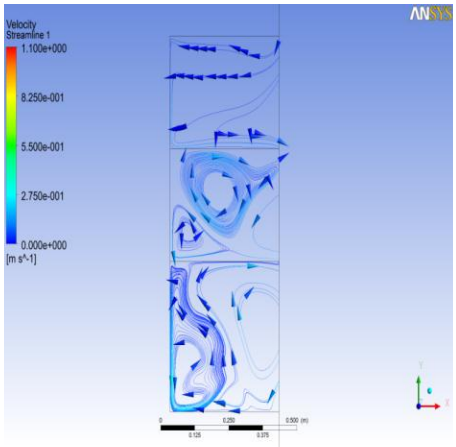


2^{0}C, 120 sec

Figure 3.13 Velocity contours at the 5th and 120th seconds at the refrigerator with 2 shelves at 2^{0}C and 6^{0}C starting temperature.

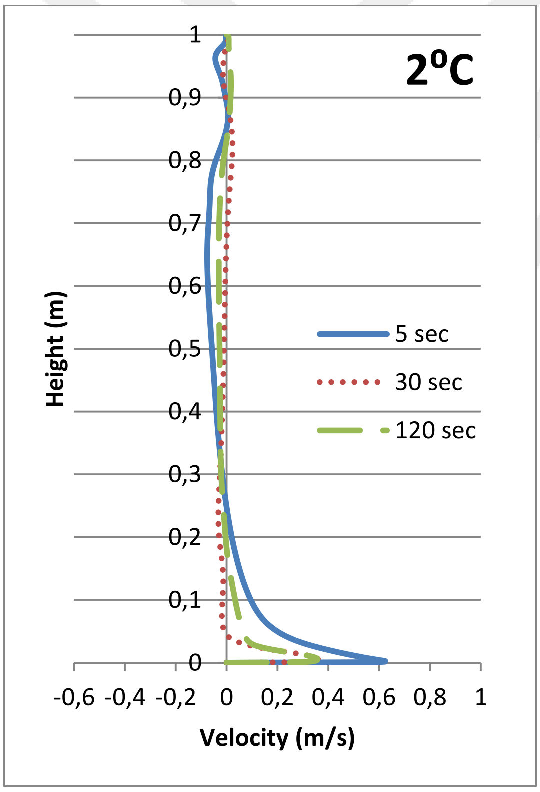


6^{0}C, 5 sec



6^{0}C, 120 sec

Figure 3.13 Velocity contours at the 5th and 120th seconds at the refrigerator with 2 shelves at 2^{0}C and 6^{0}C starting temperature. (continue)



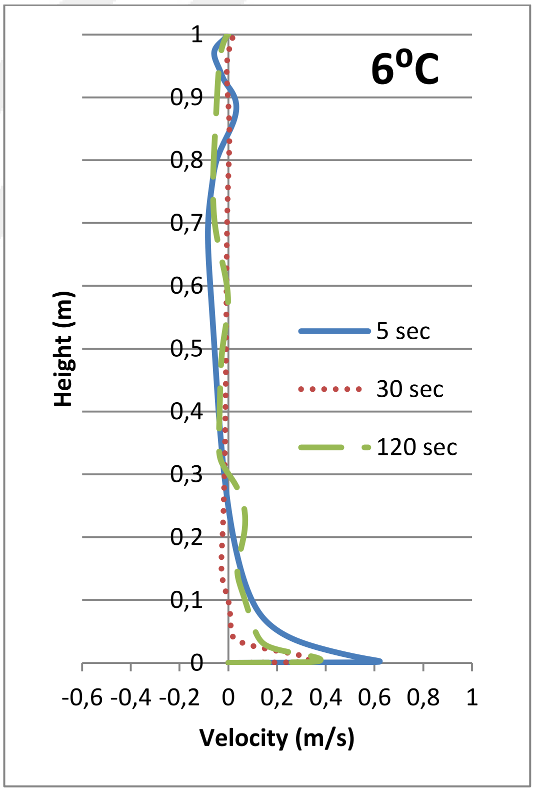


Figure 3.14 Velocity distributions at the refrigerator door plane for no shelf condition at 5th, 30th and 120th seconds when the starting temperature is 2^{0}C and 6^{0}C.

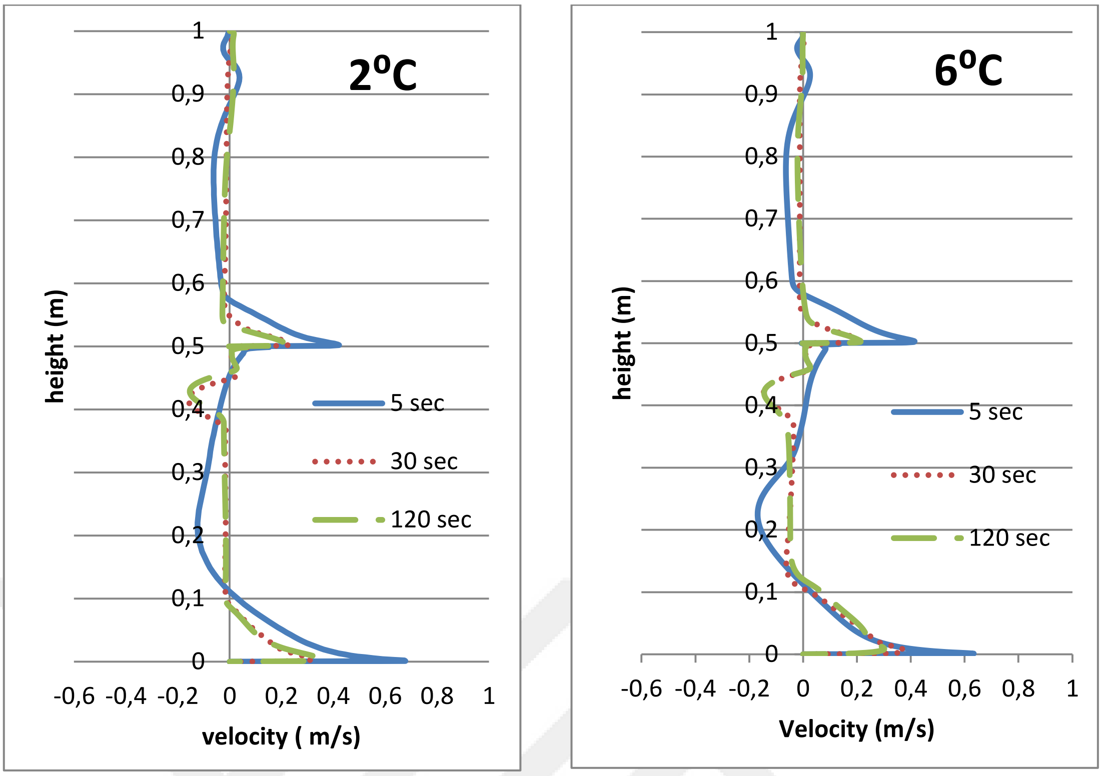


Figure 3.15 Velocity distributions at the refrigerator door plane for 1 shelf condition at 5th, 30th and 120th seconds when the starting temperature is 2^{0}C and 6^{o}C.

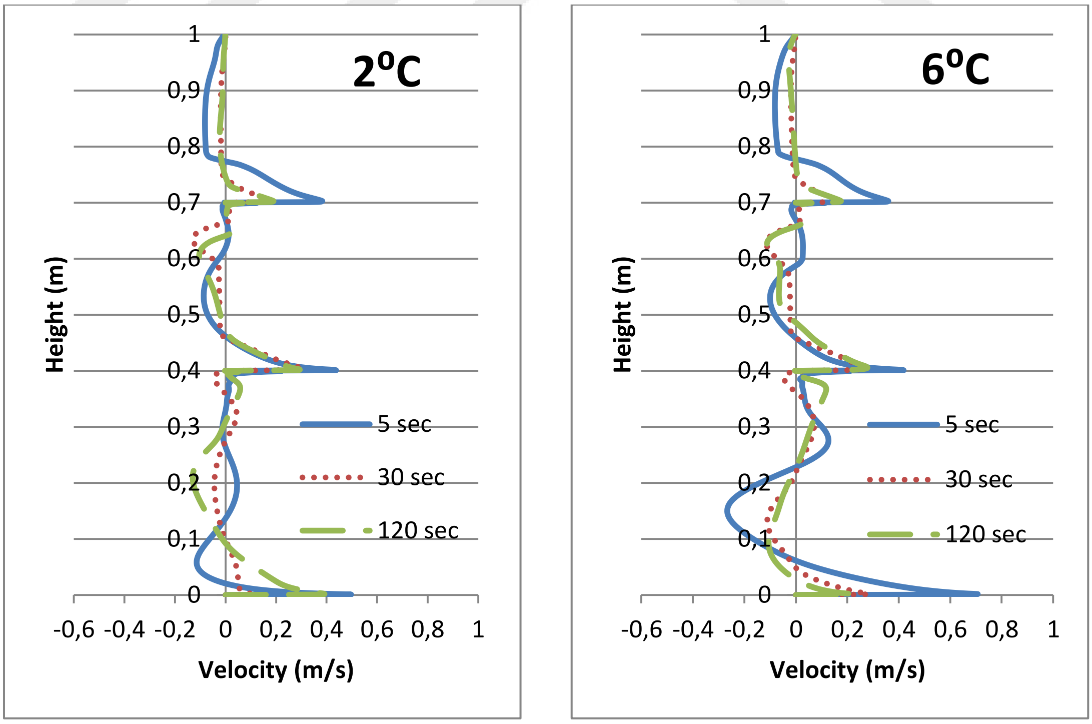


Figure 3.16 Velocity distributions at the refrigerator door plane for 2 shelves condition at 5th, 30th and 120th seconds when the starting temperature is 2^{0}C and 6^{o}C.

Achieved velocity contours and velocity distributions at the refrigerator door plane shows that most of air flow which is from the outside to inside of the refrigerator is occurs at the middle of shelf plane (Figure 3.11-3.13). In the same way the air flow which is from inside to outside seems just after the shelves. We concluded that it is because of the velocity of cold air flow which is occurs at the shelf area from inside to outside is bigger than the velocity of warm air flow which is normally occurs after shelf (from inside to outside) so there is vacuum effect just after the shelf area(Figure 3.14-3.16).

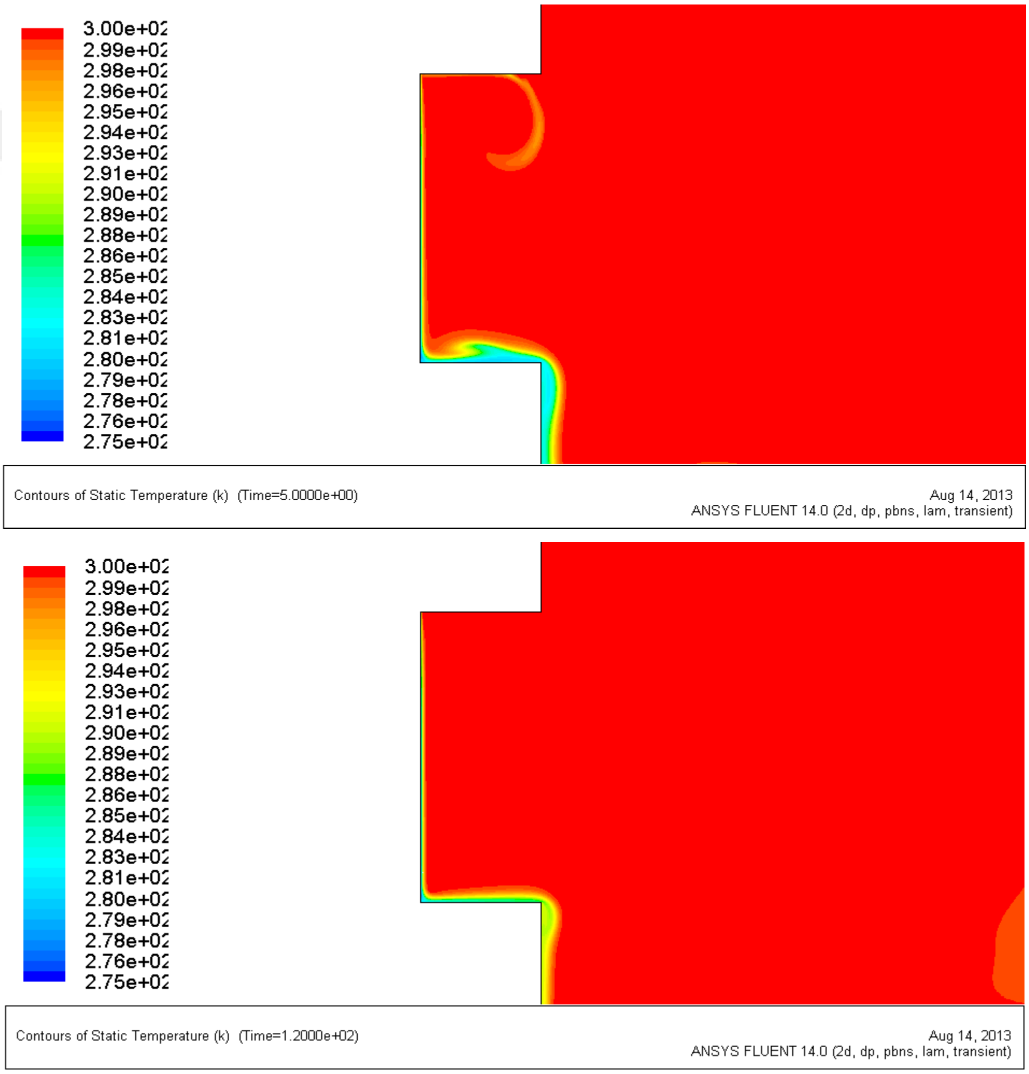


Figure 3.17 Temperature contours of the refrigerator and the opening on it at no shelf condition with 2^0C starting temperature at 5th and 120th sec.

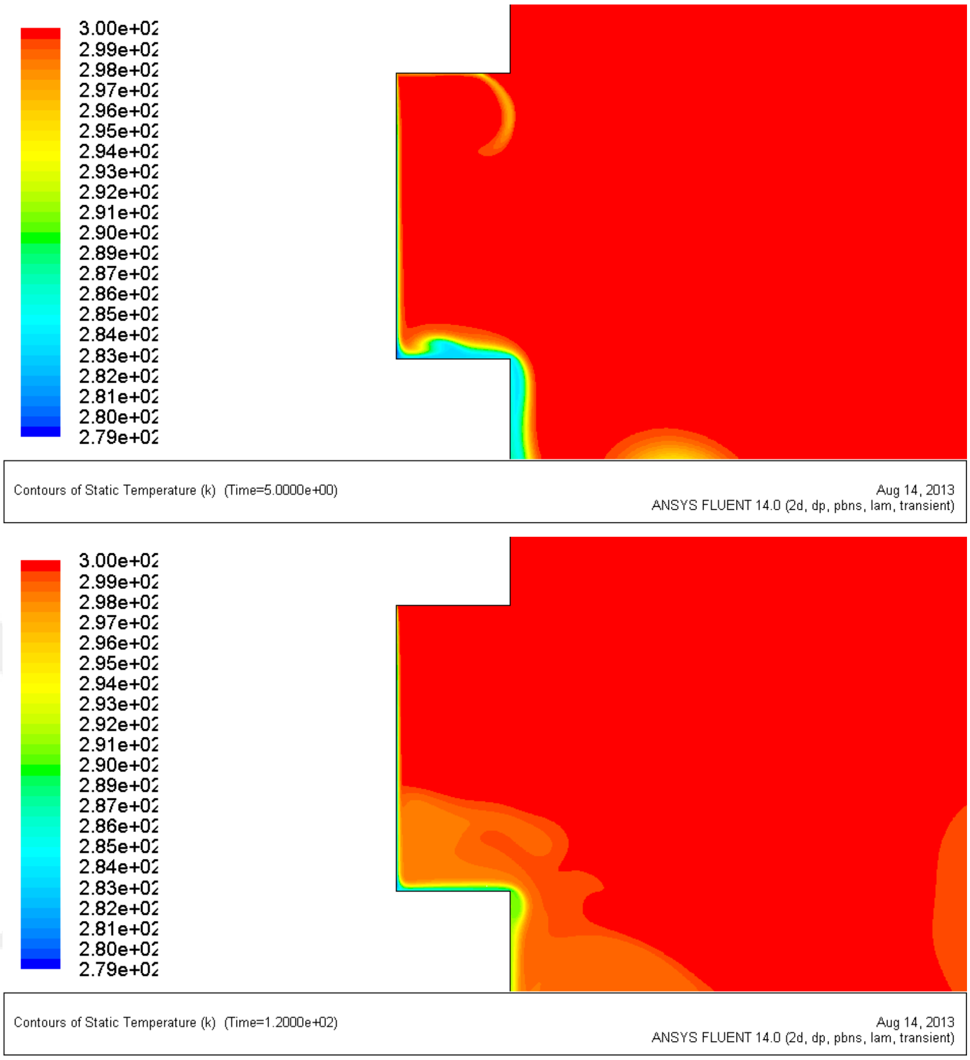


Figure 3.18 Temperature contours of the refrigerator and the opening on it at no shelf condition with 6^0C starting temperature at 5th and 120th sec.

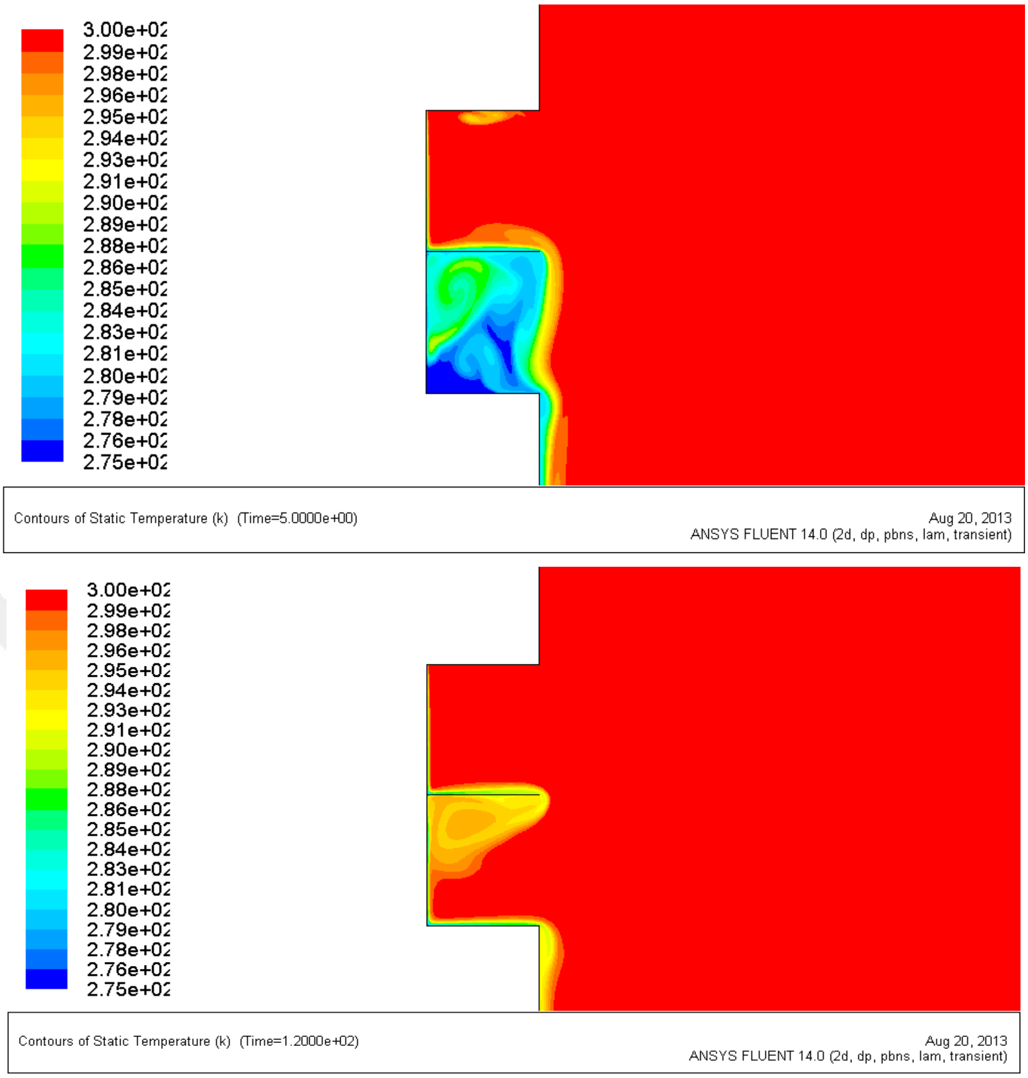


Figure 3.19 Temperature contours of the refrigerator and the opening on it at 1 shelf condition with 2^{0}C starting temperature at 5th and 120th sec.

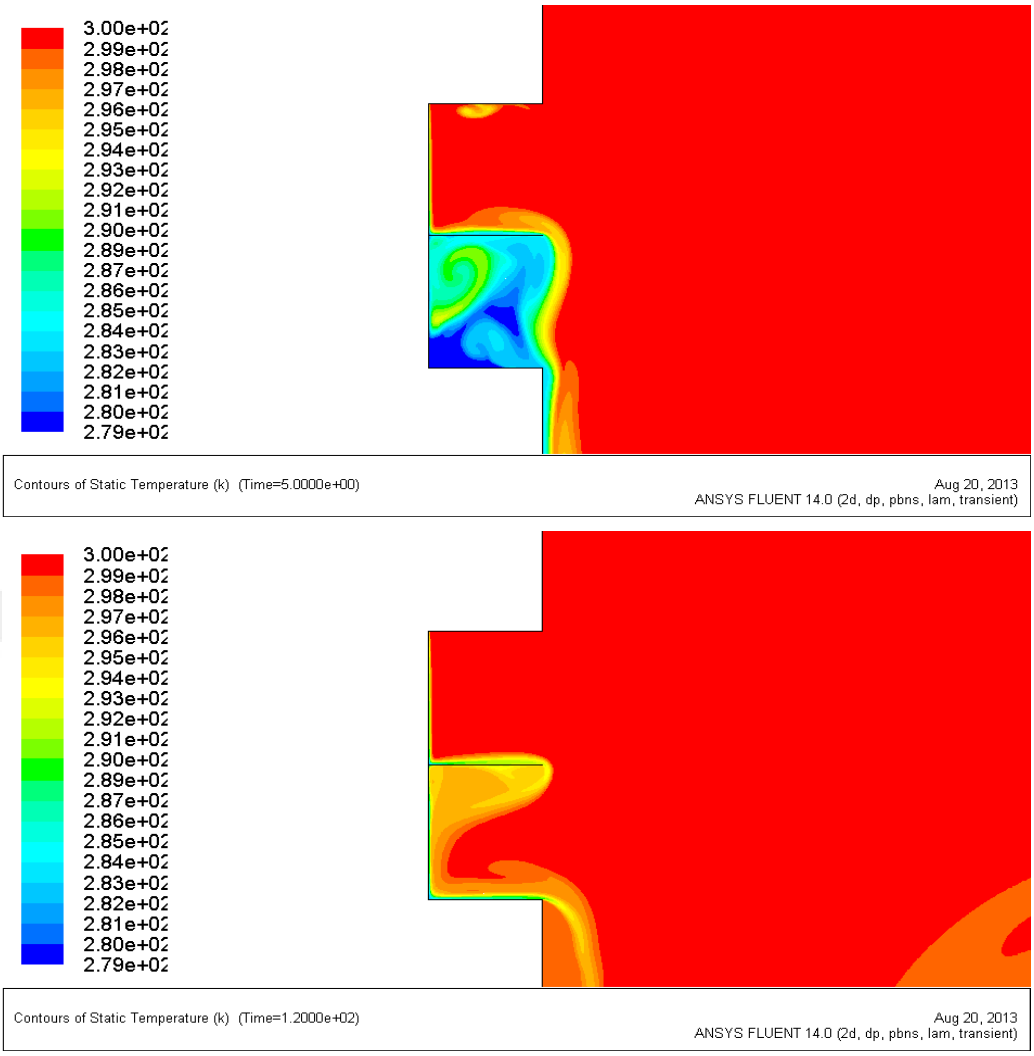


Figure 3.20 Temperature contours of the refrigerator and the opening on it at 1 shelf condition with 6^0C starting temperature at 5th and 120th sec.

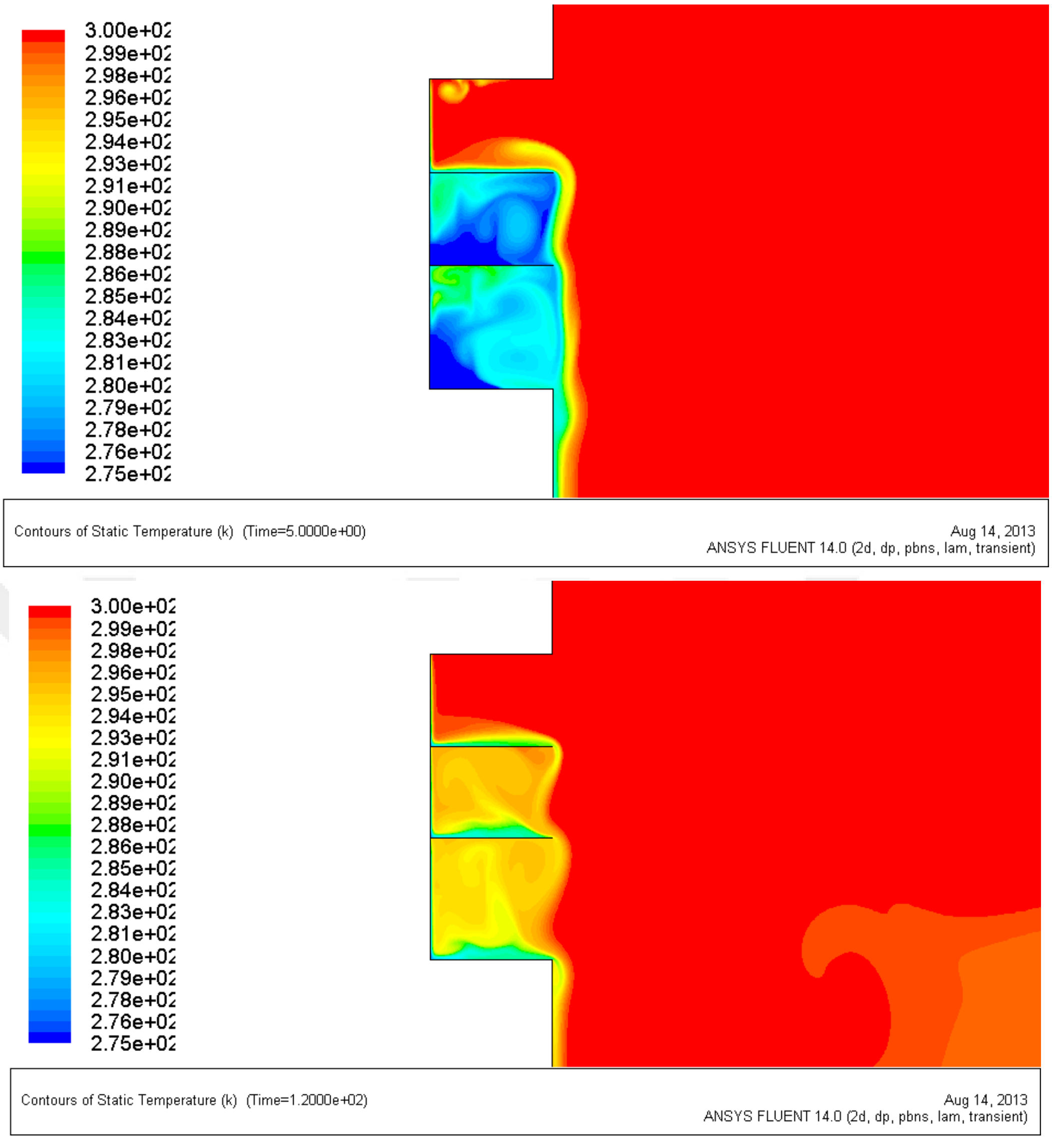


Figure 3.21 Temperature contours of the refrigerator and the opening on it at 2 shelves condition with 2^0C starting temperature at 5th and 120th sec.

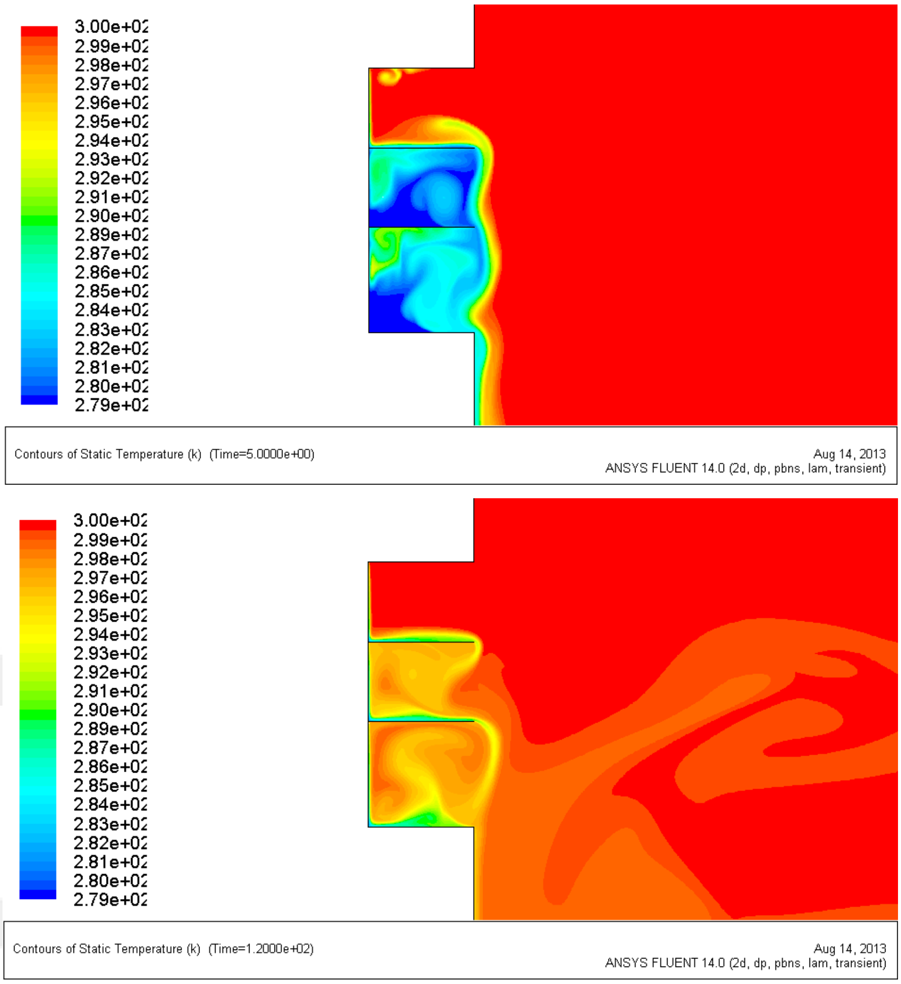


Figure 3.22 Temperature contours of the refrigerator and the opening on it at 2 shelves condition with 6^{0}C starting temperature at 5th and 120th sec.

The temperature contours show that most of energy earning is occurs at first 5 seconds. Near of that shelves are really helpful for saving refrigerator energy because they are blocking fluid flow which is coming from ambience (Figure 3.17-3.22).

Table 3.2 The Energy earning at the 120th second for different shelf and starting temperature conditions.

	2^{o}C	**4^{o}C**	**6^{o}C**
No shelf	11328.027	10395	9298.669
One shelf	10697.599	9782.097	8921.93
Two shelves	9288.687	9010.127	8289.638

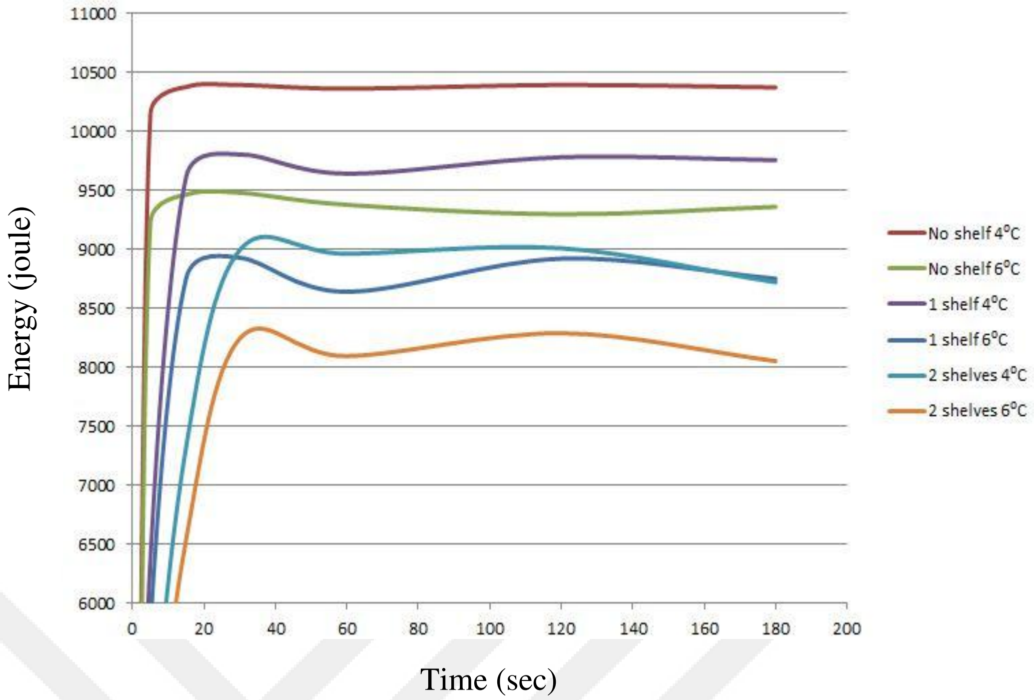


Figure 3.23 Energy earnings for different shelf and working temperature conditions.

The analysis what we done shows the shelves are making lesser the heat gain and higher working temperature of refrigerator causing less energy earning (Figure 3.23).

3.4 Results and Discussion of 3D Model

3.4.1 Results for 4 Shelves and Initial Temperature of Refrigerator of 2^0C

In 1 second, the warm air goes in the fridge in room volume (Figure 3.26). Because the cold air is heavier than the warm air, it leaves the close places from the bottom surfaces of the shelf volume (Figure 3.24). The cold air with more intense is like a barrier against the warm air while going down.

In 5 seconds, because the cold and warm airs almost shift places, the barrier caused by the cold air disappears and the warm air starts shifting place with the cold air on the 2nd shelf (Figure 3.26). The cold air which goes out from second shelf protects the lower shelves from the warm air which is coming from room volume.

In 15 seconds, After the big part of the cold air goes out from the 1st and the 2nd shelves, the warm air newly entering the 1st shelf touching the cold surfaces of the fridge and losing the heat and for that, its density increases. While the air with more density and less temperature leaves the shelf from the areas close to the bottom surfaces of the shelf volume (Figure 3.26), it blocks the warmer air in room volume which wants to go into the lower shelves. It can be described as the warm air wall. Yet, it cannot prevent the heat transportation and it gets the heat by convection heat transfer on the bottom shelves.

In the 30th and the 60th seconds, the amount of energy of the shelf volume is increased (Figure 3.25 and 3.27). The shelves volumes keep gaining heat by convection heat transfer after 15th second.

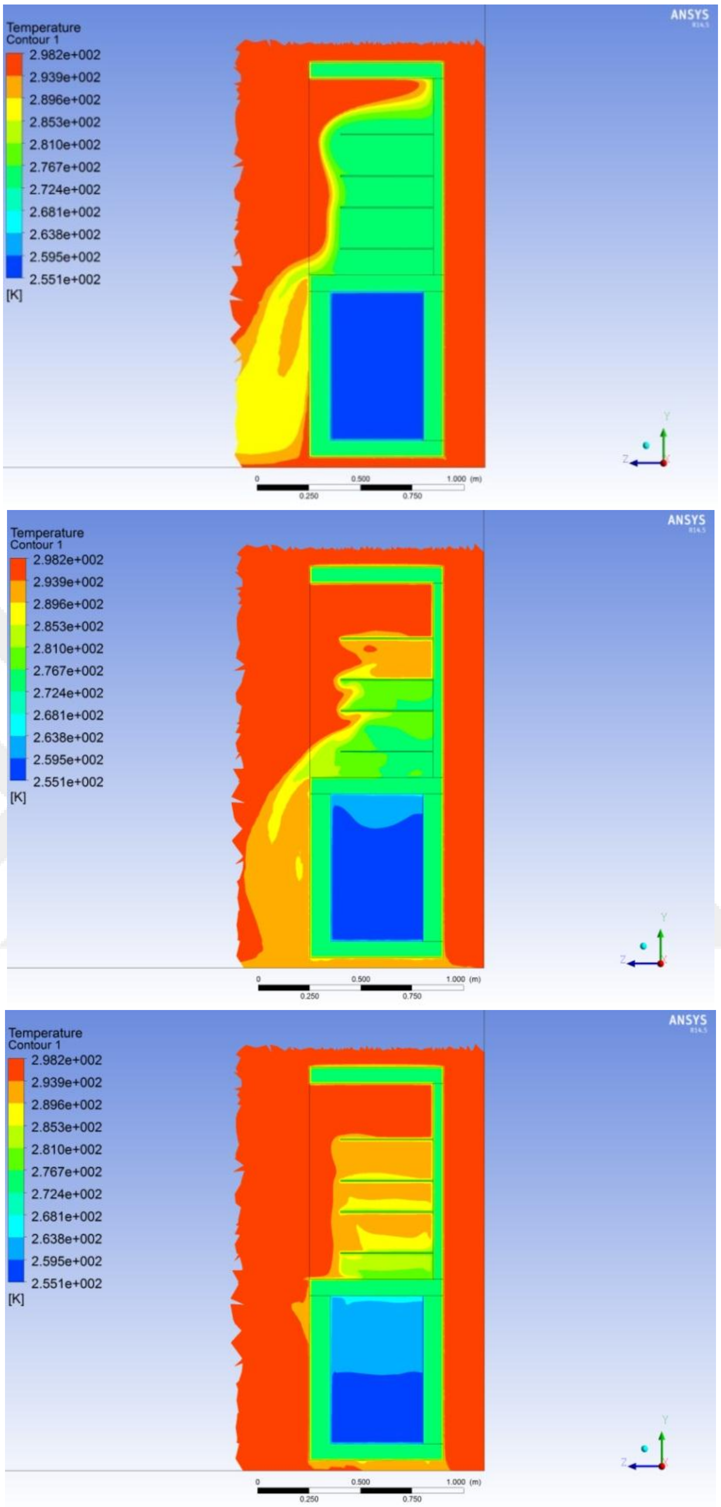


Figure 3.24 Temperature contours of the refrigerator and the opening on it at 4 shelves condition with 2^0C starting temperature at 1^{st}, 5^{th} and 15^{th} sec.

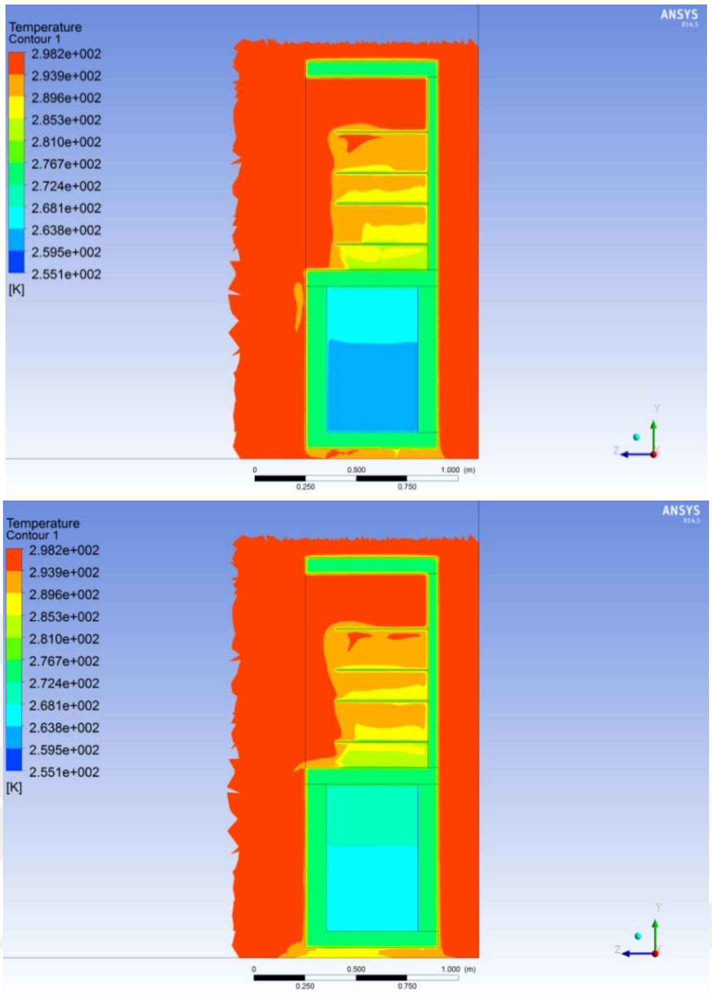


Figure 3.25 Temperature contours of the refrigerator and the opening on it at 4 shelves condition with 2^0C starting temperature at 30^{th} and 60^{th} sec.

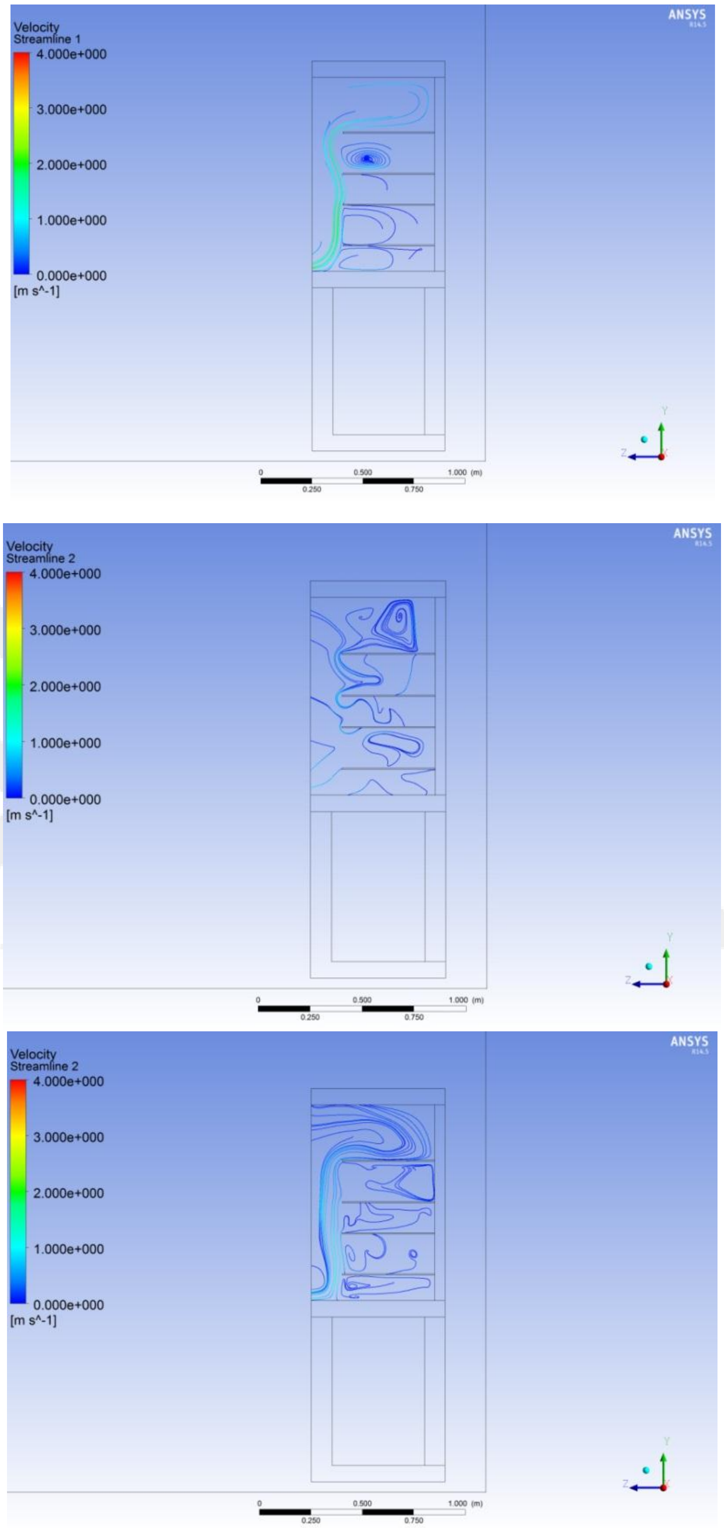


Figure 3.26 Streamline of the refrigerator and the opening on it at 4 shelves condition with 2^0C starting temperature at 1^{st}, 5^{th} and 15^{th} sec.

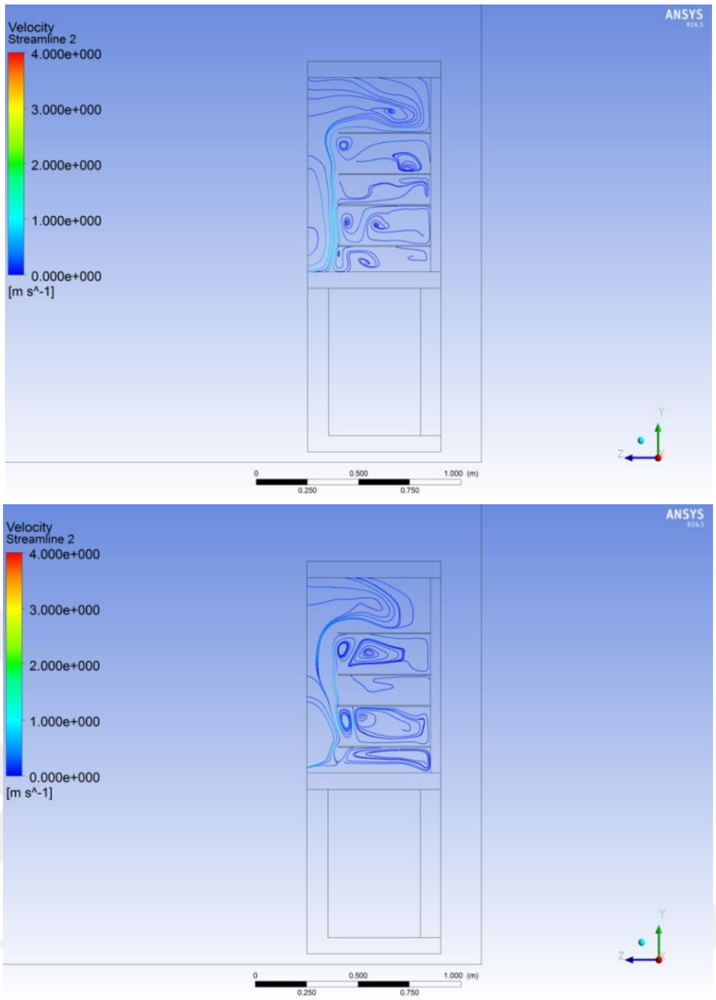


Figure 3.27 Streamline of the refrigerator and the opening on it at 4 shelves condition with 2^{0}C starting temperature at 30th and 60th sec.

3.4.2 Results for 4 Shelves and Initial Temperature of Refrigerator of 2^0C for Different Shelf Heights

In 1st second, the amount of the warm air going through the shelf volume gets less because of the first shelf volume was decreases according to the situation on 3.4.1. Nevertheless, the cold air going out from the 1st shelf keeps its barrier effect against the warm air around the lower shelves (Figure 3.28 and 3.30).

In 5th seconds, it can be observed more of the warm air enters to second shelf compared to the situation on 3.4.1. It can be seen the barrier effect gets stronger between the lower shelves and the room volume, because the increase of the cold air going out of the 2nd shelf (Figure 3.30).

In 15th second, it can be seen that the 2nd shelf volume gains more energy compared to the situation on 3.4.1. It also can be seen energy can be gained by convective heat transfer, yet the mass is not changed around the lower areas.

In the 30th and the 60 seconds, energy gain continues with convective heat transfer just like 15th second (Figure 3.29 and 3.31).

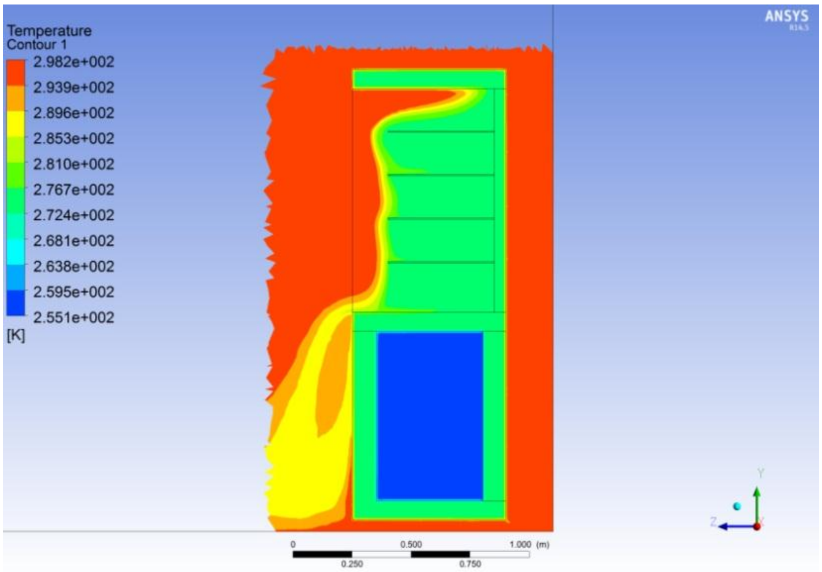


Figure 3.28 Temperature contours of the refrigerator and the opening on it at 4 shelves condition with 2^0C starting temperature at 1^{st}, 5^{th} and 15^{th} sec.

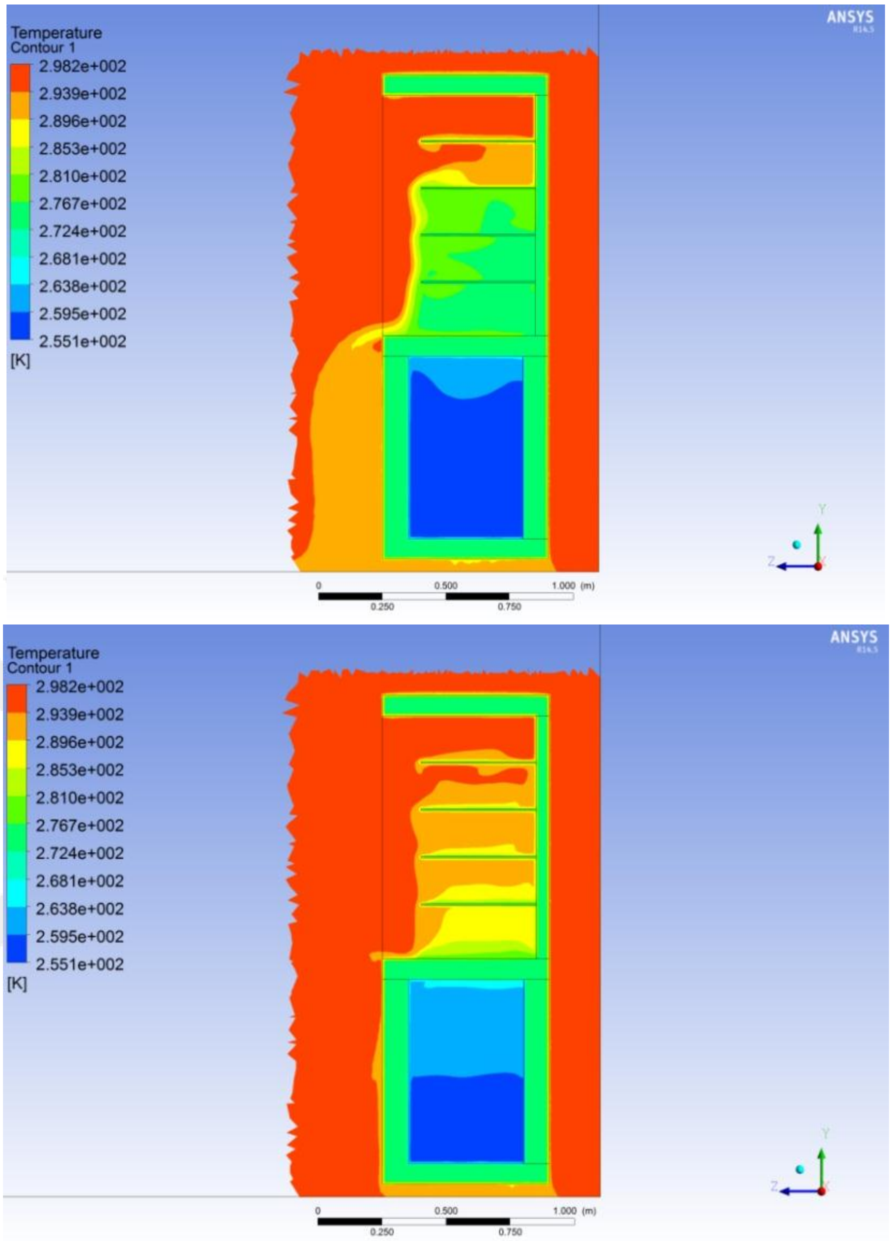


Figure 3.28 Temperature contours of the refrigerator and the opening on it at 4 shelves condition with 2^{0}C starting temperature at 1^{st}, 5^{th} and 15^{th} sec. (continue)

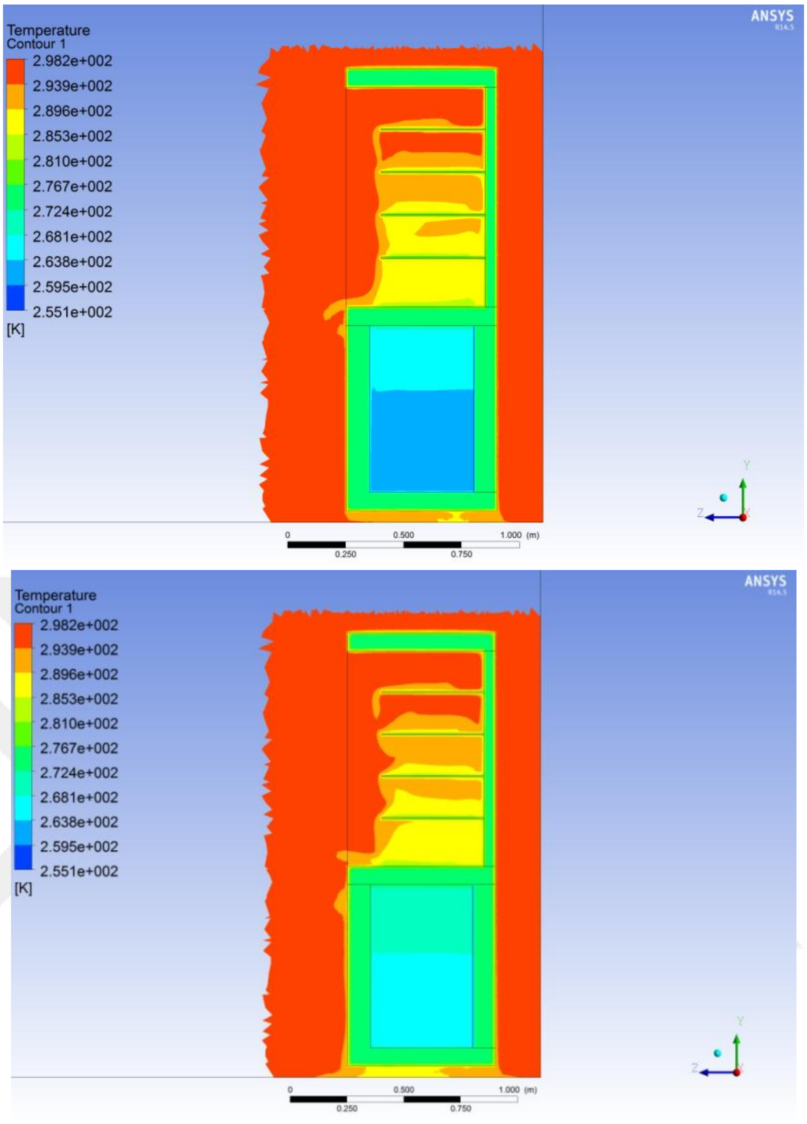


Figure 3.29 Temperature contours of the refrigerator and the opening on it at 4 shelves condition with 2^0C starting temperature at 30^{th} and 60^{th} sec.

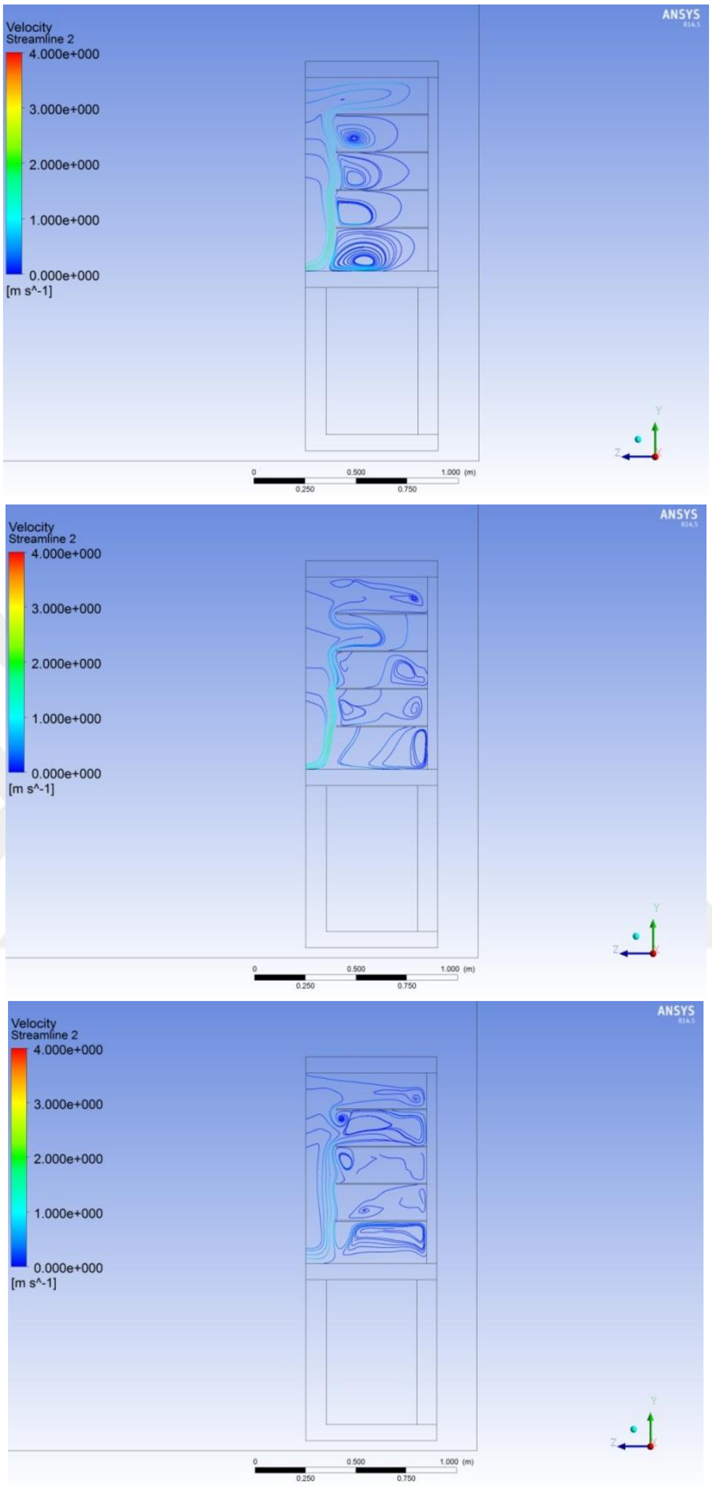


Figure 3.30 Streamline of the refrigerator and the opening on it at 4 shelves condition with 2^{0}C starting temperature at 1st, 5th and 15th sec.

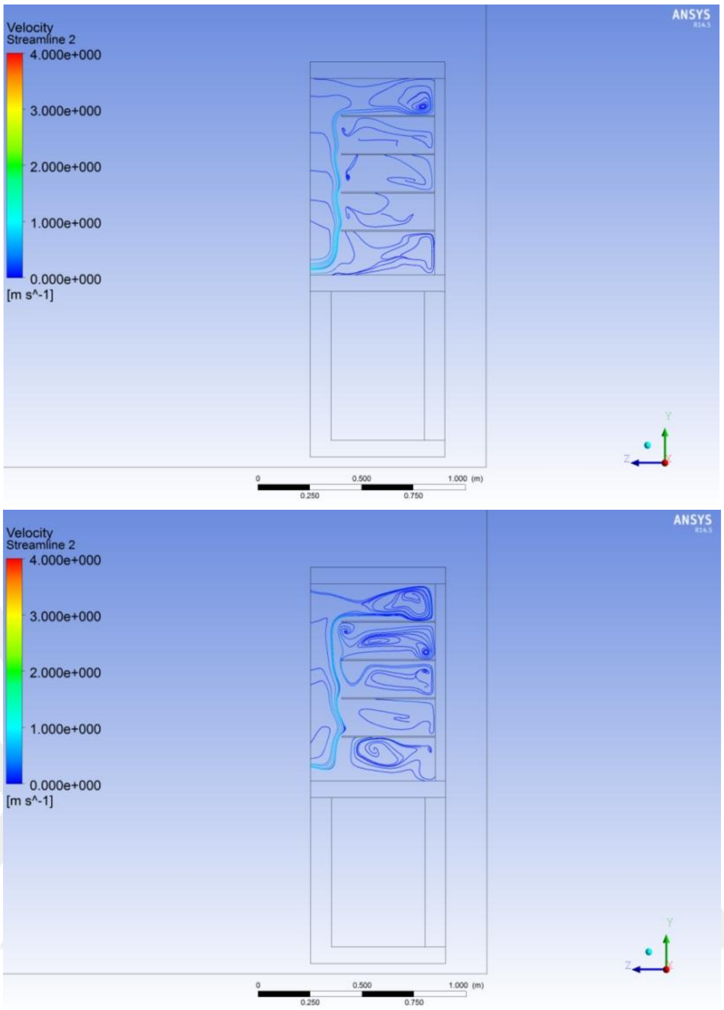


Figure 3.31 Streamline of the refrigerator and the opening on it at 4 shelves condition with 2^0C starting temperature at 30^{th} and 60^{th} sec.

3.4.3 Results for 4 Shelves and Initial Temperature of Refrigerator of 4^{0}C

2^{0}C and 4^{0}C of the starting temperature did not make a big difference on the stream lines and the heat contour which were taken on the specific seconds. The only difference in the system is the volume is 2^{0}C higher than the start. As it has more energy, it gains less energy here (Figure 3.32-3.35).

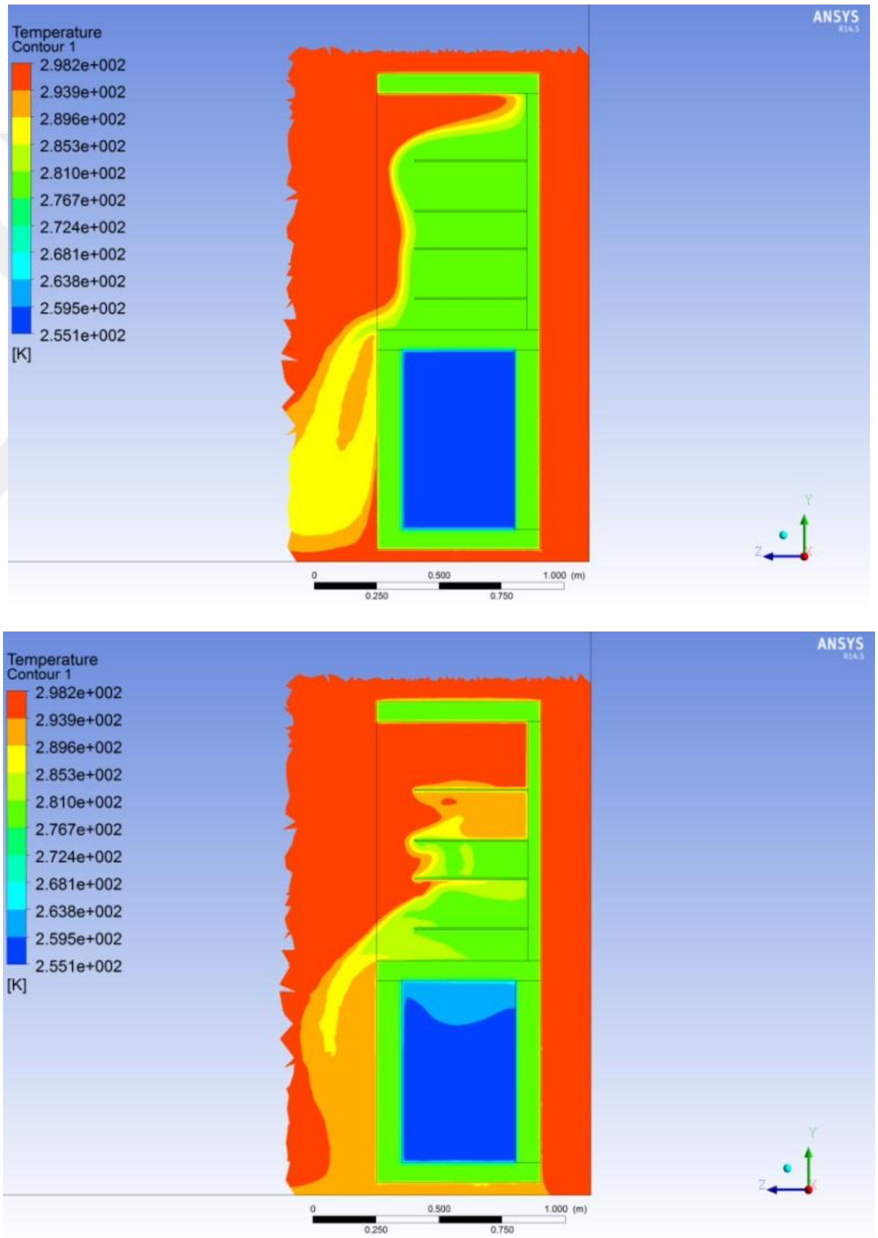


Figure 3.32 Temperature contours of the refrigerator and the opening on it at 4 shelves condition with 4^{0}C starting temperature at 1st, 5th and 15th sec.

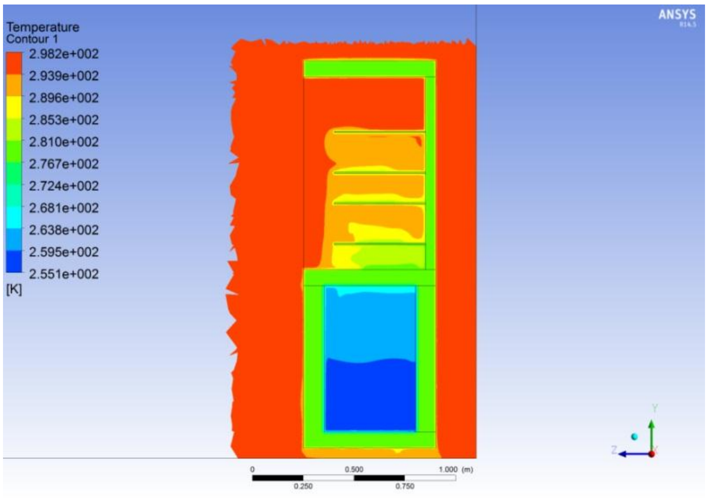


Figure 3.32 Temperature contours of the refrigerator and the opening on it at 4 shelves condition with 4^{0}C starting temperature at 1st, 5th and 15th sec. (continue)

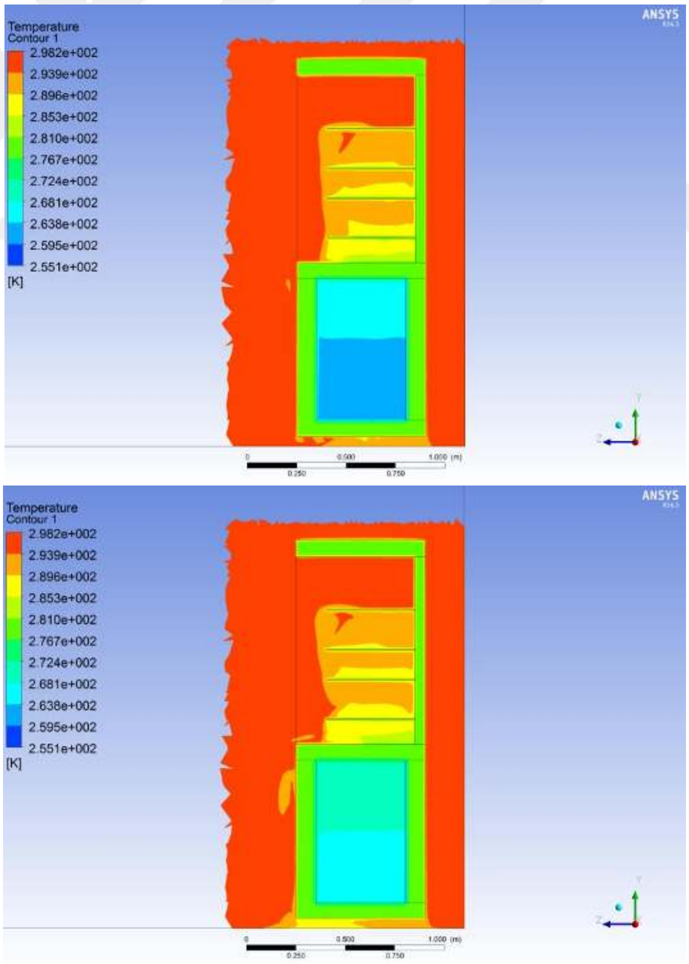


Figure 3.33 Temperature contours of the refrigerator and the opening on it at 4 shelves condition with 4^{0}C starting temperature at 30th and 60th sec.

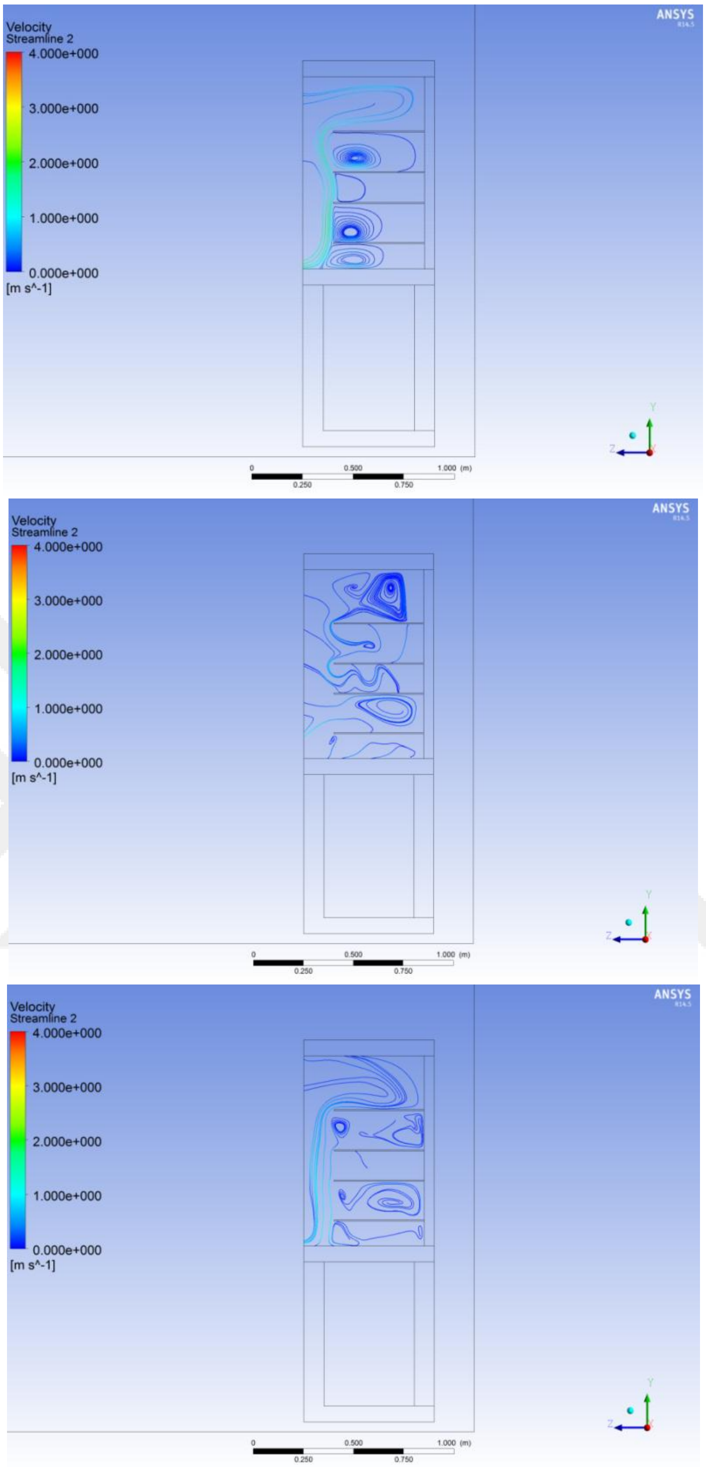


Figure 3.34 Streamline of the refrigerator and the opening on it at 4 shelves condition with 4^{0}C starting temperature at 1^{st}, 5^{th} and 15^{th} sec.

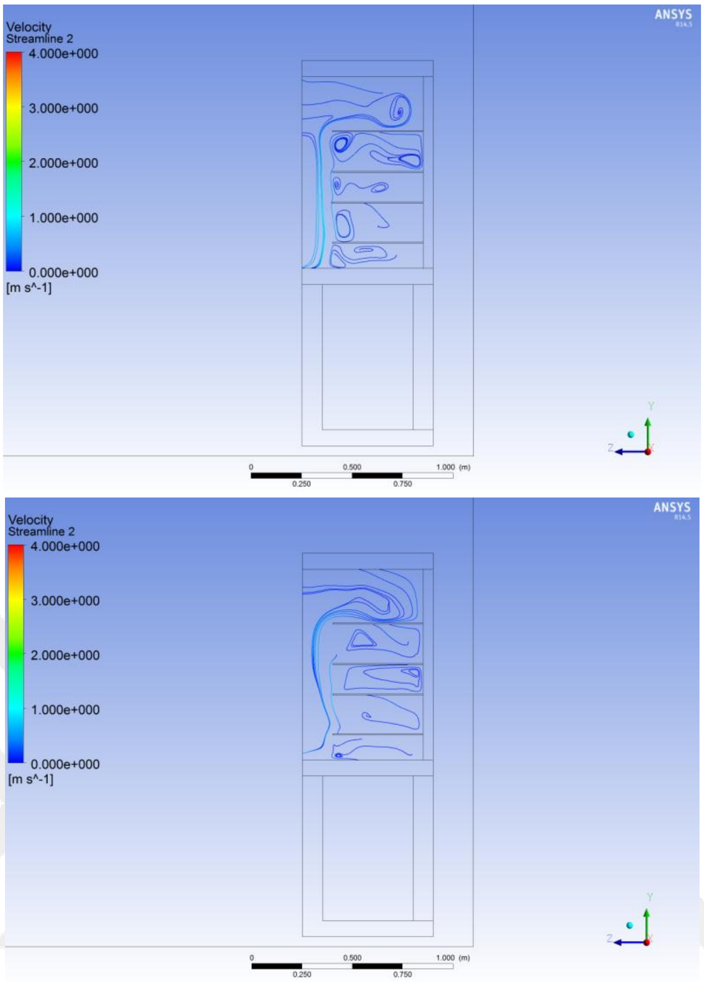


Figure 3.35 Streamline of the refrigerator and the opening on it at 4 shelves condition with 4^{0}C starting temperature at 30th and 60th sec.

3.4.4 Results for 4 Shelves and Initial Temperature of Refrigerator of 4^{0}C for Different Shelf Heights

Because the starting temperature is 2^{0}C and 4^{0}C does not create a big difference, it can be seen the same results between 4 shelves 2^{0}C and 4 shelves 2^{0}C different heights differences (Figure 3.36 -3.39). That supports the accuracy of the analysis.

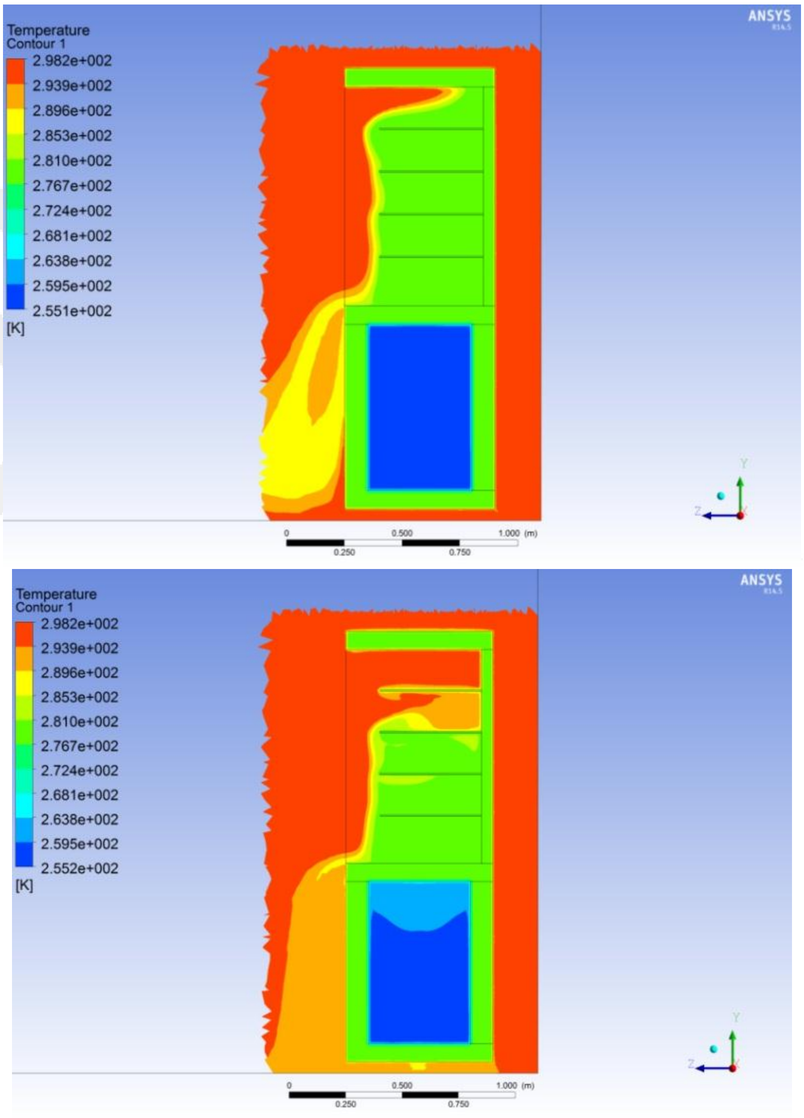


Figure 3.36 Temperature contours of the refrigerator and the opening on it at 4 shelves condition with 4^{0}C starting temperature at 1st, 5th and 15th sec.

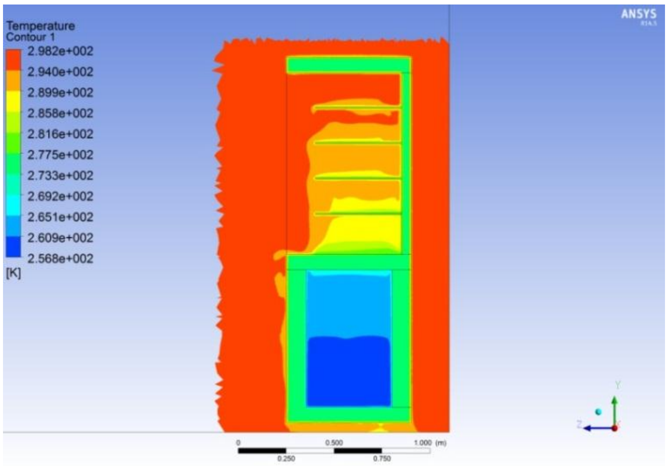


Figure 3.36 Temperature contours of the refrigerator and the opening on it at 4 shelves condition with 4^0C starting temperature at 1^{st}, 5^{th} and 15^{th} sec .(continue)

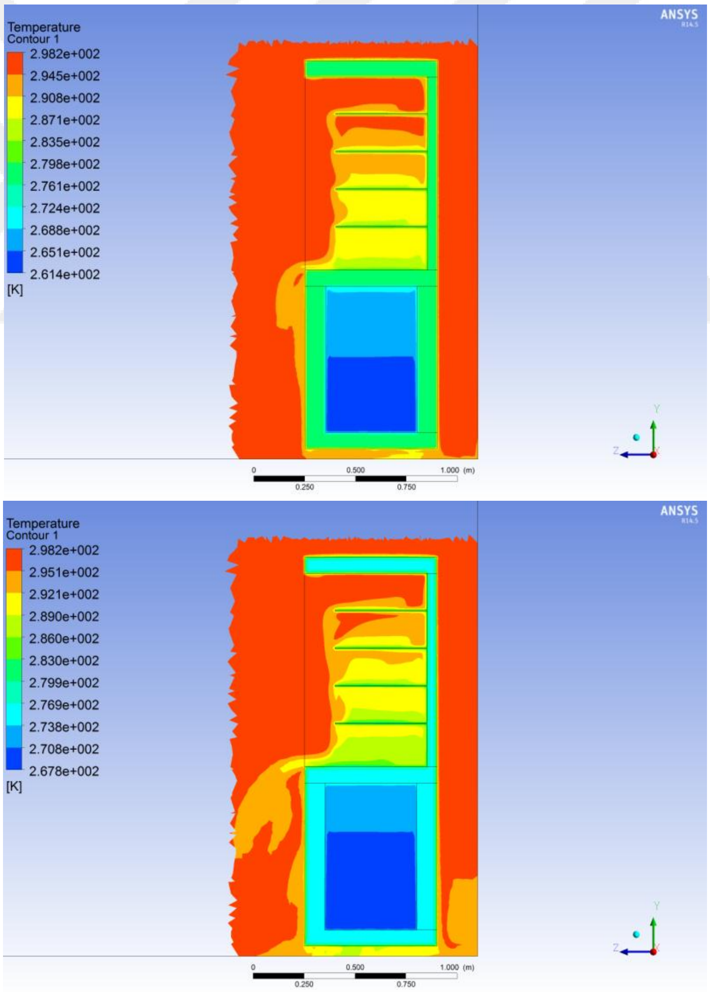


Figure 3.37 Temperature contours of the refrigerator and the opening on it at 4 shelves condition with 4^0C starting temperature at 30^{th} and 60^{th} sec.

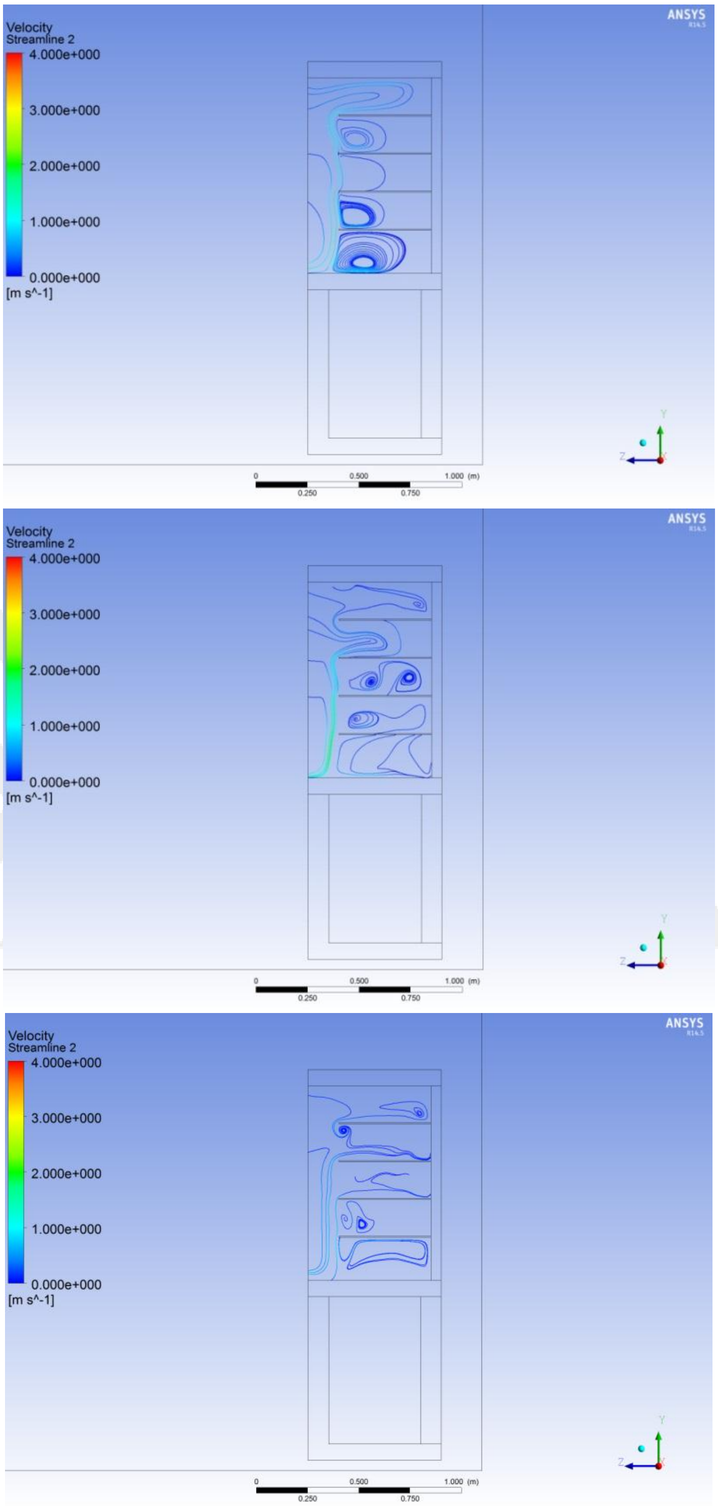


Figure 3.38 Streamline of the refrigerator and the opening on it at 4 shelves condition with 4^0C starting temperature at 1^{st}, 5^{th} and 15^{th} sec.

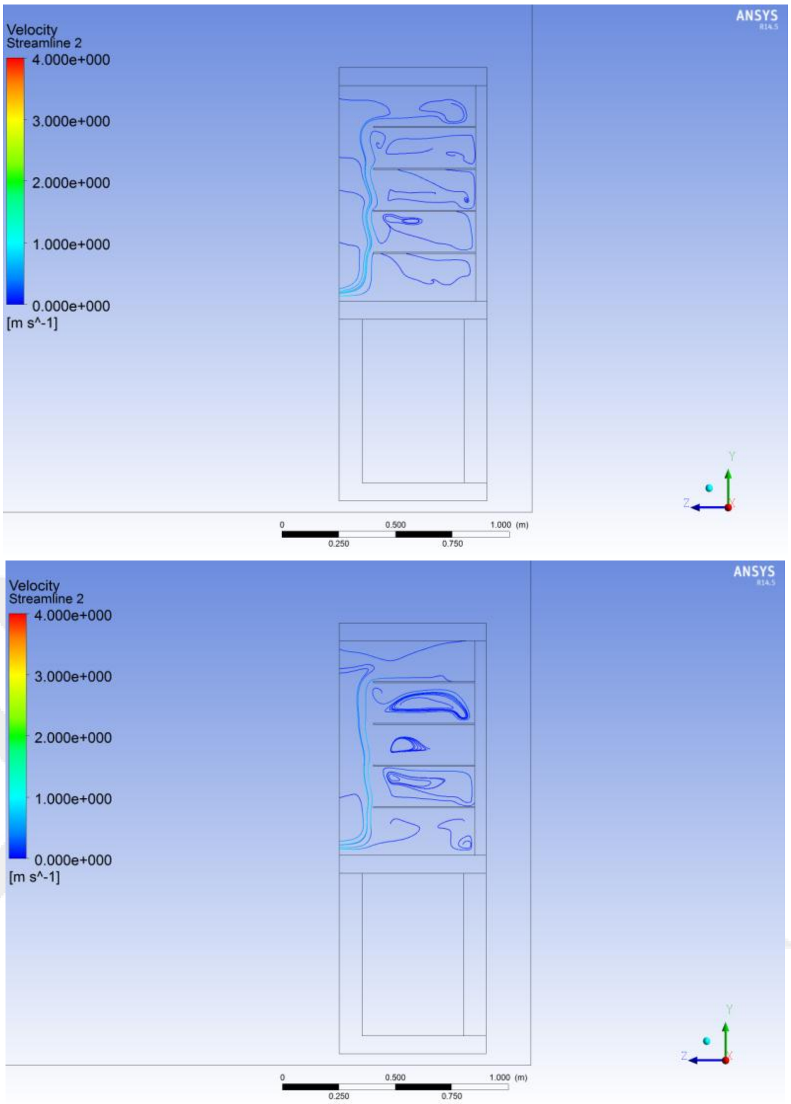


Figure 3.39 Streamline of the refrigerator and the opening on it at 4 shelves condition with 4^{0}C starting temperature at 30th and 60th sec.

3.4.5 Results for 3 Shelves and Initial Temperature of Refrigerator of 2^0C

In 1 second, the warm air goes in the fridge in room volume. Because the cold air is heavier than the warm air, it leaves the close places from the bottom surfaces of the shelf volume (Figure 3.42). The cold air with more intense is like a barrier against the warm air while going down.

In the 5th second, as we can see on the stream lines, between 1st-5th seconds shifting the cold-warm air on the 2nd shelf is done and the warm air coming, which is coming from room volume heads to the 3rd shelf (Figure 3.40 and 3.42).

In 15 seconds, After the big part of the cold air goes out from the 1st and the 2nd shelves, the warm air newly entering the 1st shelf touching the cold surfaces of the fridge and losing the heat and for that, its density increases. While the air with more density and less temperature leaves the shelf from the areas close to the bottom surfaces of the shelf volume, it blocks the warmer air in room volume which wants to go into the lower shelves. It can be described as the warm air wall (Figure 3.42). Yet, it cannot prevent the heat transportation and it gets the heat by convection heat transfer on the bottom shelves.

In the 30th and the 60th seconds, the amount of energy of the shelf volume is increased (Figure 3.41 and 3.43). The shelves volumes keep gaining heat by convection heat transfer after 15th second.

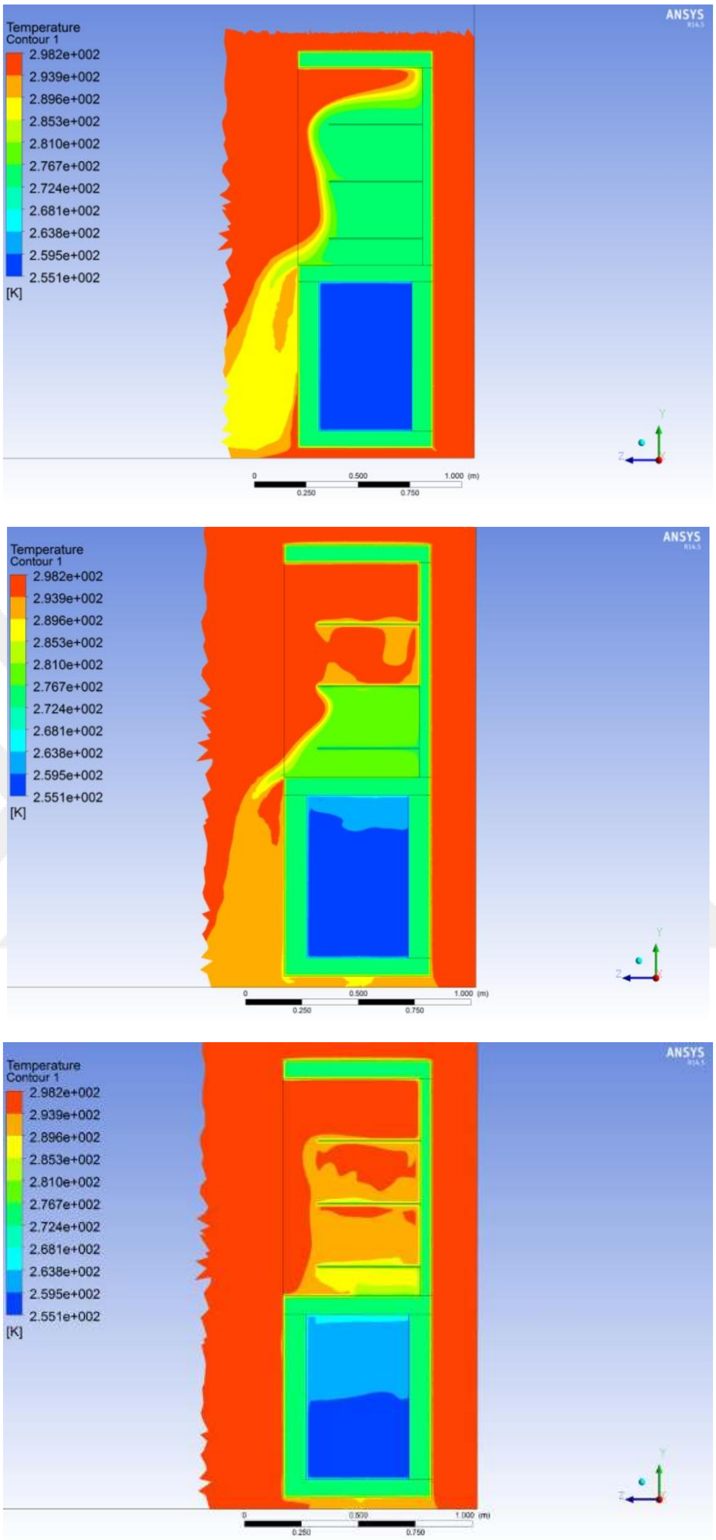


Figure 3.40 Temperature contours of the refrigerator and the opening on it at 3 shelves condition with 2^0C starting temperature at 1^{st}, 5^{th} and 15^{th} sec.

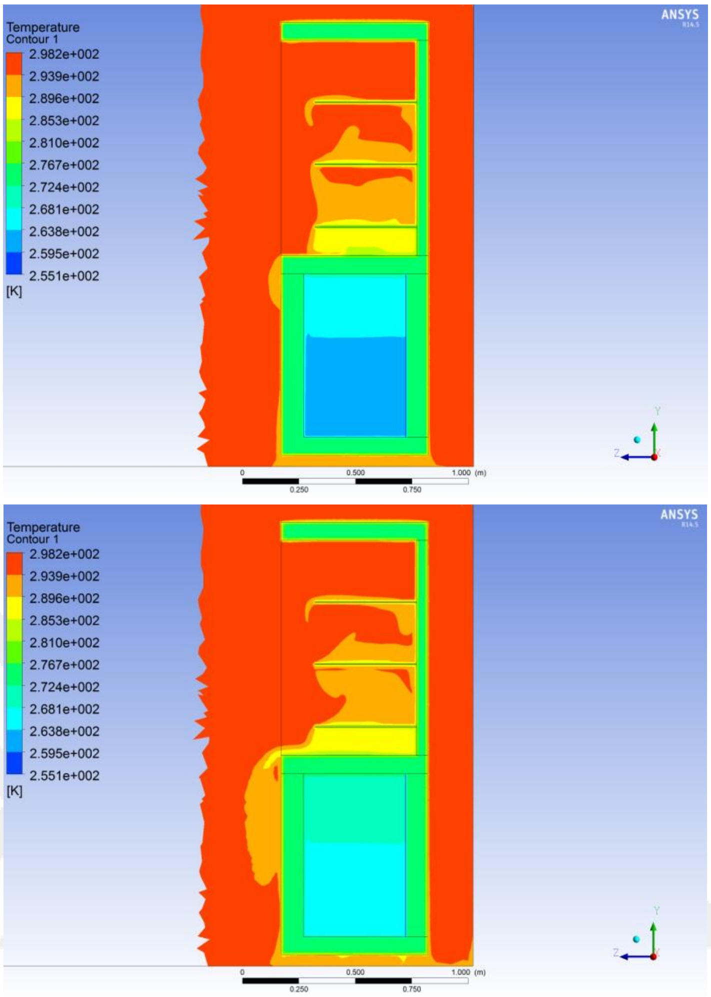


Figure 3.41 Temperature contours of the refrigerator and the opening on it at 3 shelves condition with 2^{0}C starting temperature at 30th and 60th sec.

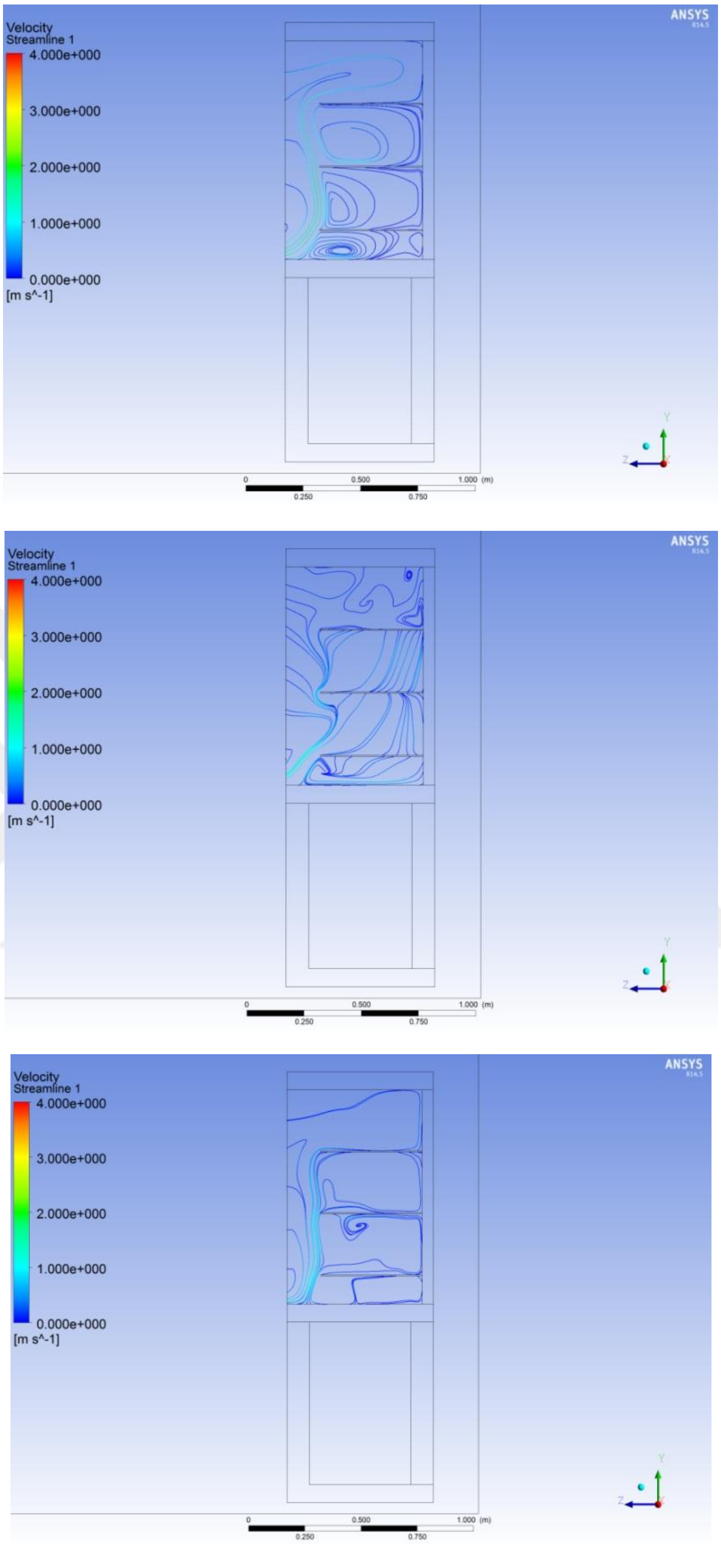


Figure 3.42 Streamline of the refrigerator and the opening on it at 3 shelves condition with 2^0C starting temperature at 1^{st}, 5^{th} and 15^{th} sec.

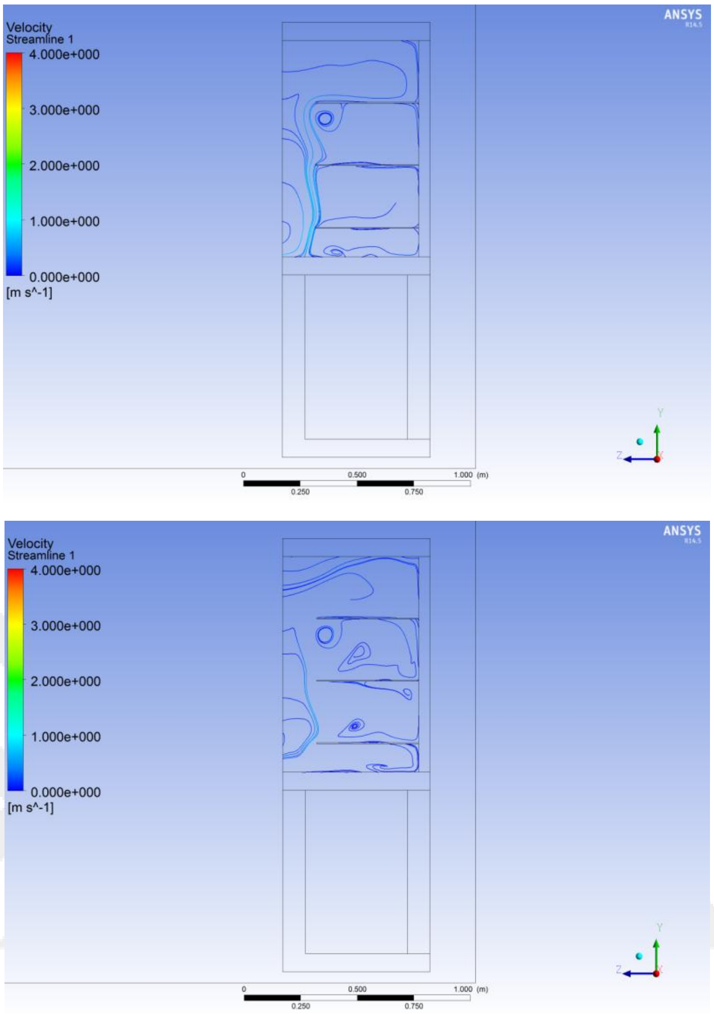


Figure 3.43 Streamline of the refrigerator and the opening on it at 3 shelves condition with2 ^{0}C starting temperature at 30th and 60th sec.

3.4.6 Results for 3 Shelves and Initial Temperature of Refrigerator of 2^0C for Different Shelf Heights

In 1st second, the amount of the warm air going through the shelf volume gets less because of the first shelf volume was decreases according to the situation on 3.4.5. Nevertheless, the cold air going out from the 1st shelf keeps its barrier effect against the warm air around the lower shelves (Figure 3.46).

In 5th seconds, it can be observed more of the warm air enters to second shelf compared to the situation on 3.4.5 (Figure 3.44 and 3.46). It can be seen the barrier effect gets stronger between the lower shelves and the room volume, because the increase of the cold air going out of the 2nd shelf.

In 15th second, it can be seen that the 2nd shelf volume gains more energy compared to the situation on 3.4.5. It also can be seen energy can be gained by convective heat transfer, yet the mass is not changed around the lower areas.

In the 30th and the 60 seconds, energy gain continues with convective heat transfer just like 15th second (Figure 3.45 and 3.47).

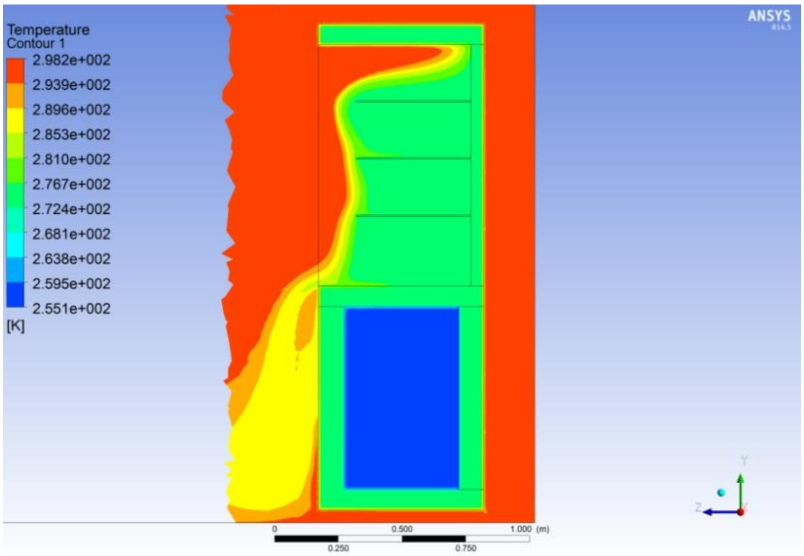
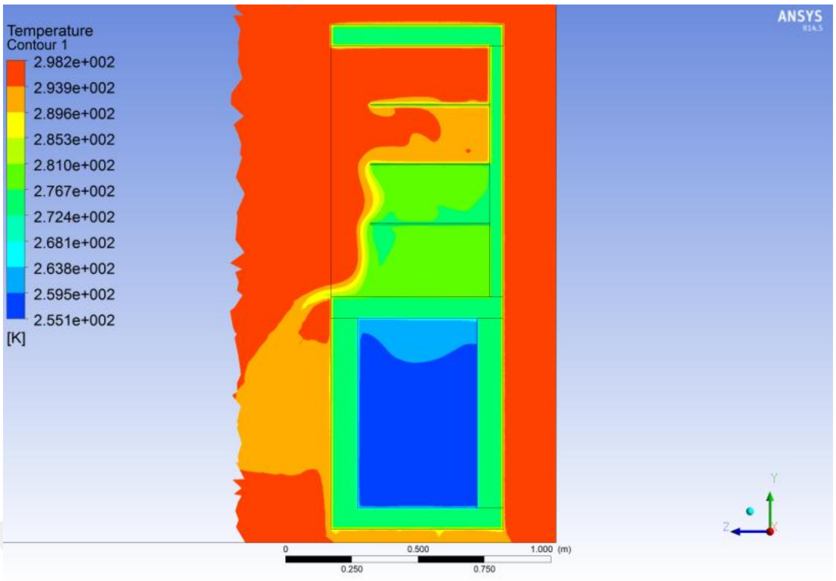


Figure 3.44 Temperature contours of the refrigerator and the opening on it at 3 shelves condition with 2^0C starting temperature at 1^{st}, 5^{th} and 15^{th} sec.



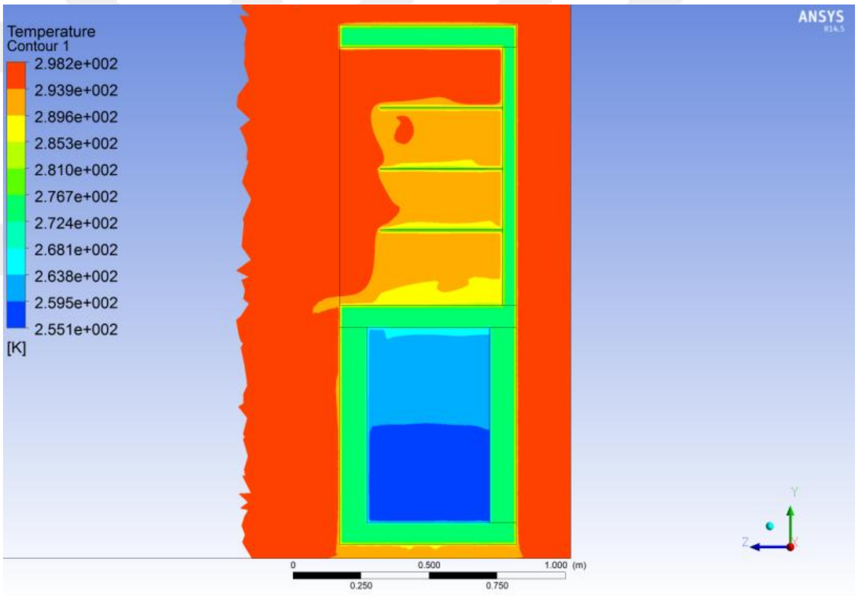


Figure 3.44 Temperature contours of the refrigerator and the opening on it at 3 shelves condition with 2^0C starting temperature at 1^{st}, 5^{th} and 15^{th} sec. (continue)

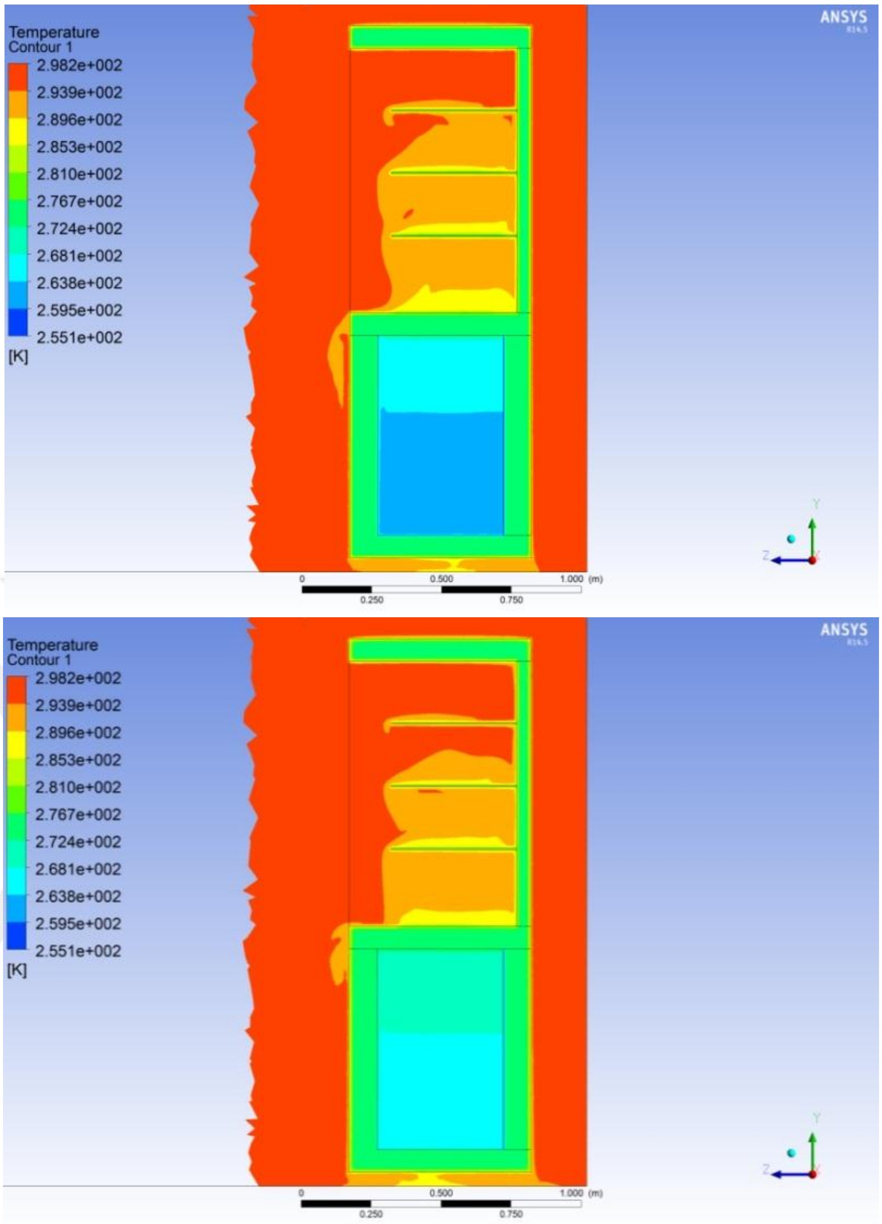


Figure 3.45 Temperature contours of the refrigerator and the opening on it at 3 shelves condition with 2^{0}C starting temperature at 30th and 60th sec.

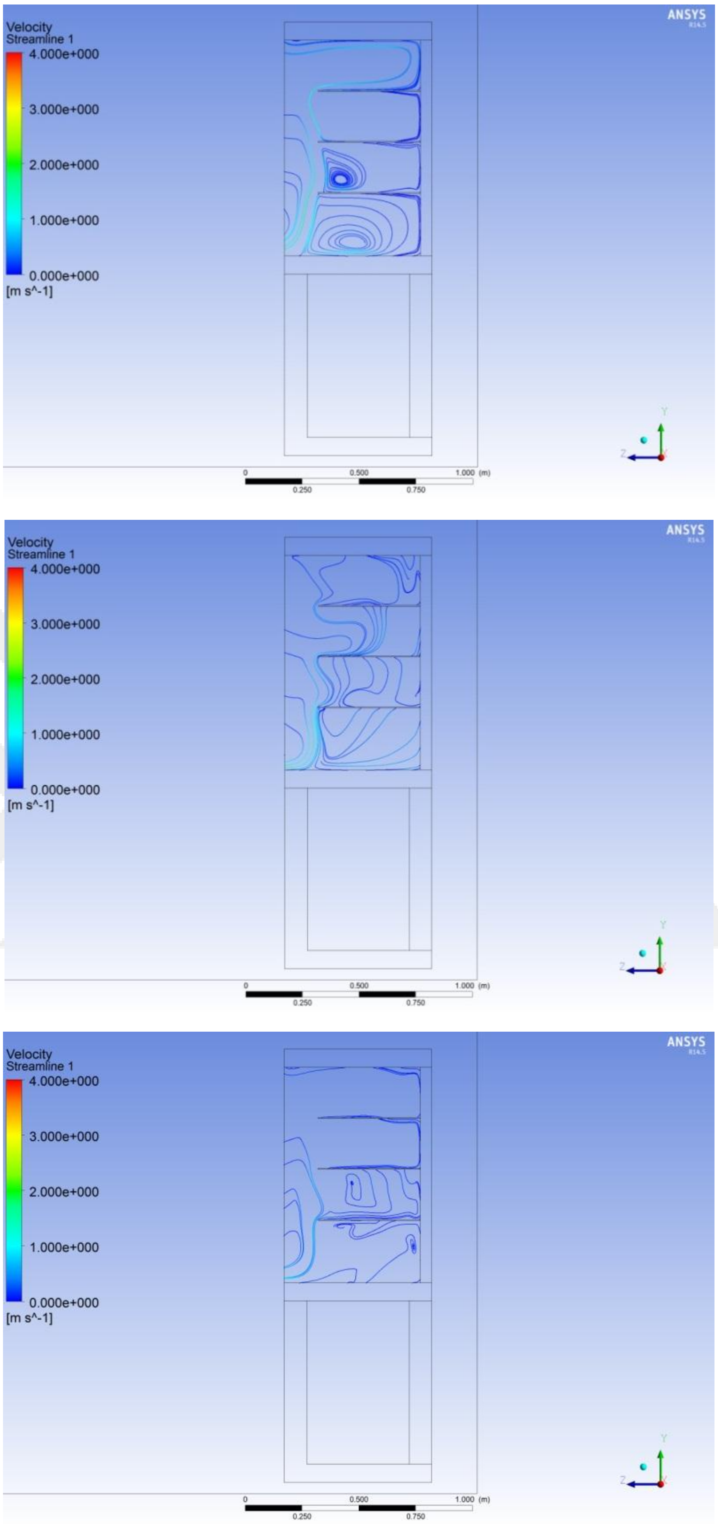


Figure 3.46 Streamline of the refrigerator and the opening on it at 3 shelves condition with 2^{0}C starting temperature at 1^{st}, 5^{th} and 15^{th} sec.

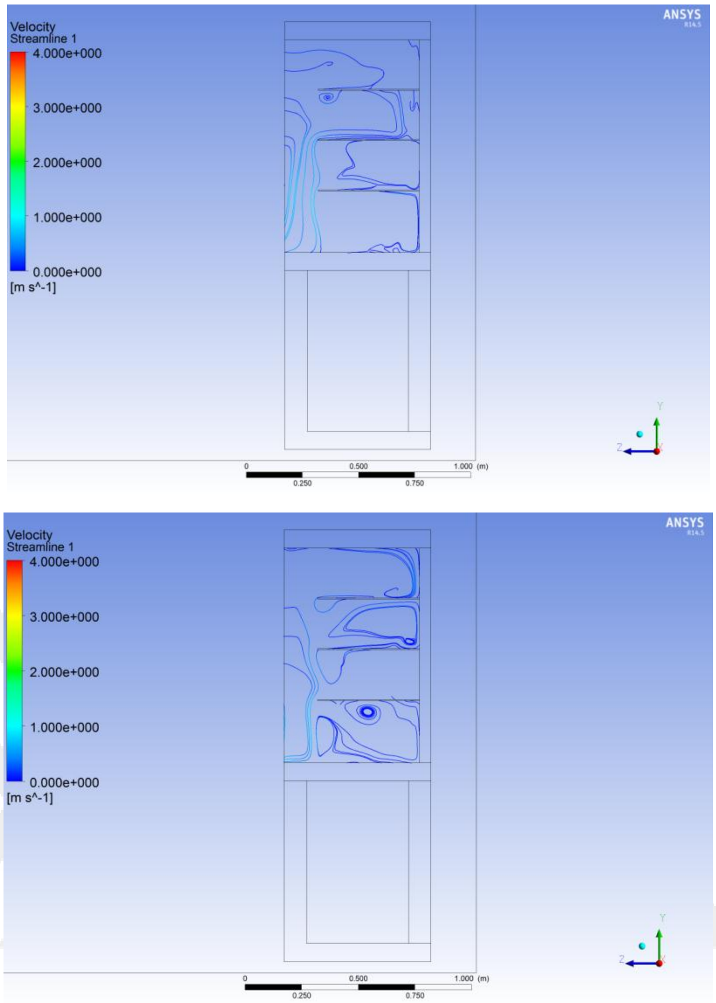


Figure 3.47 Streamline of the refrigerator and the opening on it at 3 shelves condition with2 ^{0}C starting temperature at 30th and 60th sec.

3.4.7 Results for 3 Shelves and Initial Temperature of Refrigerator of 4^{0}C

2^{0}C and 4^{0}C of the starting temperature did not make a big difference on the stream lines and the heat contour which were taken on the specific seconds (Figure 3.48-3.51). The only difference in the system is the volume is 2^{0}C higher than the start. As it has more energy, it gains less energy here.

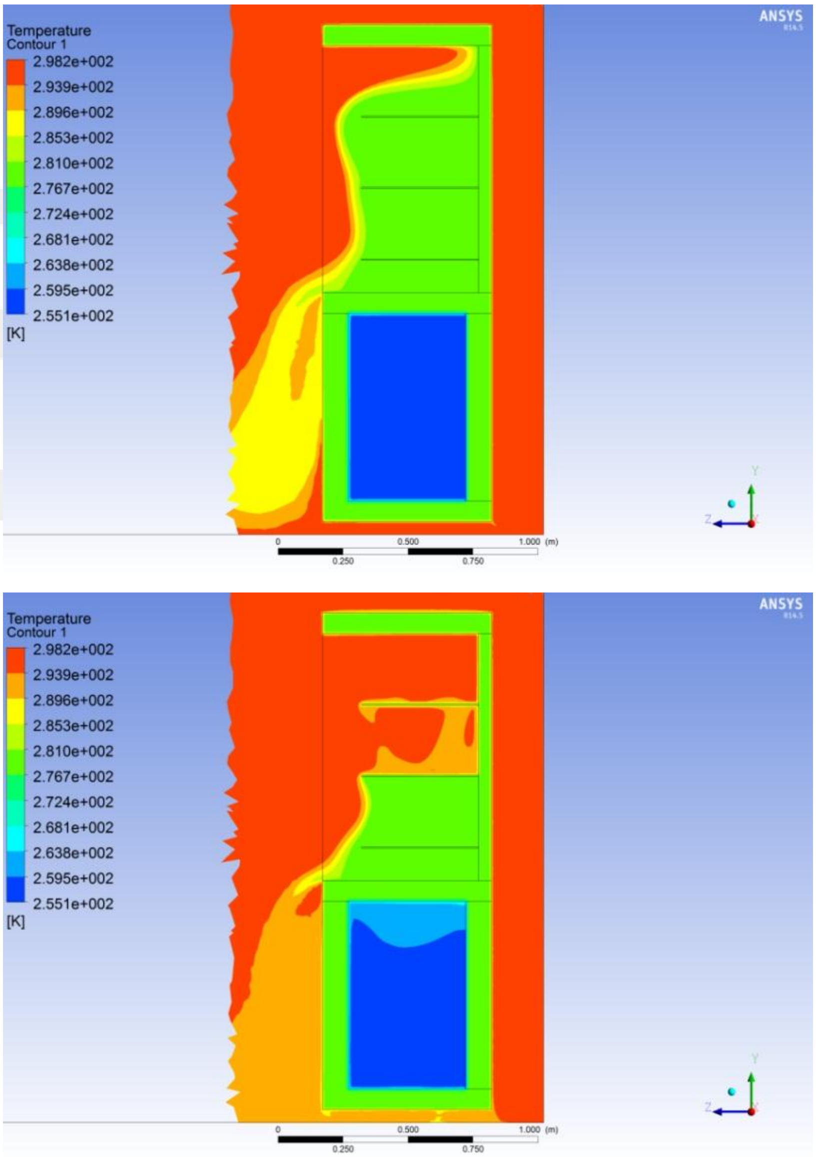


Figure 3.48 Temperature contours of the refrigerator and the opening on it at 3 shelves condition with 4^{0}C starting temperature at 1st, 5th and 15th sec.

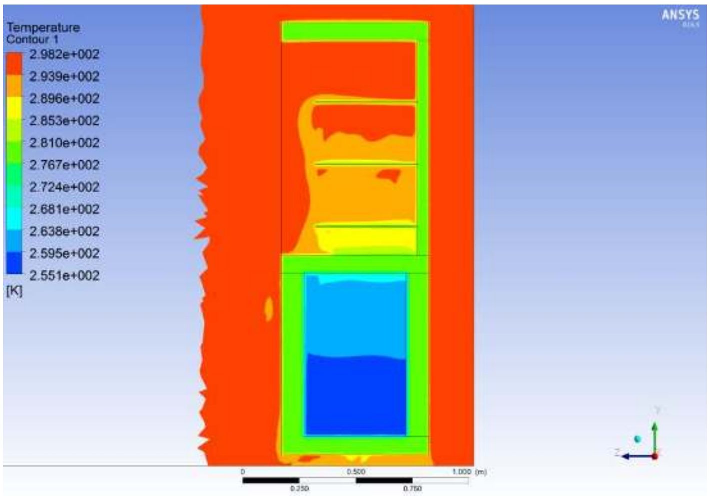


Figure 3.48 Temperature contours of the refrigerator and the opening on it at 3 shelves condition with 4^{0}C starting temperature at 1^{st}, 5^{th} and 15^{th} sec. (continue)

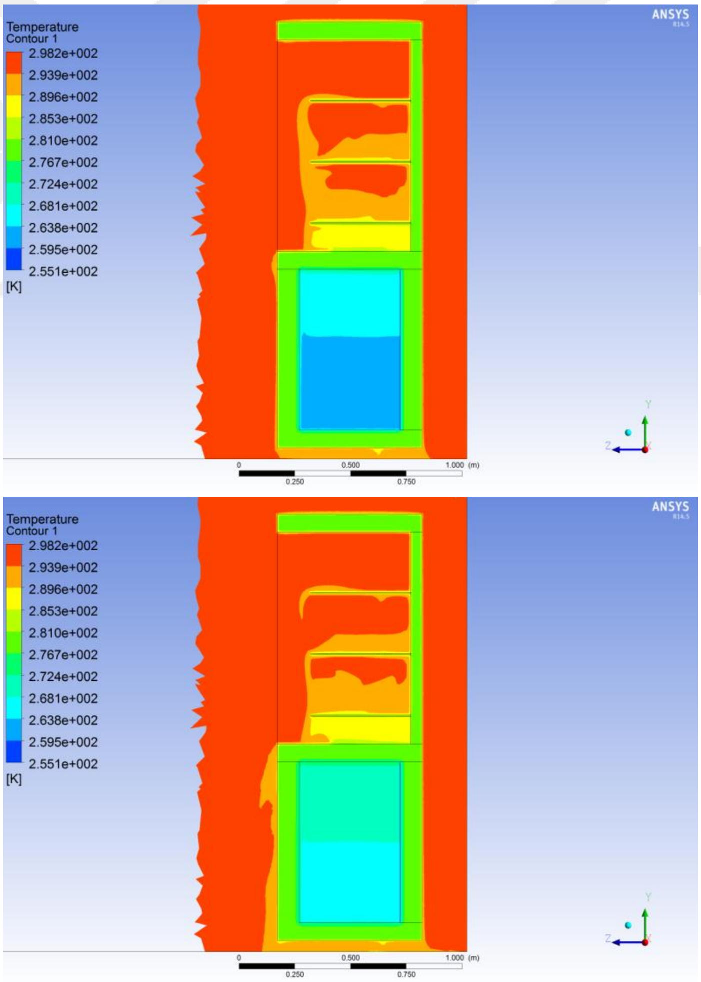


Figure 3.49 Temperature contours of the refrigerator and the opening on it at 3 shelves condition with 4^{0}C starting temperature at 30^{th} and 60^{th} sec.

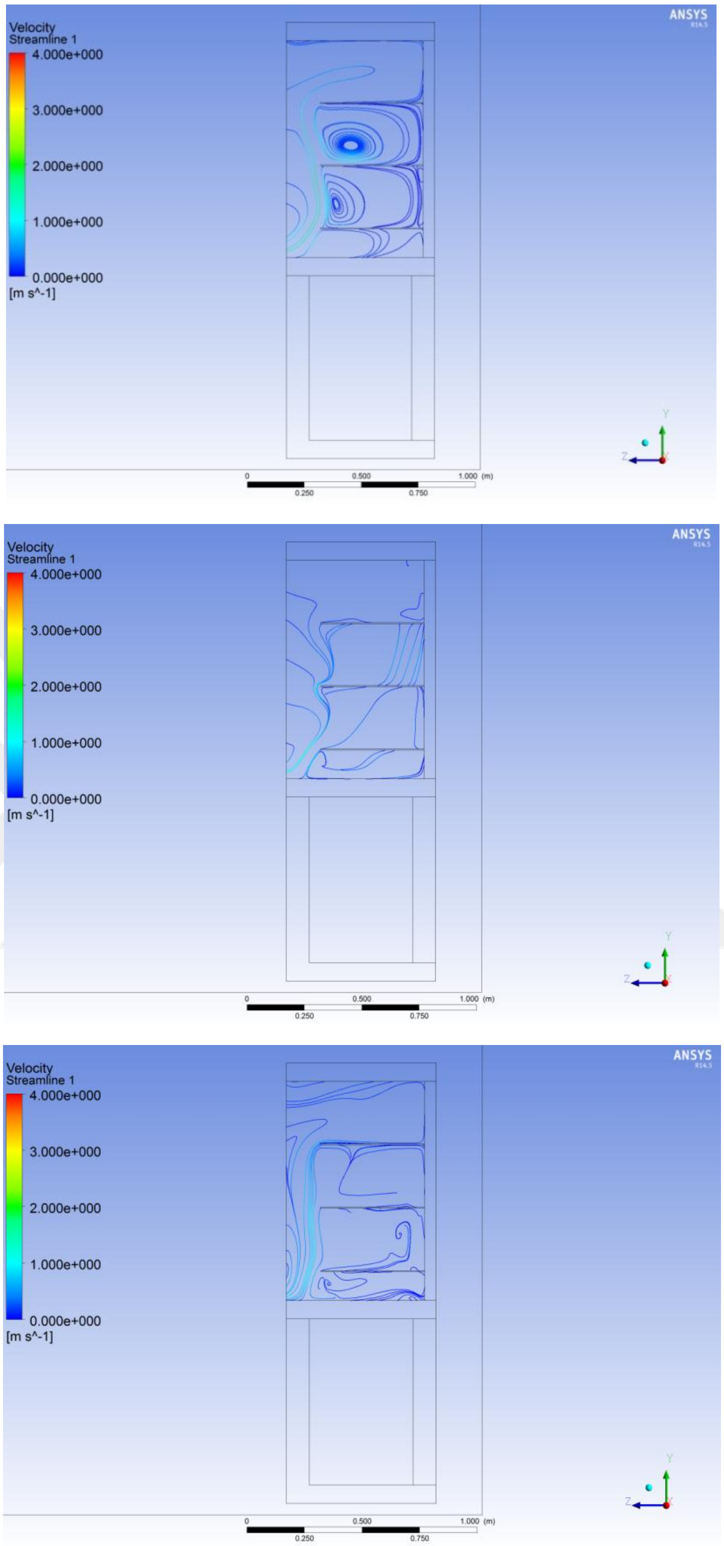


Figure 3.50 Streamline of the refrigerator and the opening on it at 3 shelves condition with 4^{0}C starting temperature at 1^{st}, 5^{th} and 15^{th} sec.

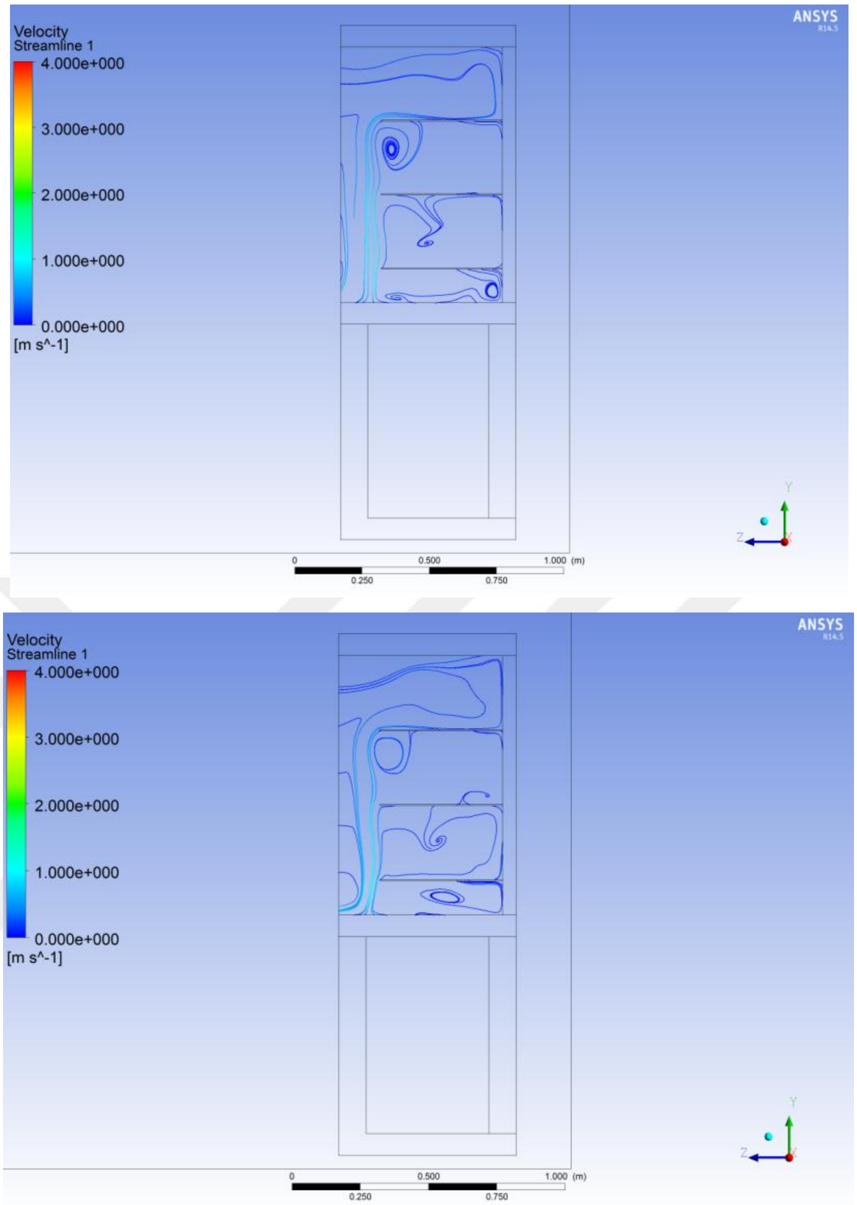


Figure 3.51 Streamline of the refrigerator and the opening on it at 3 shelves condition with4 ^{0}C starting temperature at 30th and 60th sec.

3.4.8 Results for 3 Shelves and Initial Temperature of Refrigerator of 4^0C for Different Shelf Heights

Because the starting temperature is 2^0C and 4^0C does not create a big difference, it can be seen the same results between 4 shelves 2^0C and 4 shelves 2^0C different heights differences (Figure 3.52-3.55). That supports the accuracy of the analysis.

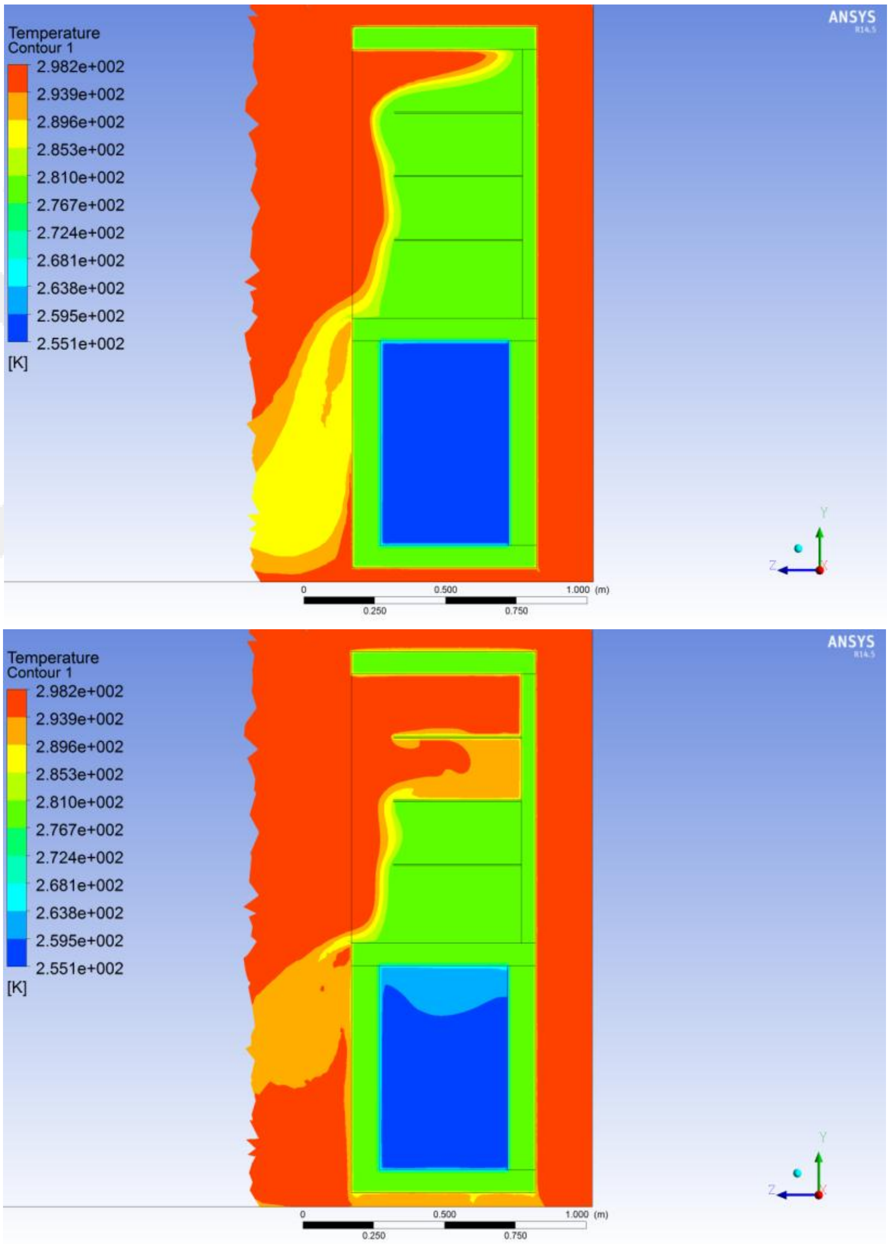


Figure 3.52 Temperature contours of the refrigerator and the opening on it at 3 shelves condition with 4^0C starting temperature at 1^{st}, 5^{th} and 15^{th} sec.

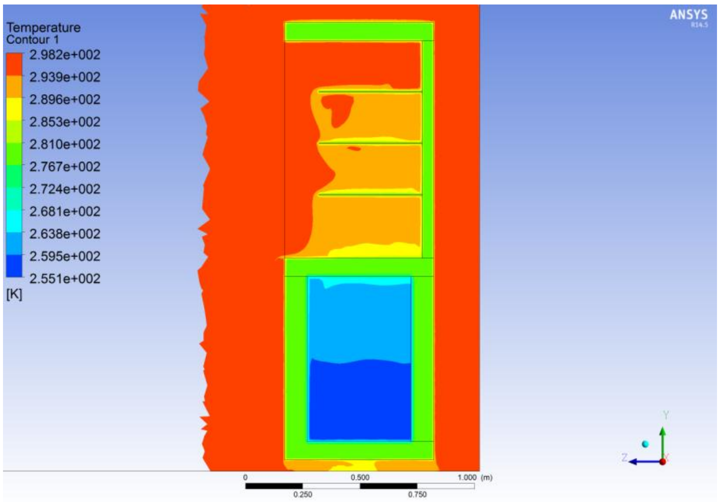


Figure 3.52 Temperature contours of the refrigerator and the opening on it at 3 shelves condition with 4⁰C starting temperature at 1st, 5th and 15th sec. (continue)

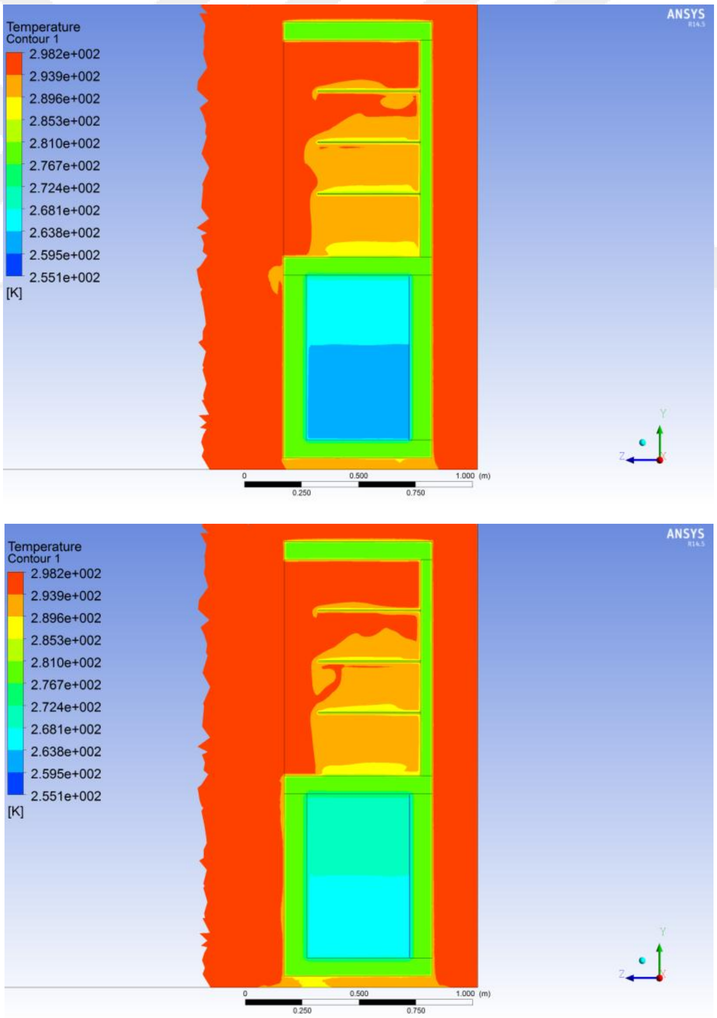


Figure 3.53 Temperature contours of the refrigerator and the opening on it at 3 shelves condition with 4⁰C starting temperature at 30th and 60th sec.

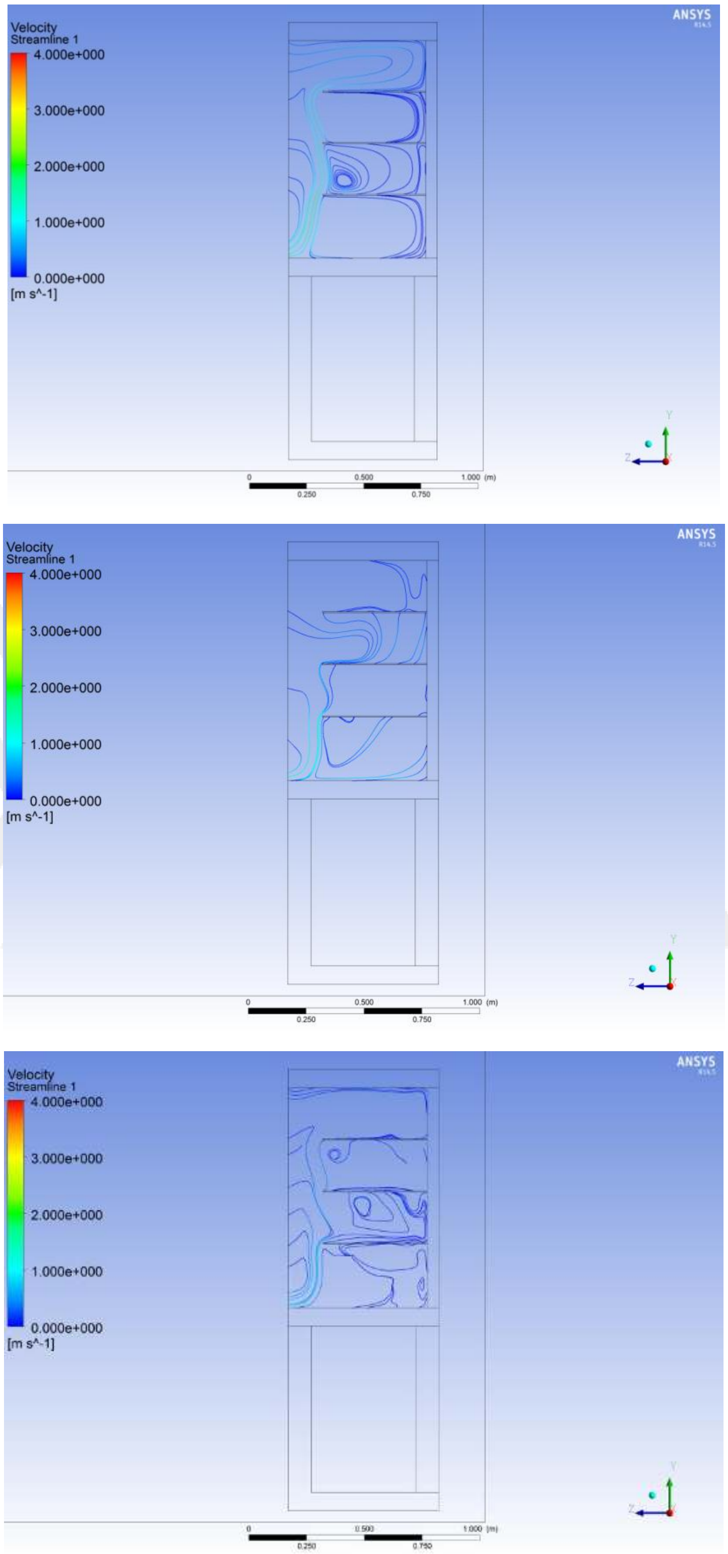


Figure 3.54 Streamline of the refrigerator and the opening on it at 3 shelves condition with 4^0C starting temperature at 1^{st}, 5^{th} and 15^{th} sec.

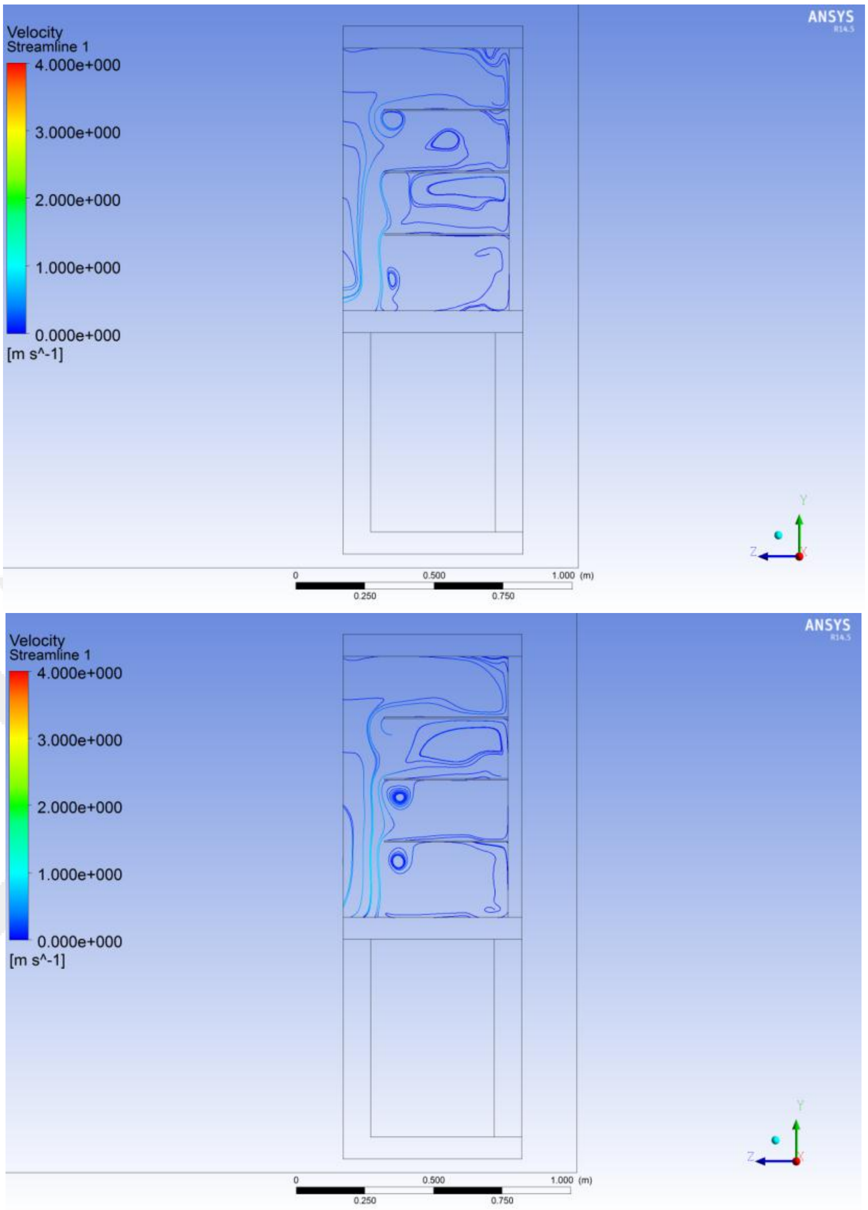


Figure 3.55 Streamline of the refrigerator and the opening on it at 3 shelves condition with4 ^{0}C starting temperature at 30th and 60th sec.

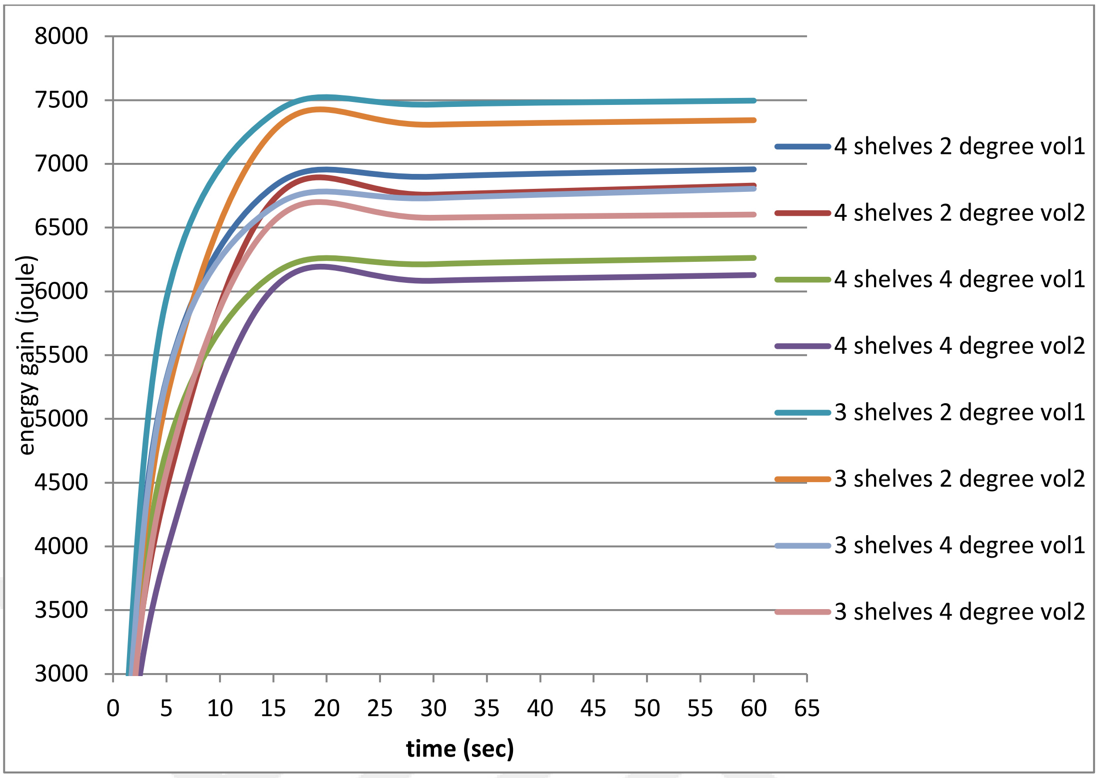


Figure 3.56 Energy earnings for different shelf and working temperature conditions

As it can be seen on the energy gain Figure 3.56, the highest energy gain is on the 3rd shelf 2^0C vol1 (because the high shelf is on the top, and the low shelf is on the bottom) the least gain is on the 4rd shelf 4^0C and vol2 (the high shelf is larger than the low bottom shelves). It can be seen there is less energy gain with more shelves and fridges working heat.

CHAPTER FOUR
CONCLUSIONS

There are lots of refrigerator manufacturer Companies in Turkey. Their largest markets, European countries, energy regulations are updated almost every year and acceptable energy consumption levels of refrigerators are reducing continuously. One of the most important factors on the energy efficiency of refrigerators is the heat and mass transfer takes place during opening and closing the refrigerator door. The starting point of this project is to calculate the heat gained in a short period of time the refrigerator door is open.

In the study conducted, literature research was carried out primarily. At the conclusion of literature research, the close and open volume analyses were done to affirm analyzing program and it was compared to the result which were seen before. After that, 2d and 3d refrigerators were designed and analyzed.

With the result of the current analysis, it was observed that the heat goes to the 1st shelf at first. After the heat on the 1st shelf was balanced, it was observed that the heat gain was through the top shelves to the bottom shelves. It was concluded that making the top shelf lower and making the lowest shelf higher, will decrease the energy gain of the refrigerator, when the doors open. Also it was seen that the increase of the number of the shelves decreased the energy gain.

It was seen that the top shelves are the most changing ones for the practical applications. It was also seen that it could have been dangerous for human health to use them for some daily food because of the heat change on the shelves.

This mathematical work can be a preparation for the future experiments. In the future, the air movements after opening the fridge door can be observed with using the PIV system, and can be compared to the mathematical results.

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APPENDIX

Nomenclature

g	Gravitational acceleration [m/s^2]
k	Thermal conductivity [W/mK]
L	Wall length [m]
Nu	Nusselt number $\left(=\frac{h.l}{k}\right)$
p	Pressure [Pa]
Pr	Prandtl Number $\left(=\frac{v}{\propto}\right)$
q	Rate of heat transfer [W]
Ra	Rayleigh Number $\left(=\frac{g\beta(T_h-T_c)L^3Pr}{v^2}\right)$
T	Temperature [K]
u,v,w	Velocity Components [m/s]
x,y,z	Cartesian axis directions [m]

Special characters

$\propto$	Thermal expansion coefficient [$k/\rho c_p$]
ε	Dimensionless temperature Differences $\left(=\frac{T_h-T_\infty}{T_\infty}\right)$
ΔT	Temperature differences [K,^{o}C]
ρ	Density [kg/m^3]
μ	Dynamic viscosity [kg/ms]
ϑ	Kinematic viscosity [m^2/s]