

**INVESTIGATION OF AIRFLOW AND TEMPERATURE DISTRIBUTION
IN A FREEZER CABINET**

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İSTANBUL TEKNİK ÜNİVERSİTESİ ★ FEN BİLİMLERİ ENSTİTÜSÜ

**EVLERDE KULLANILAN BİR NO-FROST BUZDOLABI DONDURUCU
BÖLMESİNDE HAVA AKIŞI VE SICAKLIK DAĞILIMININ
İNCELENMESİ**

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FOREWORD

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ABBREVIATIONS

CAD	: Computer Aided Design
CFD	: Computational Fluid Dynamics
DOE	: Design of Experiment
MP	: Measurement Packages
N-F	: No-Frost
PIVM	: Particle Image Velocimetry Measurements
TCRM	: Temperature Controlled Room Measurements

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INVESTIGATION OF AIRFLOW AND TEMPERATURE DISTRIBUTION IN A FREEZER CABINET

SUMMARY

Uniformity of temperature distribution in a loaded freezer cabinet is one of the most important factors affecting energy consumption of a refrigerator. This study focuses on the airflow behavior and the temperature distribution inside the freezer compartment of a domestic no-frost refrigerator. Energy consumption increases in a freezer cabinet if the temperature difference between the warmest and coldest load packages is high. The objective is to keep food fresh by providing a uniform temperature distribution and concurrently to reduce the energy consumption. The 3D transient CFD analyses have been performed to model the heat transfer by forced convection during the run-time and by natural convection during off-time period with an appropriate loading plan. The initial and boundary conditions were provided from temperature controlled room and 3D PIV measurements. After the validation of the CFD analyses results, a design of experiment (DOE) study is performed to find the optimum condition for the temperature uniformity inside the refrigerator cabinet. After an optimum design is obtained by DOE study, a prototype of the design is produced and tested with PIV measurements. The CFD analyses obtained are shown to be in a reasonable accuracy by experiments. Then the prototype is tested by temperature controlled room measurements and the desired temperature uniformity inside freezer cabinet is obtained. Furthermore, the effects of total flow rate and temperature of the air flow sent into the freezer cabinet were investigated.

EVLERDE KULLANILAN BİR NO-FROST BUZDOLABI DONDURUCU BÖLMESİNDE HAVA AKIŞI VE SICAKLIK DAĞILIMININ İNCELENMESİ

ÖZET

Bir buzdolabının enerji tüketimini belirleyen en önemli faktörlerden birisi ürünlerle doldurulmuş dondurucu bölümündeki uniform sıcaklık dağılımıdır. Bu çalışmada, evsel bir no-frost buzdolabının dondurucu bölümü içerisindeki hava akışının ve sıcaklık dağılımının incelenmesine odaklanılmıştır. Dondurucu bölümü içinde en sıcak paketin sıcaklığı ile en soğuk paketin sıcaklığı arasındaki farkın artması enerji tüketimini arttırmaktadır. Bir örnek sıcaklık dağılımının sağlanması ile hem yiyeceklerin raf ömrünün uzatılması hem de enerji tüketiminin azaltılması hedefleri gerçekleştirilebilecektir. Uygun bir paket yükleme planı dahilinde zorlanmış taşınım ile ısı geçişinin hakim olduğu kompresörün çalışma süresini ve doğal taşınım ile ısı geçişinin hakim olduğu kompresör durma sürelerini temsil etmek üzere zamana bağlı 3 boyutlu CFD analizleri gerçekleştirilmiştir. Başlangıç ve sınır şartları orijinal buzdolabı için yapılmış 3D PIV ölçümleri ve ısı odası deney sonuçlarından elde edilmiştir. CFD analiz sonuçlarının deneysel veriler ile doğrulanmasının ardından dondurucu bölümü içerisinde en iyi sıcaklık dağılımını sağlayacak durumun belirlenebilmesi için CFD tabanlı deneysel tasarım çalışması yürütülmüştür. Bu çalışma sonunda elde edilen en iyi durum için prototip imal edilmiş ve üfleme deliklerinden çıkan debi değerlerinin tespiti için PIV deneylerine tabi tutulmuştur. PIV ölçümleri ile CFD analiz sonuçlarının büyük bir uyum içinde olduğu görülmüştür. Bu tespitten sonra prototip, paket sıcaklık ölçümlerinin yapılacağı ısı odası deneyine tabi tutulmuştur. Deney sonucunda istenilen uniform sıcaklık dağılımı elde edilmiştir. Ayrıca, toplam debinin ve üfleme sıcaklığının soğutma süresine olan etkilerinin belirlenmesine yönelik bir çalışma da ele alınmıştır.

1. INTRODUCTION

Nowadays, cost of available energy increased because of reduced energy resources as a result of which energy efficiency has become more significant in science, technology and engineering applications. The companies which are in energy sector have to improve new projects to reach the targeted low production costs and energy consumption values. The utilization of no-frost refrigerators have increased nearly in all industrialized countries for the last 25 years. The primary purpose of refrigeration is to decrease the temperature of the enclosed space or substance and then maintain that lower temperature. The design of a no-frost refrigerator includes criteria such as reducing the overall energy consumption, decreasing run-time of the compressor, lasting freshness of food, while satisfying the demands for an environmentally-friendly and quiet appliance. One of the parameters which affect the overall energy consumption dominantly is the refrigerator's capability of providing a homogeneous air circulation in the cabinet. In a no-frost refrigerator, air is sucked from the freezer compartment and sent over to evaporator to be cooled. The vapor-compression cycle is used in most household refrigerators as well as in many large commercial and industrial refrigeration systems. The cold liquid-vapor mixture inside evaporator travels through the evaporator coil and is completely vaporized by cooling the warm air (from the space being refrigerated) being blown by a fan across the evaporator coil. The cooled air is blown back to freezer and fresh-food compartment through air supply channels by a fan. Utilization of a single or multi-fan is an important means for the homogenization of the air circulation inside the cabinet. The homogenization of the air temperature inside the cabinet also increases fresh-life quality of the food on the shelf. If homogenization can not be succeeded, shelves will have different temperatures and local cavities between the packages will occur and run-time of the compressor will increase. Thus, great interest has focused on homogenization of cabinet air circulation to improve efficiency of the refrigeration. Some of the work focused on design and analysis of air management inside the cabinet by computational fluid dynamics (CFD) methods. Generally, in CFD analyses, the steady state solutions were investigated. However, steady state

solutions do not reflect the real temperature difference that occur in the local (cavity) regions inside the freezer cabinet. These solutions do not simulate the experimental measurements very well.

In this work, the objective is to keep food fresh by providing a uniform temperature distribution and concurrently to reduce the energy consumption. Parameters should be known to optimize the refrigeration system. Several experiments should be performed to determine the effect of each parameter. But this process causes time consuming and increased costs. CFD analyses are important tools to shorten the design period and decrease the cost of the experiments. Hence, transient CFD analyses have been performed for the freezer compartment of a specified no-frost refrigerator and results are compared with the experimental measurements. After the reliability of the CFD simulations are provided, the design process to optimize the refrigeration system started. The parametric study to optimize the refrigeration system had been performed to determine the position of the air blowing holes and the flow rates at each hole. The optimum design is obtained by DOE study. A prototype of the new design is produced and tested with PIV and temperature controlled room measurements. The CFD analyses obtained are shown to be in a reasonable accuracy with experiments.

2. LITERATURE REVIEW

In literature, several studies are carried out to supply a uniform temperature distribution inside the refrigerator cabinets. Essentially, to get a uniform temperature distribution inside refrigerator cabinets is an ideal situation. In reality, there could be temperature differences in some regions. For instance, because of the evaporator effect the air becomes colder and it is known that the air sinks when it becomes colder. Consequently, the lowest temperatures occur at the bottom part of the back wall in conventional refrigerators. On the other hand, the air expands and ascends when it becomes warmer. The highest temperatures occur at the door side and upper part of the refrigerator if the distance to the cold air channels taken into account. In the absence of air supply system, there could be high temperature differences such as 7 – 8 °C.

The studies in literature are aimed that to get a minimized temperature difference in refrigerator cabinets. Some of the studies which have both experiments and commercial CFD codes were investigated in details. Result of these studies indicate that to get a uniform temperature distribution the supplementary air systems, the position of air blowing holes, the mass flow rate and the temperature of the flow which is blowing inside the cabinet should be investigated.

Kazuhiro Fukuyo et al [1] investigated thermal uniformity and a high cooling rate by improving the air-supply system in the fresh food cabinet. The aim is to make the upper side of the refrigerator colder. To realize this aim, it is decided to design an air channel at the upper side of the refrigerator and inject cold air inside from this channel. The blower helps to supply a uniform temperature distribution by making air circulation. By that successful design, it is achieved that the biggest part of the refrigerator has the temperature value of 2°C. In this study, a fan and a jet channel had been added to the conventional air supply system to investigate the temperature uniformity and the speed of cooling. To examine the temperature distribution inside cabinets a commercial CFD code had been used. The jet cooling process had been

analysed by using mathematical model which was derived from theoretical and empirical equations.

The different configurations were given in Figure 2.1. There are 10 air supply holes and 6 grills on the back wall. The fresh-food cabinet volume is 0.24 m^3 . The temperature of the blown air is -10°C and the volumetric flow rate is $0.0028 \text{ m}^3/\text{s}$. The air turns back to the evaporator from the grills. The temperature inside the fresh-food cabinet is aimed to be 5°C . The system A has a conventional air supply system. In this configuration, high temperature values were observed at the top and door shelves. Then, the conventional air supply system had been improved by adding air supply holes to the top of the cabinet, named system B. The last configuration is called as system C. In this configuration, a fan, three jet air channels and two grills had been added to the system A. The grills absorb the air from the cabinet and blows directly (does not go into the evaporator) from the jet air channels inside the cabinet by the fan. The volumetric flow rate at the jet air channels is $0.005 \text{ m}^3/\text{s}$. The velocity of the air at jet air holes is 10 times greater than the air supply holes.

The finite volume method had been driven. The k-epsilon turbulence model and the SIMPLE algorithm had been used. For the validation of the results, the CFD analysis results had been compared with the steady state experimental measurements. The ambient temperature had been assigned as 30°C . The heat transfer coefficient for the insulated wall and the convective heat transfer coefficient at the outer wall surfaces had been assigned as 0.018 W/mK and $5.0 \text{ W/m}^2\text{K}$, respectively.

The temperature difference between the prediction and the measurement was resulted as 1K . This result had been accepted for the validation of the CFD model.

The results for the system A, B and C were given in Figure 2.2. For system A, high temperature regions had been observed at top and door shelves of the cabinet. The bottom and the mid-regions of the shelves had low temperature values. For system B, by the help of the air channel which was located to the top of the cabinet the high temperature regions observed at system A had been averted. For system C, the biggest part of the cabinet has a temperature value of 2°C by aid of the jet air channels.

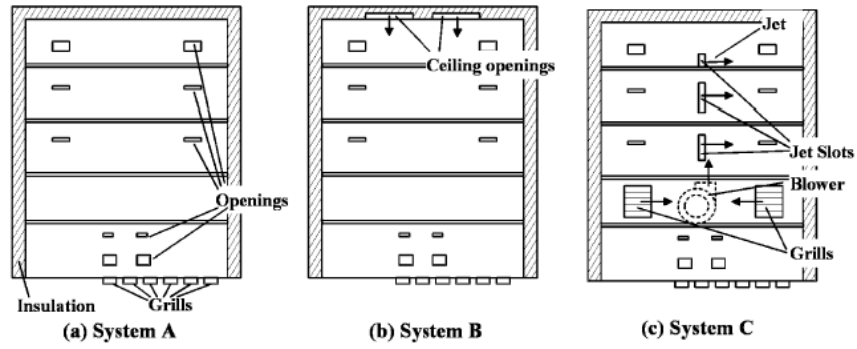


Figure 2.1: Examined configurations [1]

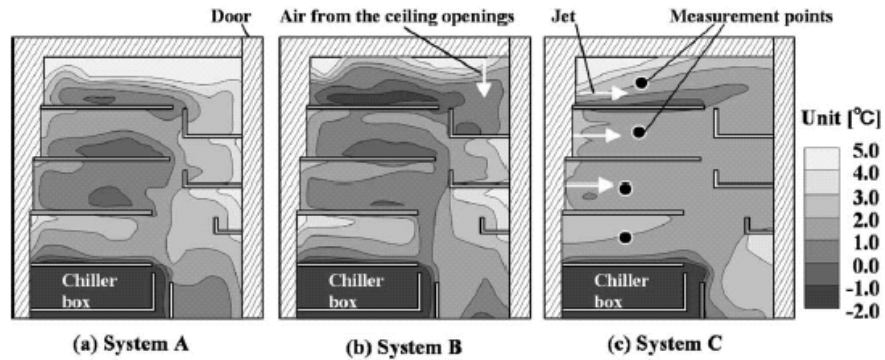


Figure 2.2: Temperature distribution at mid-cross section of the cabinet [1]

The lowest standard deviation was observed at system C. The system A has a standard deviation which is approximately two times greater than sytem C. This result indicates that the temperature uniformity was supplied by means of jet air channels. Rapid cooling process was modeled as 2D heat transfer problem of a cylindirical can which is cooled down by jet. For this purpose, previous studies in literature were investigated and obtained mathematical models were used. These mathematical models were validated by an experimental contrivance shown in Figure 2.3.

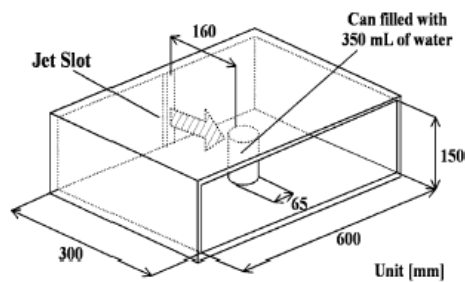


Figure 2.3: The scheme for the rapid cooling experiment [1]

The experimental results demonstrating that the water filled can cools from 70°C to 40°C in 2400 seconds, on the other hand the mathematical model resulted as 2700 seconds. There is a 9% relative error between experiments and mathematical model. The reason of the error is thought to be the 2D mathematical model. The jet velocity versus cooling time graph was given in Figure 2.4. From the figure, it is obvious that the cooling time is highly effected by the jet velocity at the interval of 0.5 m/s and 1 m/s. The cooling time is also decreasing for the greater values of jet velocity but the gradient is small. As a result, the jet circulates the air inside a fresh food cabinet at high velocity and improves the thermal uniformity. The jet also increases the cooling rate.

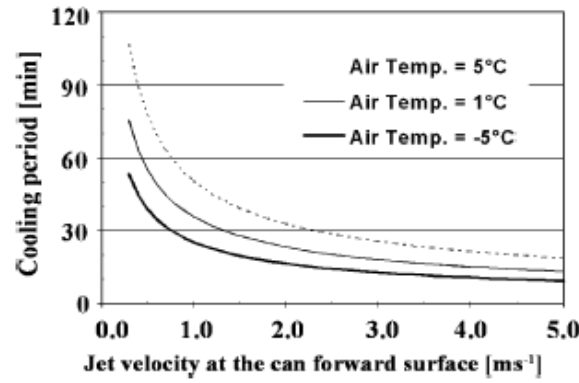


Figure 2.4: Jet velocity versus cooling period [1]

O. Laguerre et al [2] studied the heat transfer sourced from natural convection for the refrigerators that do not have ventilation systems. Several experimental and CFD analysis have been performed for three configurations such as empty cabinet, cabinet fitted with glass shelves and loaded cabinet with products. It is obtained that the CFD analysis results are too close to experimental datas when the effect of radiation is taken into account. The discrete ordinate method were used for the coupled analysis of the convection and radiation. The laminar airflow and a constant evaporator temperature were assumed in the CFD analysis.

In this study, it is aimed to find the hot and cold places in refrigerator cabinet. To realize this, the flow and the heat transfer characteristics had been investigated inside refrigerator.

The thickness of the glass shelves is 5 mm and the heat transfer coefficient 0.75 W/mK. The material of the products is methyl cellulose and its heat transfer coefficient is 0.5 W/mK. The laminar flow assumption had been made. The boundary

conditions had been assigned from the experimental results. The convective heat transfer coefficient at the outer wall surfaces had been assigned as $10 \text{ W/m}^2\text{K}$. The ambient temperature was chosen as 20°C . A constant evaporator temperature was assumed to be 0.5°C . The half volume of the refrigerator had been modeled because of the symmetry. The structured mesh performed for the domain. The model had been solved by Fluent program with finite volume method. The discrete ordinate method was used for the coupled analysis of the convection and radiation. The temperature at the bottom of the refrigerator is approximately 3°C and at the top of the refrigerator the temperature is approximately 7°C excluding the vegetable box. It is seen that the cold regions were effected by the shelves and products when comparing the results of the empty and loaded refrigerator. The warmer temperature values are seen at the top of the refrigerator. In addition to this, at the back wall of the refrigerator the cold regions had been observed because of the evaporator effect. The results show that the products which are close to back wall colder than the products which are close to door. The temperature of the vegetable box had been observed as approximately 8°C in all configurations.

Air flows downwards along the evaporator while its velocity increases along the course to attain a maximum value at the bottom of the refrigerator. Air flows upwards along the door and the side walls of the refrigerator while its velocity decreases progressively and becomes stagnant at the top of the refrigerator. The different configurations such as empty refrigerator (a), refrigerator with glass shelves (b), refrigerator with glass shelves and products (c) were given in Figure 2.5. Figure 2.6 and 2.7 present the temperature and velocity contours on symmetry plane of the empty refrigerator (a), symmetry plane of the empty refrigerator with glass shelves (b), symmetry plane of the empty refrigerator with glass shelves and products (c), respectively.

When radiation was taken into consideration in CFD analysis, the predicted air temperatures were in good agreement with the experimental values. However, when radiation was not taken into account, the temperature was found a little bit higher at the top of the refrigerator.

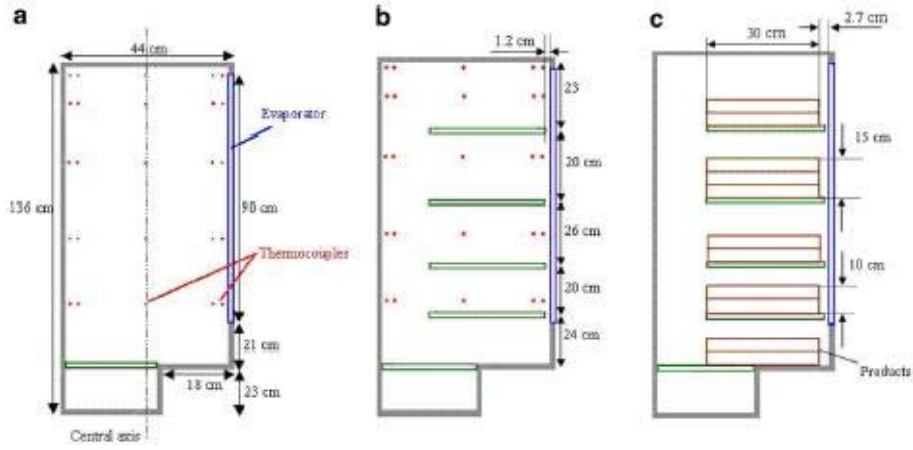


Figure 2.5: Different configurations for examined refrigerator [2]

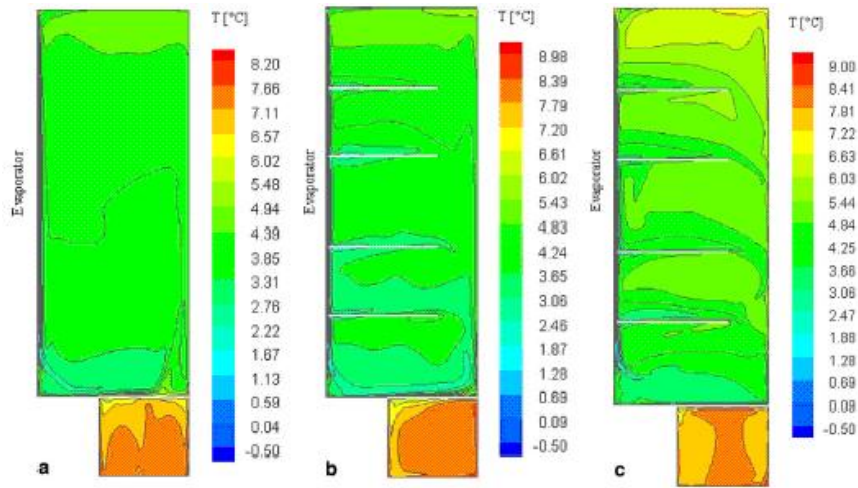


Figure 2.6: Temperature contours [2]

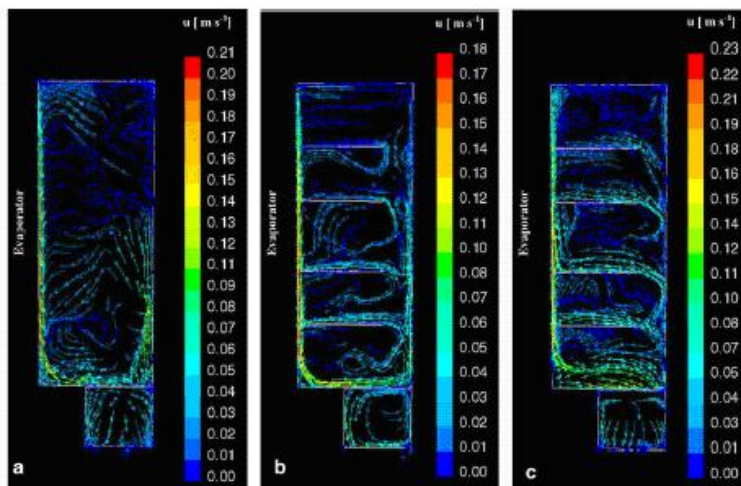


Figure 2.7: Velocity contours [2]

Guo-Liang Ding et al [3] studied to improve thermal uniformity inside of two different refrigerators. It is thought that the distance between the shelves and evaporator and the distance between shelves and refrigerator's door are the other factors that effects the temperature distribution. The authors have studied on temperature field and airflow field inside a cooling chamber in order to improve temperature uniformity inside directly cooled refrigerators. It is found that temperature variation will decline and more uniform temperature distributions will be achieved if reducing that distances, but at the same time heat convection of air is weakened. As an improvement, a new system is suggested which includes an air duct and a blower. However, mathematical details of the model and effects of operating conditions on the refrigerator performance have not been discussed in their study.

The cabinet is 1012 mm in height and 414 mm in width. The evaporator was located to the back wall of the refrigerator. The thickness of all glass shelves is 4 mm. In Figure 2.8, D_e , the space between evaporator and glass shelves, is 10 mm. D_l , the space between door and shelves, is 6 mm. Θ_1 and Θ_4 are the temperature values of the points at the top and bottom of the refrigerator, respectively. Θ_2 and Θ_3 are the temperature values of the points located from the top and bottom with the distance of 5% of overall-height, respectively. T_1 and T_2 are the temperature values of the points located from the top and bottom with the distance of 1/3 of overall-height, respectively. These six points are assumed to be at the midpoint between the door and back wall. The schematic of cabinets inner configurations is given in Figure 2.8.

When D_e and D_l distances in Figure 2.8 are small, the air flows slowly in the space of the adjacent shelves. Therefore, the thickness of the fluid boundary layer increases. Consequently, the heat transfer coefficient between air and solid surface decreases. If D_e and D_l increase, the result is just the reverse. In this case, more air flows along the surfaces of the evaporator and the inner surfaces, heat transfer is enhanced with the boundary thickness growing thinner, and it is more difficult to obtain temperature uniformity.

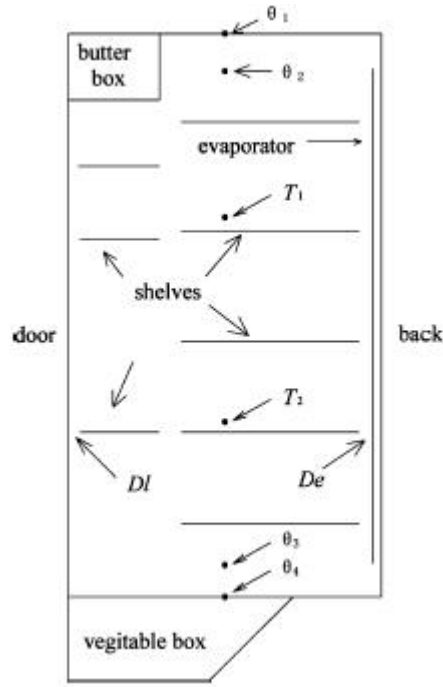


Figure 2.8: Side view of the refrigerator cabinet [3]

Gupta et al [4] mentioned that the main objective of a refrigerator is to keep the quality of food products. The quality of food products directly depends on temperature and air distribution inside the refrigerator. For this purpose, a comprehensive thermo-fluidic model for a domestic frost-free refrigerator was developed. The governing equations were solved by employing a conservative control volume formulation for a three-dimensional unstructured mesh.

In reality there is a continuous on and off cycling for compressor. The analysis had been done for a steady state case by assuming a continuous compressor work. Boussinesq assumption was employed for flow modeling inside the refrigerating compartment, which is governed by mixed convection whereas buoyancy effects are neglected for the freezer component, because of strong inertial effects. Viscous dissipation terms in the energy equation had been neglected. Fluid flow was taken to be incompressible. The flow was assumed to be laminar in both the compartments. Radiation heat transfer within the refrigerator was not considered. The refrigerator was analyzed for unloaded condition. The condenser and evaporator coils were considered as isothermal walls, because of the nearly isothermal phase change processes associated with these components. Heat transfer between freezer and fresh food compartments is neglected. Uniform velocity and temperature profiles are assumed at the inlet.

Figure 2.9 shows the variation of temperature along the central longitudinal axis, as one moves from one lateral side to the other. From the figure, the variations along this direction are negligible except near the wall.

The computational and experimental results are in a good agreement, though there is a comprehensible deviance. For the freezer compartment, the computationally predicted temperatures are higher than the experimental values. On the other hand, for the refrigerating compartment, the computationally predicted temperatures are lower than the experimental ones. This situation can be explained by the heat leakage through the door gaskets which was not considered in the computational model. Also the temperature behind the back wall is considered to be uniform, which in reality, varies from de-superheating to sub-cooling temperatures.

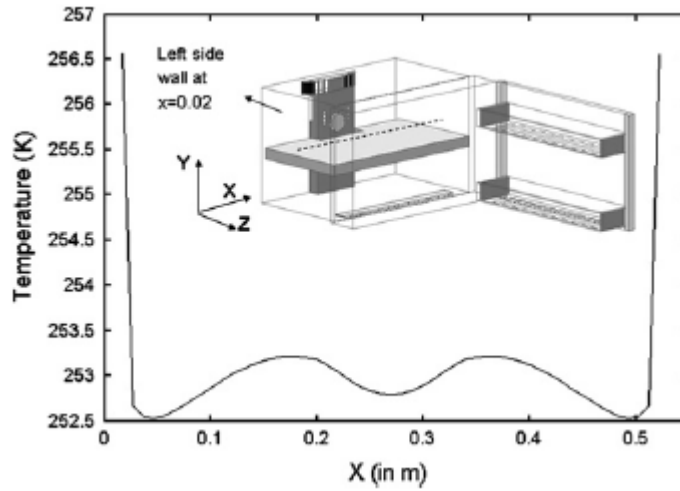


Figure 2.9: Variation of temperature in freezer cabinet [4]

Cho et al [5] investigated the flow inside the refrigerator by experimental and numerical analysis. Two-dimensional PIV measurements were carried out at the shelf regions and door basket regions inside the refrigeration compartment. The commercial software FLUENT was used for the numerical analysis of the conventional duct system and the new duct system of the refrigerator in order to analyze the velocity and temperature distribution inside the whole refrigeration compartment. The new duct system exhibits an improved uniform cooling performance of the refrigerator. The 2-head Nd:YAG laser (max. 300 mJ/7ns pulse), light sheet forming optics, a laser arm to direct the light source, a pulse synchronizer, and a 1K×1K CCD camera (30 frames/sec) was used in the PIV system. Glycerin

was used as lubricating oil with an atomizer to supply the seeding particles. Field-of-views of 120mm by 120mm were used to capture 50 particle images and calculate the mean velocity field.

Measurement of air flow in a freezer compartment under real operating conditions was carried out by Lacerda, Melo, Barbosa, and Duarte [6] using PIV (particle image velocimetry). Insulated windows which enabled clear visualization of the flow field under real operating conditions were equipped to perform the tests. It was observed that the flow field was strongly influenced by the temperature variations due to the “on” and “off” operation cycles of compressor. This behavior was attributed to natural convection and the physical properties (viscosity) of air, which strongly depend on the temperature.

Another study on air flow in a ventilated domestic freezing compartment was carried out by Lee, Baek, Chung, and Rhee [7]. In this study, comparison of the velocity field obtained by CFD simulation and by PIV measurement. Jet-like flow around entrance ports, impinging and stagnation flow on the walls and large recirculation flow in the cavity was observed.

3. GENERAL DEFINITIONS AND CONCEPTS

In this section, basic informations and definitions which are used in this thesis work will be explained. These definitions are frequently used concepts about refrigerators.

3.1 No-frost Refrigerators

A high fraction of the energy consumption of a refrigerator is consumed by the freezer compartment since freezer cabinet is a volume which should be colder than the other volumes of the refrigerator. Having uniform temperature distribution in the refrigerator is important in terms of energy efficiency. Improvement of energy consumption in freezer cabinet is a focus of research both in academia and industry and has become topic of many patents. The cooling process in a no frost refrigerator is maintained by means of an air supply system which circulates dry and cold air in the refrigerator cabinets. Air supply system consists of evaporator, condenser, compressor, fan and air channels.

Evaporator and condenser are the most basic elements of the refrigerator. The refrigerant circulates through evaporator and condenser channels in a thermodynamic cycle which is shown in Figure 3.1. In this cycle, a circulating refrigerant enters the compressor as a vapor. From point 1 to point 2, the vapor is compressed at constant entropy and exits the compressor superheated. From point 2 to point 3 and on to point 4, the superheated vapor travels through the condenser which first cools and removes the superheat and then condenses the vapor into a liquid by releasing heat at constant pressure and temperature. Between points 4 and 5, the liquid refrigerant goes through the expansion valve where its pressure suddenly decreases, causing abrupt evaporation.

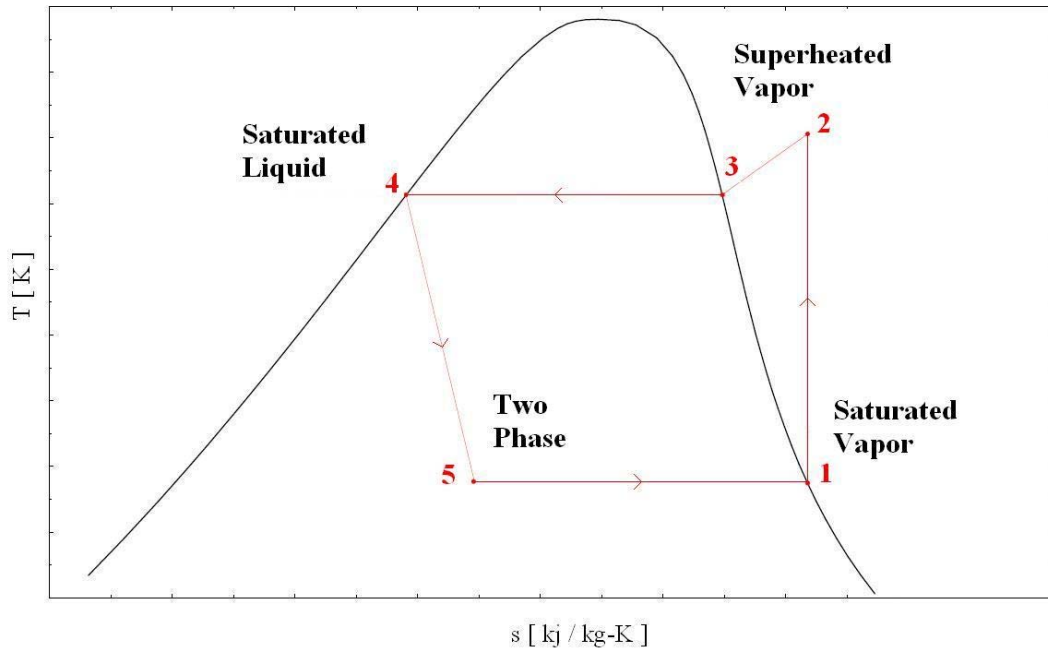


Figure 3.1: Temperature-Entropy Diagram

That results in a mixture of liquid and vapor at a lower temperature and pressure as shown at point 5. The cold liquid-vapor mixture then travels through the evaporator coil and is completely vaporized by cooling the surrounding warm air (from the space being refrigerated) which is blown by a fan across the evaporator coil. The resulting refrigerant vapor returns to the compressor inlet at point 1 to complete the thermodynamic cycle.

The fan is the other significant part of the air supply system. Fan provides the fluid flow over the evaporator and sends the cold air inside refrigerator compartments. Another significant part of air supply system is air channels. The air in the air channels flows due to the pressure difference which is created by a fan. Air supply channels send the cold air which is forced by fan to the air blowing holes and then manage the air flow by pulling the warm air in the cabinet and sending it again to evaporator. A sample configuration is given for no-frost refrigerator in Figure 3.2.

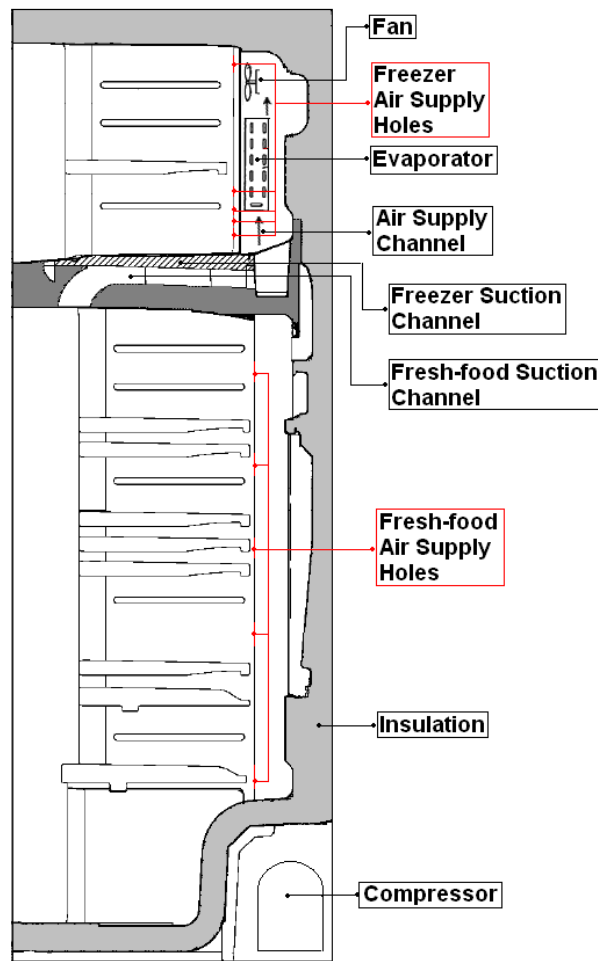


Figure 3.2: A Sample configuration for no-frost refrigerator

After a while, icing is observed in the freezer cabinet of the conventional refrigerators and the icing in the freezer cabinet needs to be defrosted. This brings a difficulty in practical use. Additionally, the useable volume of the freezer cabinet decreases because of icing. The cooling process of the conventional refrigerators is managed by cooling of freezer cabinet walls. The humid air jumps to the liquid phase when it meets the cold surface, then icing is observed on the wall surfaces if the ambient temperature is below 0 °C.

In no-frost refrigerators, the humid air is circulated over the evaporator fins and tubes to be cooled and then the cold, dry air is sent to the freezer cabinet. Consequently, icing is not observed inside the no-frost refrigerators freezer cabinet since all the icing forms on the evaporator surfaces. In order to defrost the evaporator surfaces a heater which is located on the evaporator is used. The icing on the evaporator can be seen in Figure 3.3.



Figure 3.3: Icing on the evaporator surfaces

3.2 Experimental Measurements

3.2.1 Temperature Controlled Room Measurements

The experimental measurements are performed in temperature-controlled rooms for the conditions that CFD analyses will be performed for the refrigerator. The ambient temperature is kept at 25 °C. Instead of real food, packages are used in the freezer to measure the temperature difference due to air circulation. The placement of the packages inside the freezer are done according to TS-EN-ISO-15502 (Turkish standard household refrigerating appliances-characteristics and test methods). The temperature readings are done by the thermocouples placed in the center of packages. In Figure 3.4 the package loading plan are shown and the thermocouples are shown with numbers on the packages. The packages have properties like meat products and these properties will be listed in CFD analyses section. In the temperature controlled room the temperature distribution in the freezer compartment and the overall energy consumption of the refrigerator is determined.

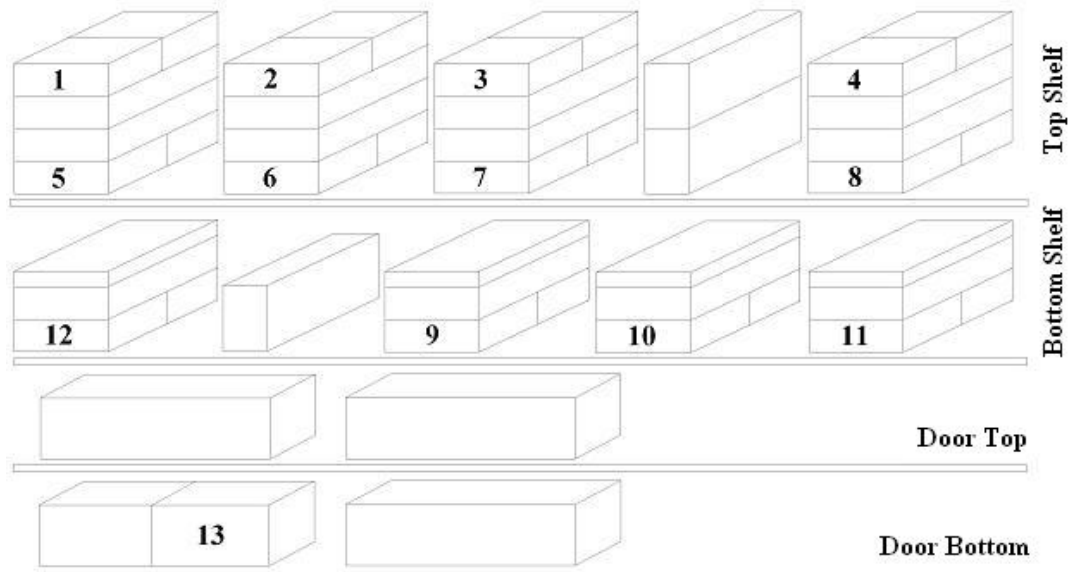


Figure 3.4: The package loading plan

3.2.2 Particle Image Velocimetry (PIV) Measurements

PIV measurements are done to determine the total air flow rate blown to the freezer and also individual air flow rates for each air supply holes. To calculate the volumetric flow rate at each air supply hole, velocities are determined. To minimize the error that will occur in velocity measurements, the exit of the air supply holes are extended towards inside the cabinet. To send the laser ray inside the cabinet the side wall of the refrigerator is cut and replaced with the plexiglass material. Also the door of the refrigerator is demounted and replaced with the plexiglass to obtain a display by cameras. Inner walls of the cabinet was painted with black colour to prevent the reflection of light. Two cameras (Dantec Dynamics Hisense MKI) are used so that 3D velocity vectors can be obtained in one cross section. Nd-YAG laser with 8Hz energy pulse is used. Oil particles which have diameter of 25 micro meter were used for seeding. The repeatability and reproducibility (R&R) of these measurements are guaranteed by Gauge R&R study according to 6 sigma criteria which is $GR\&R < 30$.

The calculated volumetric flow rates will be used as boundary conditions for air supply holes in CFD analyses. The experimental setup for PIV is given in Figure 3.5.



Figure 3.5: The experimental setup for PIV

4. COMPUTATIONAL FRAMEWORK

The task of obtaining solutions to the governing equations of fluid mechanics represents one of the most challenging problems in science and engineering. The flow is needed to be simulated precisely to obtain more accurate results. A technique that has gained popularity in recent years is computational fluid dynamics. It is possible to solve equations in fluid mechanics using a variety of numerical techniques by the help of improvements in computer hardware, resulting in increased memory and efficiency. Unlike experimental fluid mechanics, the geometry and flow conditions can be easily varied to obtain various design goals. The solution that any such numerical program generates should be validated by comparing it to a set of experimental data; but once its validity has been established, the program can be used for various design purposes, within the limits imposed by the assumptions on which it was based [8].

4.1 Refrigerator 3D CAD Model

In the refrigerator CAD model as shown in Figure 4.1 there is only one shelf in the freezer cabinet and two shelves on the freezer door. Control volume, air-supply holes and suction holes are shown in Figure 4.2. The following assumption is done in CFD analysis : The air which is sent to the control volume through air-supply channels circulates in the freezer cabinet and then is collected back from the suction holes and leaves the control volume. The suction region is composed of three channels as shown in Figure 4.2. The channels which are on the side of the suction region provide flow of air which comes from freezer and the channel which is in the middle provides the flow of warmer air which comes from the fresh-food cabinet. There is insulation between these channels. In Figure 4.3 the package models which are loaded in the freezer cabinet is shown and the packages which are in purple are the ones which have thermocouples. The net volume of the freezer is 90 lt.

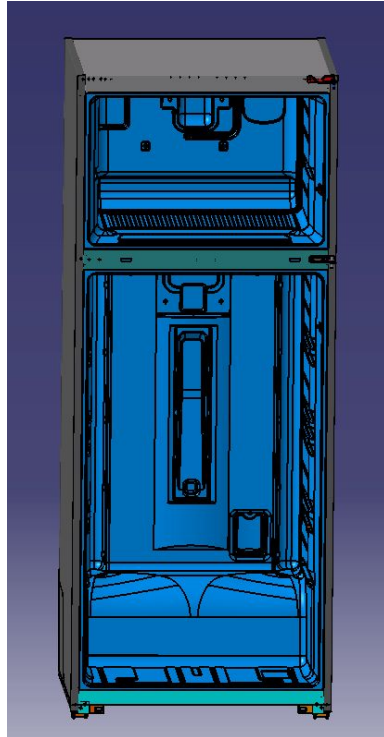


Figure 4.1: 3D CAD model of the refrigerator

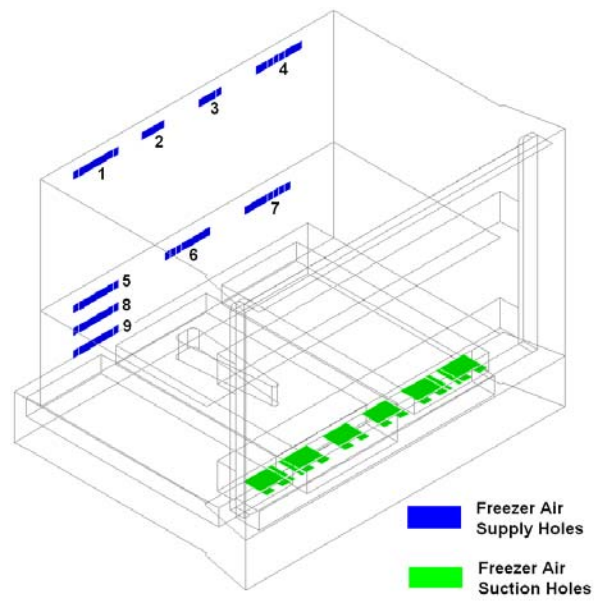


Figure 4.2: The control volume, freezer air supply and suction holes

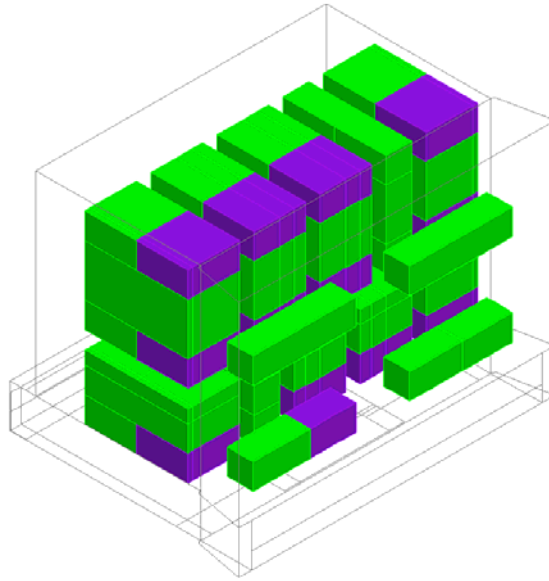


Figure 4.3: Loaded freezer cabinet

4.2 CFD Method

In this subsection, application of the CFD method to the no-frost refrigerator was written. The CFD model can be constituted as open or closed system.

Closed system includes the whole domain of the system inside the control volume. The total mass does not change, but there is flow induced by fan. The most important advantage of the closed system is that the entire system can be solved through the definition of working conditions of evaporator and fan. However, modelling these components, especially evaporator, are very tough and the problem is intricate. To model these components, very small cells have to be utilized for representation of the detailed structure. This causes an increase in the computational time due to the increase in the element size.

In open systems, only the flow domain in the cabinet is modeled and the air supply system are not included in the model. It is considered that there are warm air that exits from suction holes and cool air that enters through the blowing holes. Therefore, it is needed to have information about the direction, rate and the temperature of the flow. The direction and the rate of the flow are measured from PIV measurements. In addition, this information would be obtained from CFD results if the air supply system is modeled as open system. The temperature of the air at supply holes is obtained from temperature controlled room measurements.

In this study, the freezer is considered as an open system and the mesh is generated when the packages are loaded. Approximately, 4 million elements are used. The unsteady CFD analysis are performed in order to investigate the on-time and off-time durations in the freezer. During on-time duration forced convection is prevailing while off-time duration is the phenomena in which the heat transfer is realized through natural convection.

4.3 Initial and Boundary Conditions, Assumptions

Mass, momentum and energy equations are solved for this heat transfer problem. The flow properties of the air which is blowed from air supply holes to the freezer cabinet are taken from 3D PIV measurements and temperature controlled room measurements. Uniform velocity and temperature profiles are assumed at the inlets. In run-time, cold air volumetric flow rate for the freezer cabinet is 10.8 lt/s while 2 lt/s air is entering the fresh-food cabinet. The air temperature blown by air supply holes is taken to be constant as -25 °C. In this study, the ambient temperature is 25°C and the convection heat transfer coefficient for the outer wall of the refrigerator is chosen as 8 W/(m²K). The thicknesses and thermal conductivities of insulation materials, plastic liners, shelves, packages are known parameters. The material properties are given in Table 4.1. The shell conduction option of Fluent is utilized to keep the mesh size affordable by not modelling the wall insulation structure in 3D. The left, right, back, top and the bottom walls have 7.5 cm insulation material called Polyurethane. The door is also made by polyurethane with 6.5 cm thickness. The back wall's temperature was given as -16 °C because the refrigerator evaporator and cold air channels are located very near to the back wall. Heat transfer between freezer and fresh food compartments is taken into consideration by defining a constant temperature to the interface of these two compartments. The bottom wall's temperature was given as 5 °C because this wall is located between freezer and the fresh food compartment. And generally, the fresh food compartment has a temperature which is about 5 °C. The other wall temperatures were set to be the same with room temperature as 25 °C.

The refrigerator is analyzed in a loaded condition. In experiments, it is seen that shelves bend due to the package loading and as a result, the packages which are adjacent to the side walls of the freezer slide and they contact the walls with nearly

half of their side area. However, this bending is not modelled in CFD, a full contact is assumed between package and wall but the thermal conductivity of the contact surfaces are taken as half of the real value. Concerning the door to cabinet contact, it is complicated to include the detailed 3D effects of gasket region (infiltration, thermal bridging etc.) in this CFD model, so an adjustment is made to increase the heat transfer coefficient at the gasket of the door by former CFD studies and experiments [9]. The time information of on and off cycle of the compressor is obtained from temperature controlled room measurements and then assigned in transient CFD analyses. For CFD analysis, the following simplifying assumptions are made: Fluid flow is taken to be incompressible because of the low Mach number.

In reality there is a continuous on and off cycling for compressor, which brings transient nature to the problem. Therefore, the on and off cycling of the compressor was taken into account in CFD analysis. The compressor and the fan is running during on-time period. The Reynolds number is based on average velocity and hydraulic diameter of air blowing holes. The equation (4.5) indicates that the inertial forces are dominant. Consequently, the effect of forced convection is more than the natural convection in the domain. Heat transfer has been maintained by forced convection during the on-cycle hence, the flow is taken to be turbulent. For the on cycle period, energy equation and $k-\varepsilon$ turbulence model was employed and gravitational forces were neglected.

$$Gr = \frac{g\beta(T_s - T_\infty)L^3}{\nu^2} \quad (4.1)$$

$$Re = \frac{\rho v_s^2 / L}{\mu v_s / L^2} = \frac{\rho v_s L}{\mu} = \frac{v_s L}{\nu} \quad (4.2)$$

$$Gr = \frac{(9.8)(1/253.15)(45)(0.583)^3}{(1.2 \times 10^{-5})^2} = 2.4 \times 10^9 \quad (4.3)$$

$$Re = \frac{(3.5 \text{ m/s})(0.583 \text{ m})}{(1.2 \times 10^{-5})} = 1.7 \times 10^5 \quad (4.4)$$

$$\frac{Gr}{(Re)^2} = \frac{2.4 \times 10^9}{(1.7 \times 10^5)^2} = 8.2 \times 10^{-2} < 1 \quad (4.5)$$

Where,

- ρ : density of the fluid
- g : gravitational force
- β : volumetric thermal expansion coefficient
- T_s : supply temperature
- T_∞ : quiescent temperature
- L : characteristic length
- μ : dynamic viscosity
- ν : kinematic viscosity
- v_s : mean velocity of the fluid

On the other hand, the flow is taken to be laminar while the off-time period. The governing equations for natural convection flow are coupled, elliptic, partial differential equations so and so forth it has a considerable complexity. Another problem in obtaining a solution to these equations lies in the inevitable variation of the density ρ with temperature. Several approximations are generally made to simplify these equations. Two of the most important among these are the Boussinesq and the boundary layer approximations. The Boussinesq approximations involve two aspects. First, the density variation in the continuity equation is neglected. Thus, the continuity equation, becomes

$$\nabla \cdot V = 0 \quad (4.6)$$

Secondly, the density difference, which causes the flow, is approximated as a pure temperature or concentration effect (i.e., the effect of pressure on the density is neglected). In fact, the density difference is estimated for thermal buoyancy as

$$\rho_\infty - \rho = \rho\beta(T - T_\infty) \quad (4.7)$$

These approximations are employed very extensively for natural convection. An important condition for the validity of these approximations is that $\beta(T - T_\infty) \ll 1$. Therefore, the approximations are valid for small temperature differences if β is

essentially unchanged. Similarly, for large temperature differences encountered in fire and combustion systems, these approximations are generally not applicable [10].

In this point of view, for the natural convection case the Boussinesq approximation and energy equation were employed to the problem. Laminar viscous model was employed due to the low Reynolds number. The gravitational forces were taken into account. The temperature difference between the surfaces facing each other (shelves, side walls) is quite small so radiation heat transfer within the refrigerator is not considered.

Table 4.1: The Material Properties

Properties	Units	Method	Value
Material: air (fluid)			
Density	Kg/m ³	Constant	1.4135
Cp (Specific Heat)	J/Kg-K	Constant	1005
Thermal Conductivity	W/m-K	Constant	0.022
Viscosity	Kg/m-s	Constant	1.6149e-05
Material: polyurethane (solid)			
Density	Kg/m ³	Constant	40
Cp (Specific Heat)	J/Kg-K	Constant	1100
Thermal Conductivity	W/m-K	Constant	0.022
Material: plastic (solid)			
Density	Kg/m ³	Constant	1050
Cp (Specific Heat)	J/Kg-K	Constant	1500
Thermal Conductivity	W/m-K	Constant	0.136
Material: packages (solid)			
Density	Kg/m ³	Constant	3106
Cp (Specific Heat)	J/Kg-K	Constant	1047
Thermal Conductivity	W/m-K	Constant	1.58

4.3.1 Standard κ - ϵ model

The standard κ - ϵ turbulent model is formed for the turbulent kinetic energy and dissipation rate of this energy. Standard κ - ϵ model for turbulent kinetic energy (κ) and the dissipation rate of this energy leads transfer equations and tests. It is assumed that the flow is completely turbulent and molecular viscosity effects are neglected. Standard κ - ϵ model is presented as:

$$\frac{\partial(\rho\kappa)}{\partial t} + \frac{\partial}{\partial x_i}(\rho\kappa u_i) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\kappa} \right) \frac{\partial \kappa}{\partial x_j} \right] + G_k + G_b - \varepsilon\rho - Y_M + S_\kappa \quad (4.8)$$

$$\frac{\partial(\rho\varepsilon)}{\partial t} + \frac{\partial}{\partial x_i}(\rho\varepsilon u_i) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_1 \frac{\varepsilon}{\kappa} (G_k + C_{3\varepsilon} G_b) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{\kappa} + S_\varepsilon \quad (4.9)$$

Turbulent viscosity,

$$\mu_t = \rho C_\mu \frac{\kappa^2}{\varepsilon} \quad (4.10)$$

Where,

κ : Turbulent kinetic energy

G_κ : The change of turbulent kinetic energy depending on average velocity gradients

G_b : The change of turbulent kinetic energy depending on bouyancy

σ_κ : Prandtl number for turbulent kinetic energy

σ_ε : Prandtl number for the dissipation rate of turbulent kinetic energy

μ_t : Turbulent viscosity

The coefficients used in standard κ - ε model were determined in the experiments which are tabulated in Table 4.2.

Table 4.2: Coefficients in κ - ε turbulent model

Coefficients	Values
C_μ	0,09
$C_{1\varepsilon}$	1,44
$C_{2\varepsilon}$	1,92
σ_κ	1
σ_ε	1,3

4.4 CFD Mesh Model

The geometric model that is used in CFD analyses is simplified to capture a better quality of flow domain mesh. The computational domains are discretized by using a mesh generating software, GAMBIT. A comprehensive mesh-dependency study has also been undertaken and the numerical accuracy. By the way, the numerical accuracy and computational economy is supplied. Hybrid elements are used in the mesh; 60% are hexahedral and the rest is tetrahedral elements. Mesh is composed of 4 million elements. The generated mesh is exported to a commercial CFD software, FLUENT. The assumptions, initial and boundary conditions mentioned as above are applied in this software. The analyses are run in a cluster of 10 Itanium nodes. For mesh partition, Metis algorithm is utilized.

4.5 Grid Independence

In off-time period, fan is not running, so heat transfer occurs by natural convection. The mesh dependency was investigated for the natural convection case. Several meshing operations were performed to investigate the dependency of the solution on the mesh size. A vertex was chosen inside the freezer compartment and the temperature values were read from this point. Also, the mean temperature for the top wall of the freezer is used for the correlation. The graphics which demonstrate the independency of the solution on the mesh size were given in Figure 4.4 and Figure 4.5. For this problem, it is seen that the solution is not dependent to the mesh size for greater values of 1 million elements. It is decided to use approximately 4 million hybrid elements for the following CFD analysis. The CPU time required for a 21 minute forced convection flow solution, is 138000 seconds. The CPU time required for a 29 minute natural convection flow solution, is 195000 seconds.

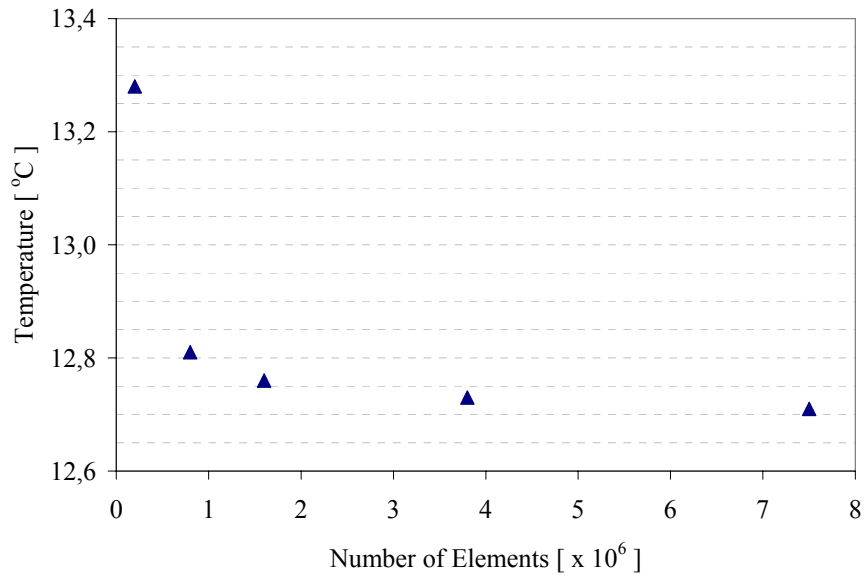


Figure 4.4: The temperatures at the center of the fluid domain

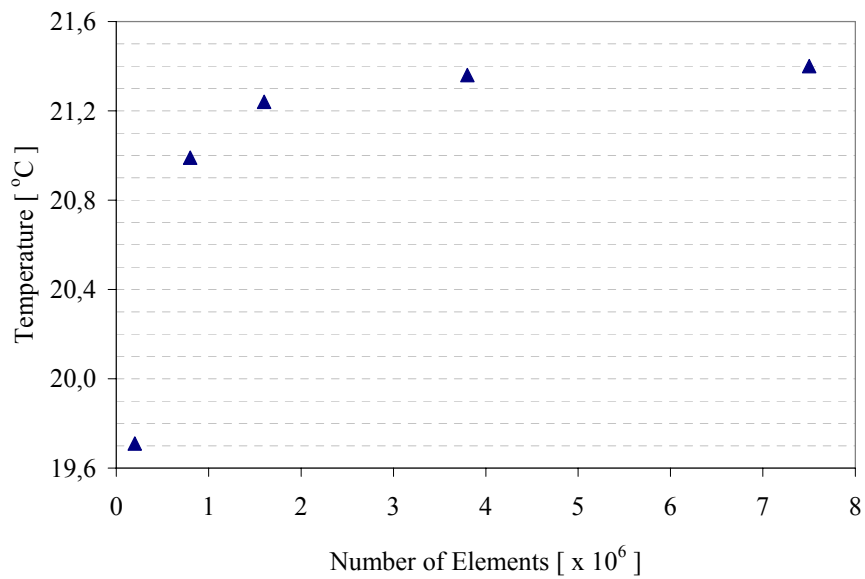


Figure 4.5: The mean temperatures for the top wall of the freezer

4.6 Flow Chart

The objective is to keep food fresh by providing a uniform temperature distribution. The 3D transient CFD analyses have been performed to model the heat transfer by forced convection during the on-time and by natural convection during off-time period with an appropriate loading plan. After the validation of the CFD analyses results a DOE study is performed to find the optimum condition for the temperature uniformity inside the refrigerator cabinet. The optimum design is obtained by DOE

study. The new design of the air supply channel is necessary to provide the optimum condition for the position of the air blowing holes and distribution percentage of air flow. The design stage of the air supply channel was performed by CFD analyses. A prototype of the new design is produced and tested with PIV measurements. The CFD analyses obtained are shown to be in a reasonable accuracy by experiments. Then the prototype is tested by temperature controlled room measurements and the desired temperature uniformity inside freezer cabinet is obtained. A flowchart of the entire study is given in Figure 4.6.

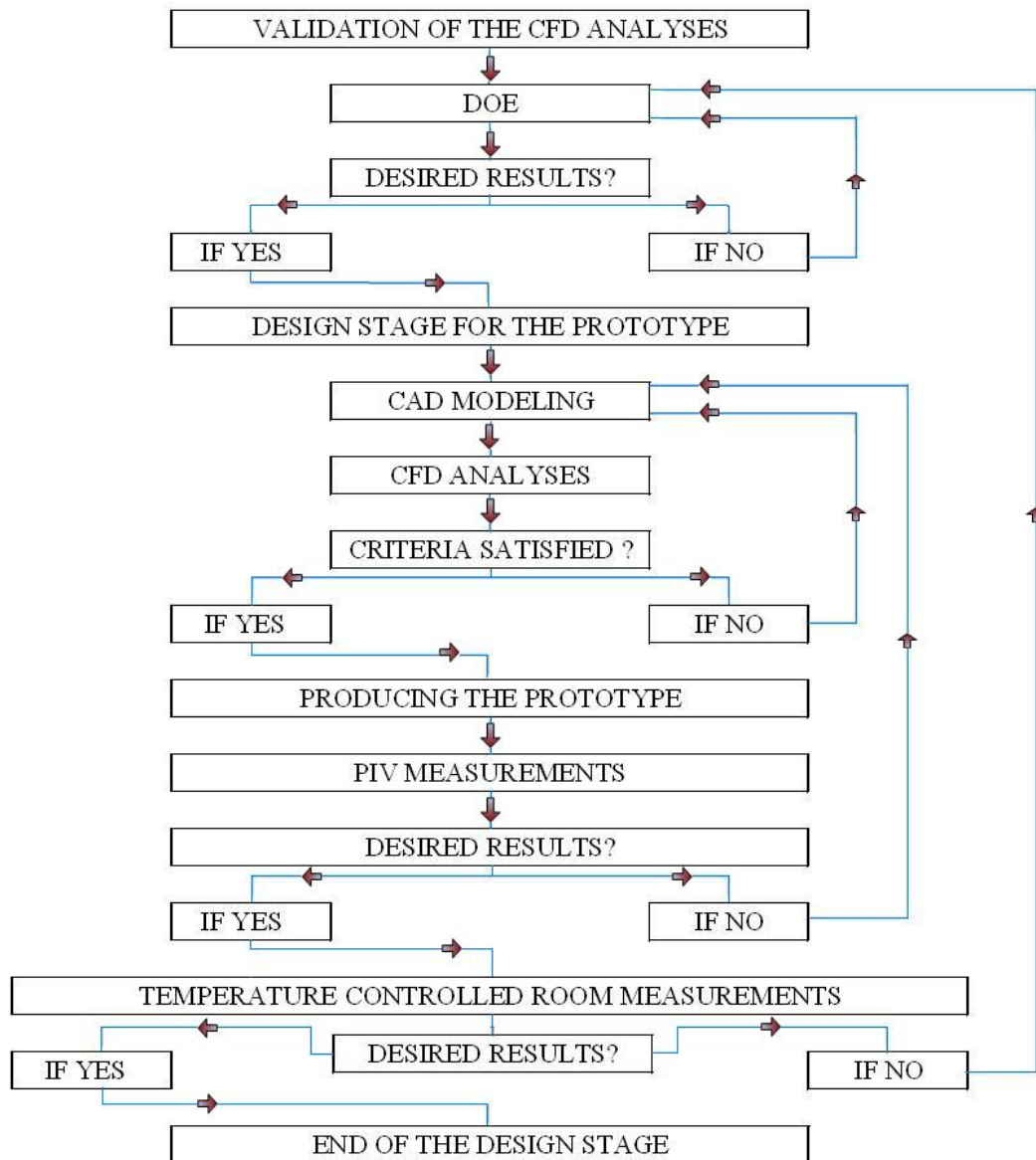


Figure 4.6: The flowchart of the entire study

5. INVESTIGATION OF THE ORIGINAL FREEZER COMPARTMENT OF THE REFRIGERATOR WITH EXPERIMENTAL AND CFD ANALYSIS

First of all, the temperature controlled room measurement was performed to investigate the temperature distribution inside freezer compartment. The experiment was carried out to determine the energy consumption of the original refrigerator in 25°C conditioned room. The cabinet wall temperature, the package temperatures after off-time period, the temperature of air blown at air supply holes were obtained from the experiment and assigned as initial and boundary conditions in CFD analyses. The transient CFD analyses simulate the on and off time periods of the compressor. The fan was not modeled because the freezer cabinet was modeled as open system in CFD analyses. In this instance, the air flow characteristics at air supply holes such as direction and volumetric flow rate should be known. This data was obtained by PIV measurement.

5.1 TCRM to Determine the Temperature of the Packages

In Table 5.2, the temperature values in the center of 13 packages are shown at the end of on-time and off-time periods of the compressor in the 25 °C room conditions. The temperature measurements are averaged over a couple of periods. At the end of the off-time period of the compressor, the temperature gradient between the warmest and coldest packages are found to be 2.6°C. At the end of the run-time period of the compressor, the temperature gradient between the warmest and coldest packages are found to be 2.35°C. As a result of the experiments, the warmest package are found to be the 2nd package which rest at the upper shelf.

5.2 PIV Measurements to Determine the Distribution of the Flow Rate

The air supply holes of the original condition can be seen in Figure 5.1. The PIV measurement was performed to learn the total amount and also the distribution of the volumetric flow rate at air supply holes. The results for the volumetric flow rates at each air supply hole was given in Table 5.2. The results show that the 41% of total volumetric flow rate is blown from the upper part of the cabinet and the rest is blown from the bottom part. In Figure 5.2, the velocity contours exiting each air supply hole is seen from the front side of the freezer cabinet. The calculated volumetric flow rates will be used as boundary conditions for air supply holes in the CFD analyses of original condition. The total air flow rate that is sent into the freezer and fresh-food compartment is found to be 10.8 lt/s and 1.7 lt/s, respectively.

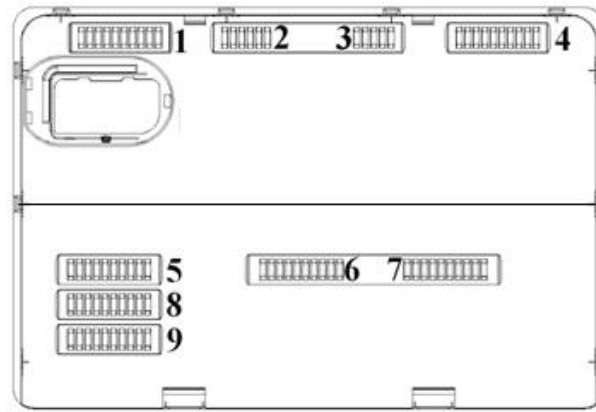


Figure 5.1: The original refrigerator evaporator cover (Back wall)

Table 5.1: PIV Measurements for Flow Rates

Position of Air Supply Holes	PIV Measurements Flow Rate [lt/s]
# 1	1.5
# 2	0.5
# 3	1.2
# 4	1.25
# 5	1.15
# 6	1.60
# 7	1.60
# 8	0.90
# 9	1.10

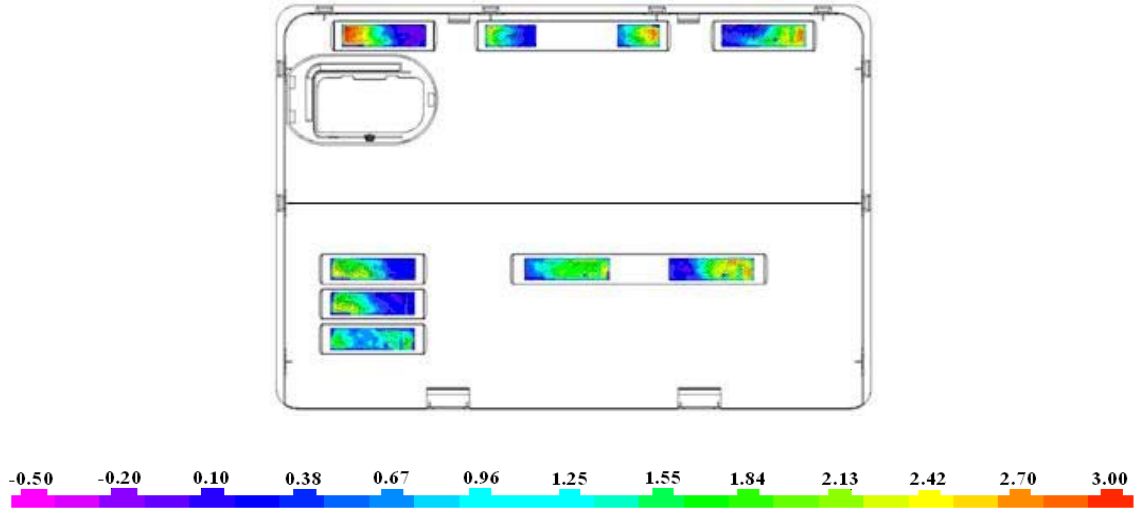


Figure 5.2: PIV Results for the original condition (m/s)

5.3 CFD Analyses to Determine the Temperature of the Packages

The assumptions, initial and boundary conditions were given in Chapter 4. The differences are less than 1°C when CFD analyses results are compared with temperature controlled room measurements. These results are taken to be satisfactory for validation of the CFD analyses. In literature review, it is seen that 1°C temperature difference can be accepted for validation of the CFD analyses [1]. The temperature distribution on the packages at the end of the off-time and on-time period of the compressor is given in Figure 5.3 and Figure 5.4, respectively. The temperatures measured from the center of the packages are given in Table 5.2. Figure 5.5 shows the variation of temperature at the end of off-time period from one lateral side to the other along the central longitudinal axis. From the figure, the variations along this direction are negligible except near the wall. A very similar result had obtained in literature review and given in Figure 2.9.

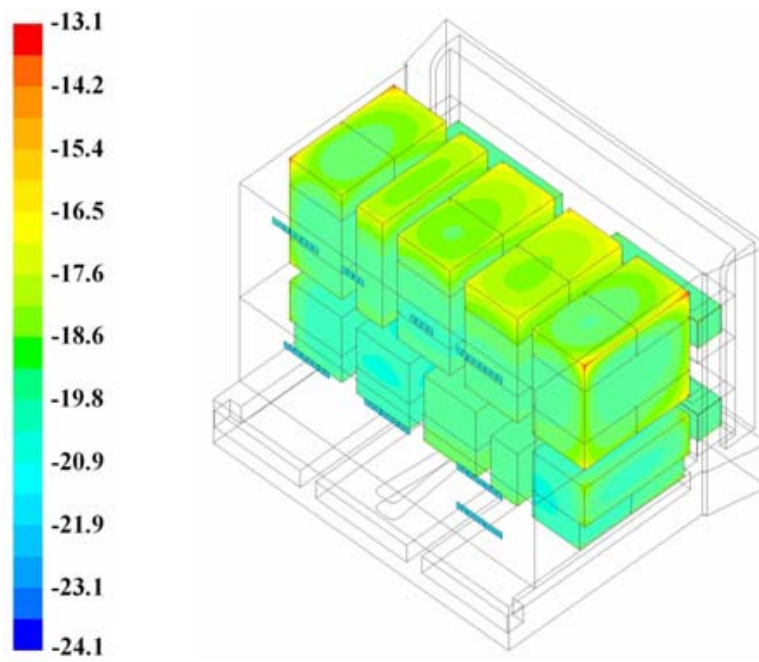


Figure 5.3: Temperature distribution on the packages at the end of off-time

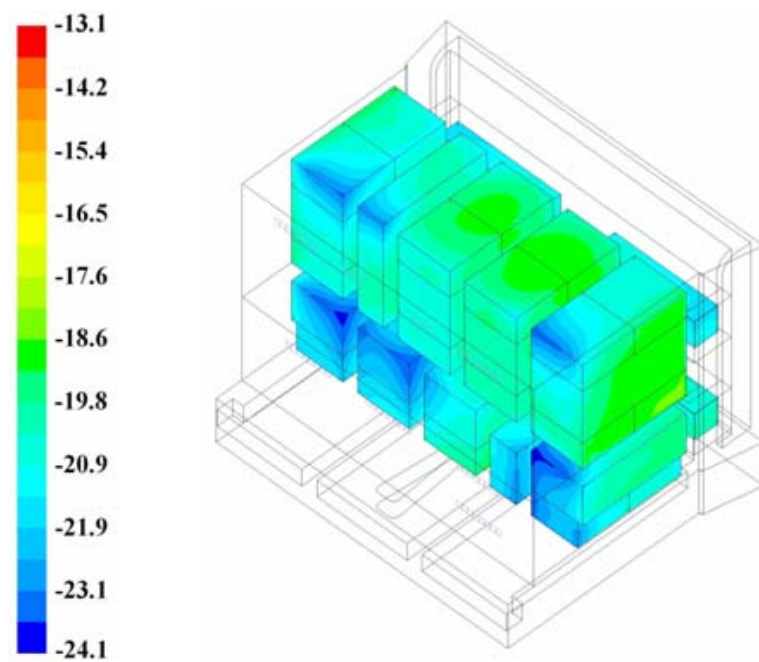


Figure 5.4: Temperature distribution on the packages at the end of on-time

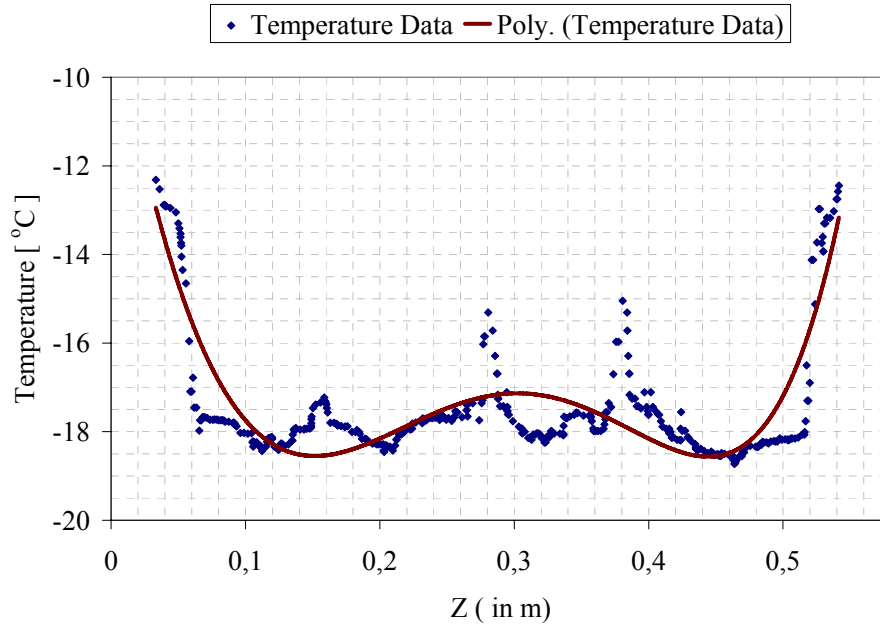


Figure 5.5: The variation of temperature along the central longitudinal axis

5.4 Comparison

The temperature controlled room measurements and CFD results are compared in Table 5.4. Both temperature controlled room measurements and CFD results indicate that the warmest package is 2nd and the coolest package is 13th at the end of on and off-time. These results are satisfactory for accurate prediction of the warmest and the coolest packages.

According to Figure 3.4 which was given for loading plan of packages, the warmest package numbered 2 is located at the top shelf of the freezer and the coolest package numbered 13 is at the bottom door shelf. Three cross-sections inside freezer cabinet were taken to examine the temperature contours at the end of on-time period of the compressor. The results were given in Figure 5.6 and 5.7. According to figures, the bottom part of the freezer cabinet is cooler than the top of the freezer. This is due to the fact that 59% of the air flow of freezer is blown from the bottom holes where 41% of the air flow is blown from the upper holes in the freezer. The 9th package is located at the bottom part of the freezer cabinet and above the air channel which absorbs warm air from the fresh-food cabinet. Consequently, this package is the warmest package of the bottom part of the freezer. Figure 5.7 depicts that the temperature values inside freezer cabinet is increasing from back wall to door side.

Table 5.2: Experimental Measurements and CFD Results

Packages	Experimental Results		CFD Analysis Results	
	Max [°C]	Min [°C]	Max [°C]	Min [°C]
MP1	-18.57	-19.58	-18.64	-19.69
MP2	-18.08	-19.10	-18.11	-19.23
MP3	-18.41	-19.37	-18.28	-19.37
MP4	-19.43	-20.43	-19.00	-20.09
MP5	-18.31	-19.18	-18.63	-19.36
MP6	-19.04	-19.99	-19.24	-20.28
MP7	-19.76	-20.68	-19.68	-20.58
MP8	-19.44	-20.31	-19.23	-20.12
MP9	-18.89	-19.70	-18.79	-19.54
MP10	-19.97	-20.75	-19.85	-20.74
MP11	-20.20	-20.88	-19.89	-20.66
MP12	-20.11	-20.93	-19.73	-20.61
MP13	-20.68	-21.45	-20.22	-21.15
Average	-19.30	-20.18	-19.18	-20.11
Max ΔT	2.60	2.35	2.12	1.92

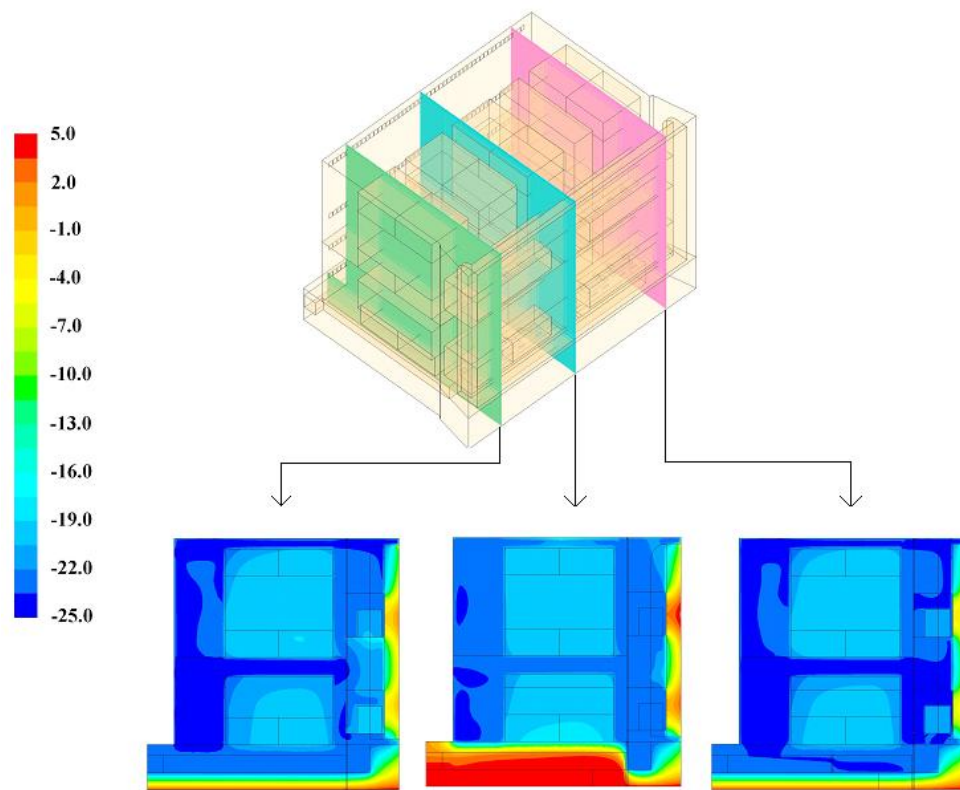


Figure 5.6: Temperature distribution of XY planes at the end of on-time (°C)

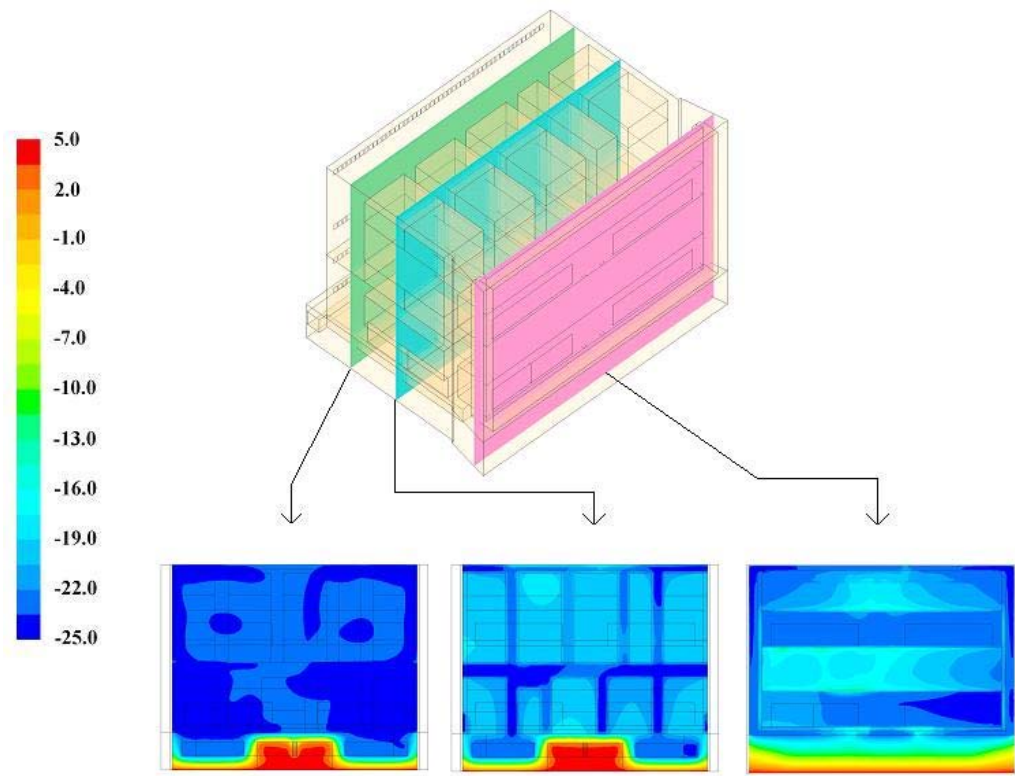


Figure 5.7: Temperature distribution of YZ planes at the end of on-time (°C)

Figure 5.8 and 5.9 show the temperature contours for different cross-sections at the end of off-time period of the compressor. Heat transfer occurs by natural convection during this period. It is seen that the temperature values are increasing near the cabinet walls.

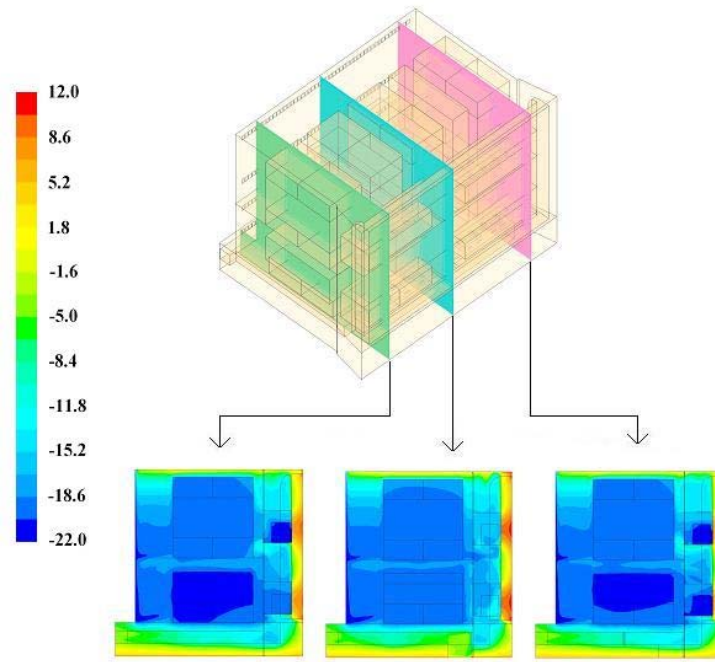


Figure 5.8: Temperature distribution of XY planes at the end of off-time (°C)

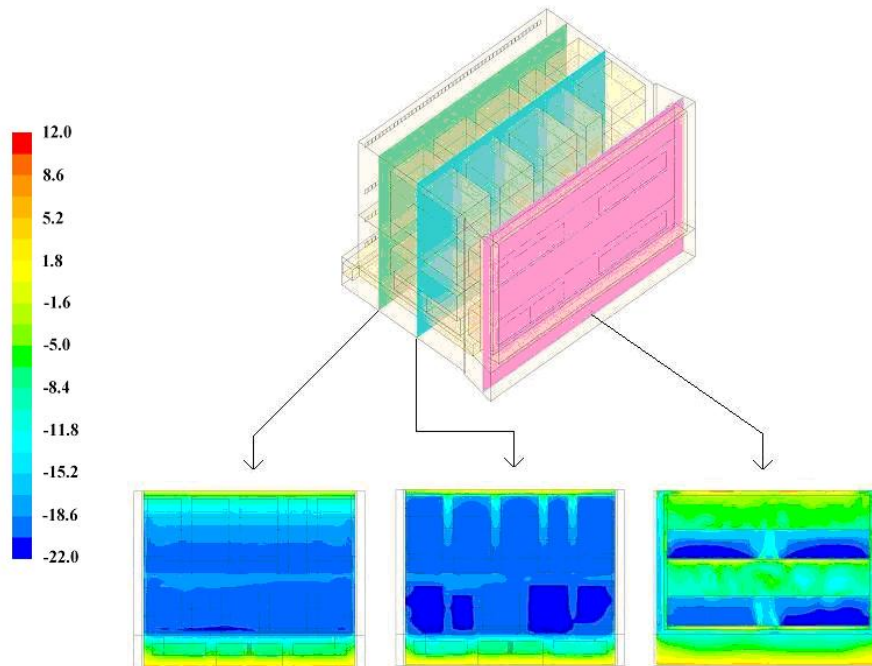


Figure 5.9: Temperature distribution of YZ planes at the end of off-time (°C)

The streamlines coloured by temperature can be seen in Figure 5.10. The streamlines are examined to understand that the flow paths and temperature distribution depends on the region. The cold air blown from air supply holes to the cabinet is becoming warmer by transferring heat from the cabinet walls and packages.

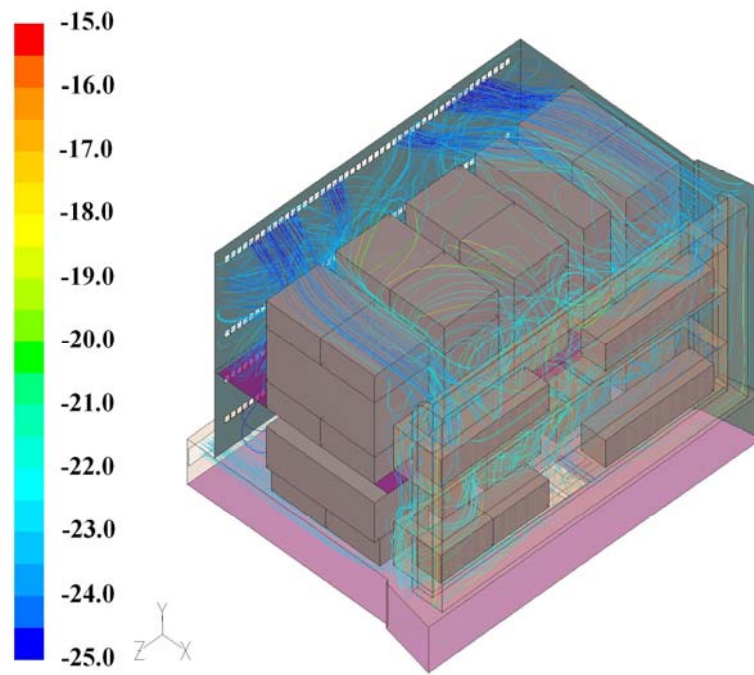


Figure 5.10: Streamlines coloured by temperature

6. IMPROVEMENT STUDIES IN FREEZER COMPARTMENT

The purpose of this study is to provide a uniform temperature distribution inside the freezer cabinet. The row and the position of the air blowing holes, the angle of blowing, the percent distribution of flow rates at each air blowing hole, total flow rate, temperature of blown air and the volume of the freezer compartment are the parameters which effect the temperature distribution inside the freezer cabinet. The temperature controlled room and PIV measurements are performed to determine the effects of these parameters. The design of experiment study which known as DOE is carried out to improve the system. Several experiments should be performed to determine the effect of each parameter. Producing each different design and testing them causes time consuming and increased costs. Therefore, a DOE study which aims to improve the system consists of 220 different design was realized by a commercial CFD code, Fluent. The design table was constituted by a program, named Minitab. The design table for the DOE study were given for the first row of air supply holes in Appendix-A. Total examined row number is 4. The transient CFD analyses were performed just for the on-time period of the compressor. In Chapter 5, the CFD model was validated for a reference refrigerator by experimental measurements before starting the DOE study. The information about the position of air blowing holes and the volumetric flow rates at each hole for the freezer compartment are the results of this parametric study. The analyses was solved by using the facility of ITU UYBHM (National High Performance Computation Center) laboratories. This chapter does not include the detailed results of these analyses. The goal is the design of the air supply channels located at the rear side of the freezer back wall in order to provide the optimum condition for the position of the air blowing holes and percent distribution of air flow. Once the optimum positions of air blowing holes are identified by DOE study, a further seperate CFD study is performed to design the air supply channel shape. The design stage of the air supply channel was performed by CFD analyses then the prototype of air supply channel which supplies the desired condition was produced. The prototype was tested by PIV measurements and the validation of the CFD analyses had been assured with a good

agreement. The prototype was mounted to the related place on the refrigerator. After this process the refrigerator was tested by temperature controlled room measurements. The CFD analyses results obtained a good agreement with the temperature controlled room measurements in terms of package temperature uniformity.

6.1 CFD Analysis

6.1.1 CFD Analyses to Determine the Package Temperatures

The results of DOE study for optimum condition is given in Figure 6.1. The optimum position of the air blowing holes determined by the DOE study is given in Figure 6.2. The 70% of total volumetric flow rate should be blown by the upper holes, and the rest by the lower holes. The temperature difference between the warmest and the coolest packages was obtained as 1.04 °C from the result of DOE study. The CFD analysis result for the package temperatures at the end of the on-time period of the compressor was given in Table 6.1. The temperature difference between the warmest and the coolest packages was obtained as 1.92 °C from the result of CFD analysis performed for the original refrigerator. Therefore, according to the CFD analysis results it can be claimed that the temperature uniformity inside the freezer cabinet is provided. These results which were obtained for the optimum condition should be validated by the experimental measurements. The temperature distribution on the packages for the optimum condition at the end of on-time can be seen in Figure 6.3. Three cross-section from back wall to the door of the freezer cabinet was taken to examine the temperature contours at the end of on-time period of the compressor. The results were given in Figure 6.4. Figure 6.5 demonstrates the temperature contours at the end of on-time period of the compressor for lateral cross-sections taken from the freezer cabinet.

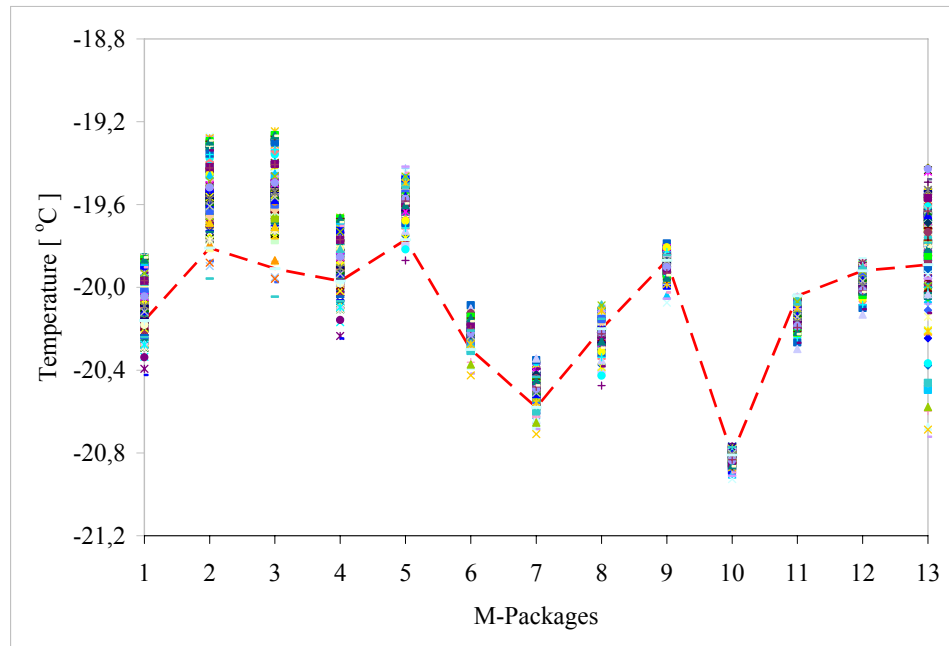


Figure 6.1: The results of DOE study for optimum condition

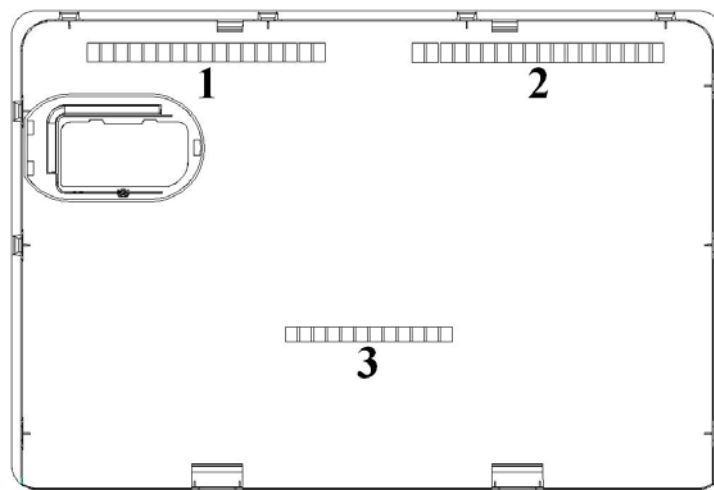


Figure 6.2: Optimum position for air supply holes

Table 6.1: CFD Analysis result for the package temperature at optimum condition

Packages	CFD Analysis Results
	Min [°C]
MP1	-20.16
MP2	-19.81
MP3	-19.91
MP4	-19.97
MP5	-19.77
MP6	-20.30
MP7	-20.58
MP8	-20.20
MP9	-19.87
MP10	-20.81
MP11	-20.04
MP12	-19.92
MP13	-19.89
Average	-20.10
Max ΔT	1.04

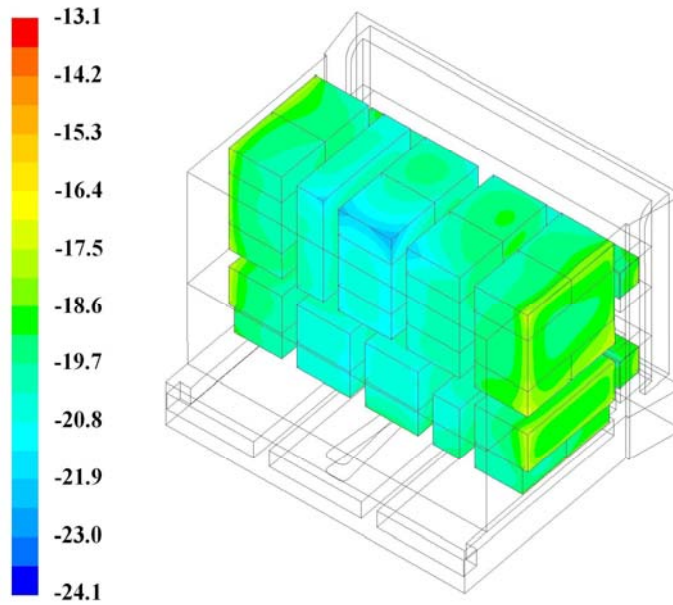


Figure 6.3: Temperature distribution on the packages at optimum condition (°C)

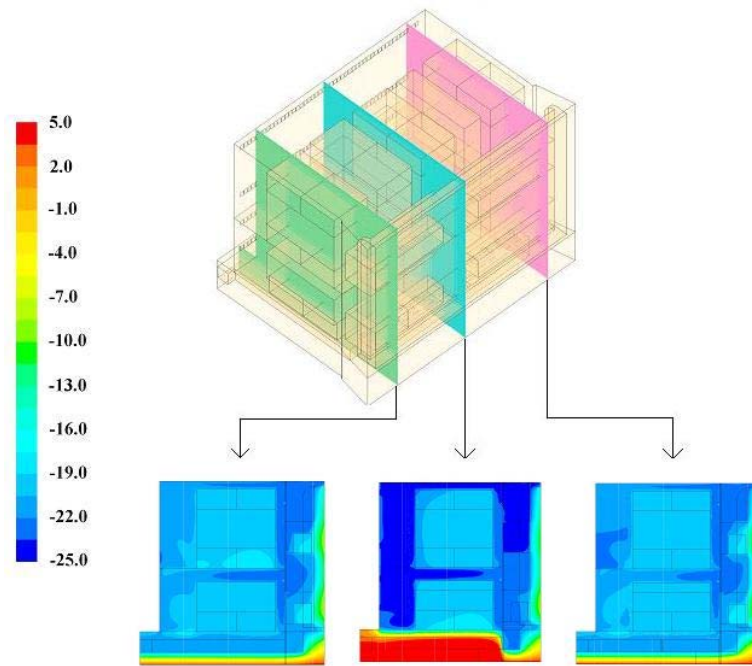


Figure 6.4: Temperature distribution of XY planes at the end of off-time (°C)

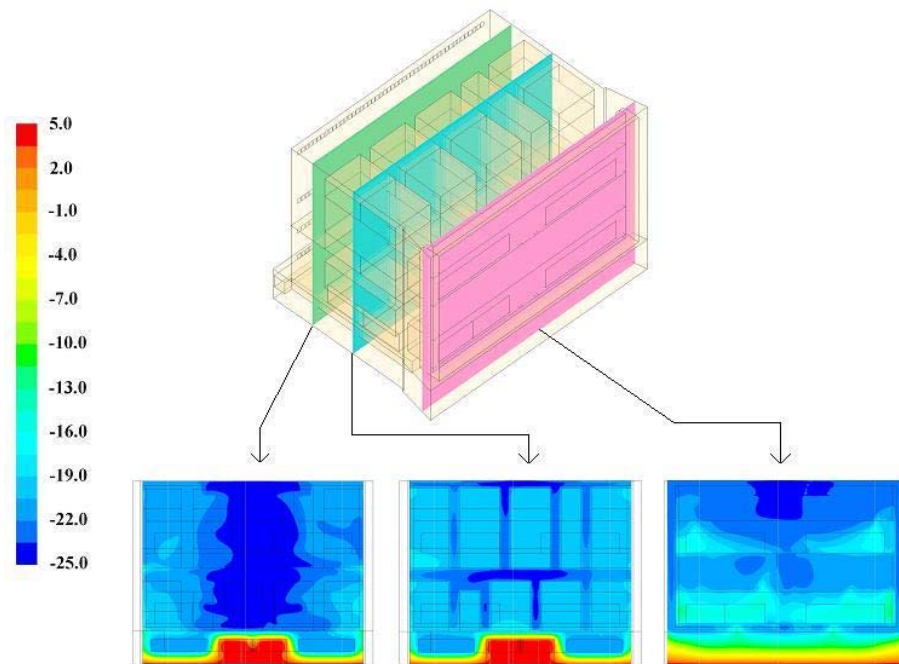


Figure 6.5: Temperature distribution of YZ planes at the end of off-time (°C)

6.1.2 CFD Analyses to Determine the Distribution of the Flow Rates

The new design of the air supply channel is necessary to provide the optimum condition for the new positions of air blowing holes and percent distribution of air flow. Different air supply channel geometries can be produced and tested by PIV measurements to acquire the optimum condition but this process was not preferred because of time consumption. The design stage of the air supply channel was performed by CFD analyses. The fan was not modeled hence the CFD model was constructed as open system. In this case, boundary condition for the volumetric flow rate should be known. It is thought to be the same volumetric flow rate of original condition can be supplied. The total air flow rate that is sent into the freezer and fresh-food compartment is found to be 10,58 lt/s and 1.93 lt/s, respectively. The results show that the 71,7% of total volumetric flow rate is blown from the upper part of the cabinet and the rest is blown from the bottom part. The results for the volumetric flow rates at each air supply hole was given in Table 6.2. The exploded and mounted views of the air supply channels were given in Figure 6.6. In Figure 6.7, the position of the inlet and outlets can be seen. For the new design of the air supply channel the velocity vectors at mid-plane were obtained by CFD analyses and given in Figure 6.8.

Table 6.2: CFD analysis results for flow rates of optimum condition

Position of Air Supply Holes	CFD Analysis Results Flow Rate [lt/s]	Distribution Percentage
# 1	3.92	37.1%
# 2	3.66	34.7%
# 3	2.99	28.3%

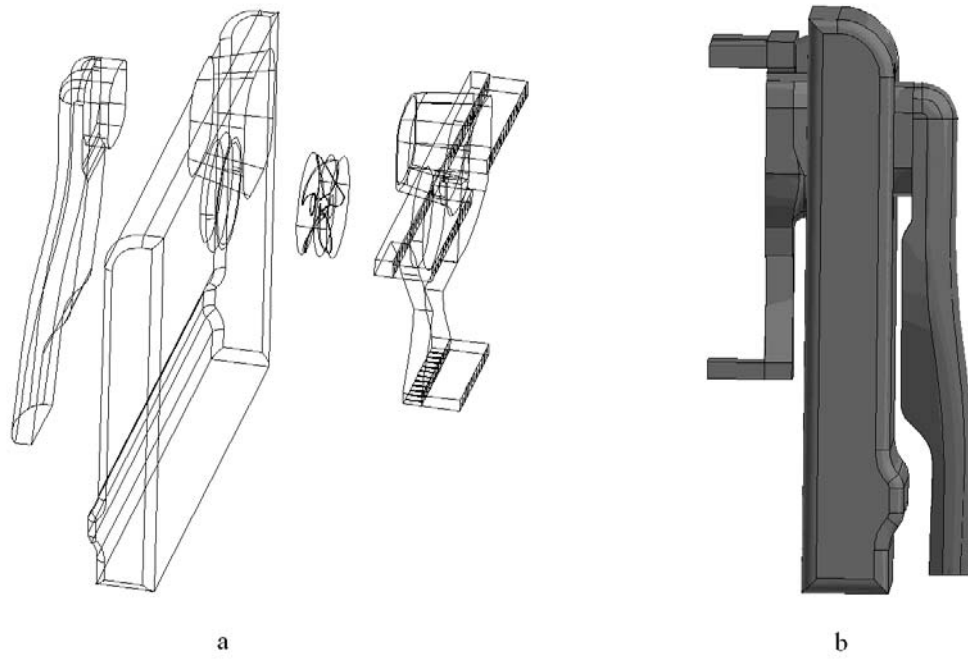


Figure 6.6: The exploded (a) and mounted (b) views of the air supply channels

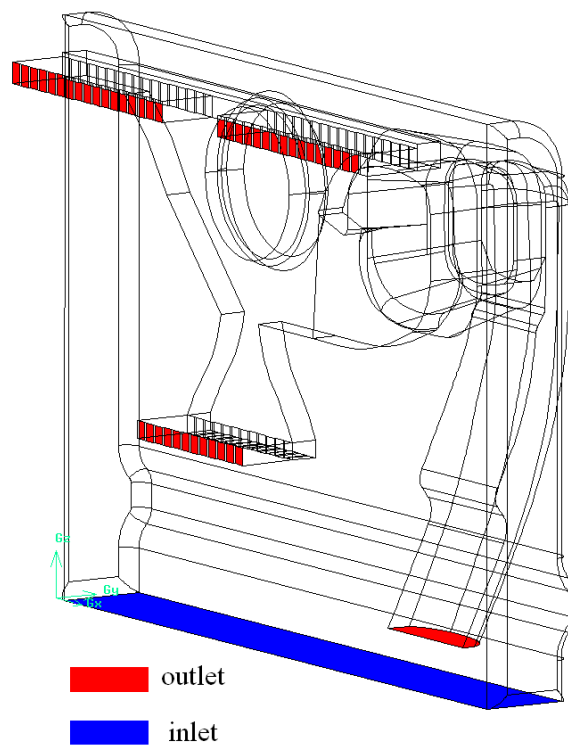


Figure 6.7: The position of the inlet and outlets

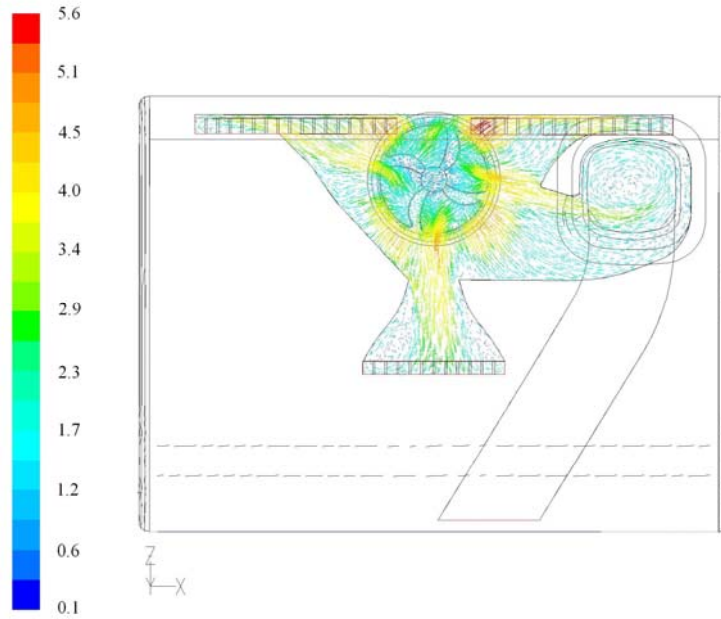


Figure 6.8: The velocity vectors for the optimum design (m/s)

6.2 PIV Measurements

The PIV measurement was performed to validate the results of CFD analyses in terms of total amount and also the distribution of the volumetric flow rate at each air supply hole for the optimum condition. The results for the volumetric flow rates at each air supply hole was given in Table 6.3. The results show that the 69% of total volumetric flow rate is blown from the upper part of the cabinet and the rest is blown from the bottom part. The total air flow rate that is sent into the freezer and fresh-food compartment is found to be 10 lt/s and 2 lt/s, respectively.

Table 6.3: PIV measurements for flow rates of optimum condition

Position of Air Supply Holes	PIV Measurements Flow Rate [lt/s]	Distribution Percentage
# 1	3.3	33%
# 2	3.6	36%
# 3	3.1	31%

6.3 Temperature Controlled Room Measurements

In Table 6.4, the temperature values in the center of 13 packages are shown at the end of off-time and on-time periods of the compressor in the 25 °C room conditions. At the end of the off-time period of the compressor, the temperature difference between the warmest and coldest packages are found to be 1.78°C. At the end of the on-time period of the compressor, the temperature difference between the warmest and coldest packages are found to be 1.24°C. As a result of the experiments, the temperature uniformity of the packages is reached when compared with the original condition.

Table 6.4: TCRM for the optimum condition

Packages	Experimental Results	Experimental Results
	Max [°C]	Min [°C]
MP1	-18.92	-20.38
MP2	-18.39	-20.04
MP3	-18.58	-20.24
MP4	-19.04	-20.49
MP5	-19.32	-20.15
MP6	-19.72	-20.73
MP7	-19.76	-20.70
MP8	-19.25	-20.15
MP9	-19.74	-20.50
MP10	-20.17	-20.82
MP11	-19.00	-19.59
MP12	-18.93	-19.58
MP13	-19.34	-20.34
Average	-19.24	-20.29
Max ΔT	1.78	1.24

7. THE EFFECT OF TOTAL FLOW RATE AND AIR BLOWING TEMPERATURE

In this section, the effects of total flow rate and temperature of the air flow sent into the freezer cabinet were investigated. Firstly, the variation of average temperature of all packages during on-time period for the original condition was examined. The flow rate was 10.8 lt/s and the temperature of blown air was -25°C for the 21 minute on-time period. Then the flow rate was increased to 12.1 lt/s as 12% for the same air temperature and on-time period was decreased as 28% to reach the same average package temperature of original condition. Furthermore, the original flow rate was kept the same but the blown air temperature was set to be -28°C by 12% increase. The on-time period was decreased as 54% to reach the same average package temperature of original condition. As a result, the air temperature sent inside the freezer cabinet has a more dominant effect on cooling time than the flow rate. The CFD analysis result was given in Figure 7.1.

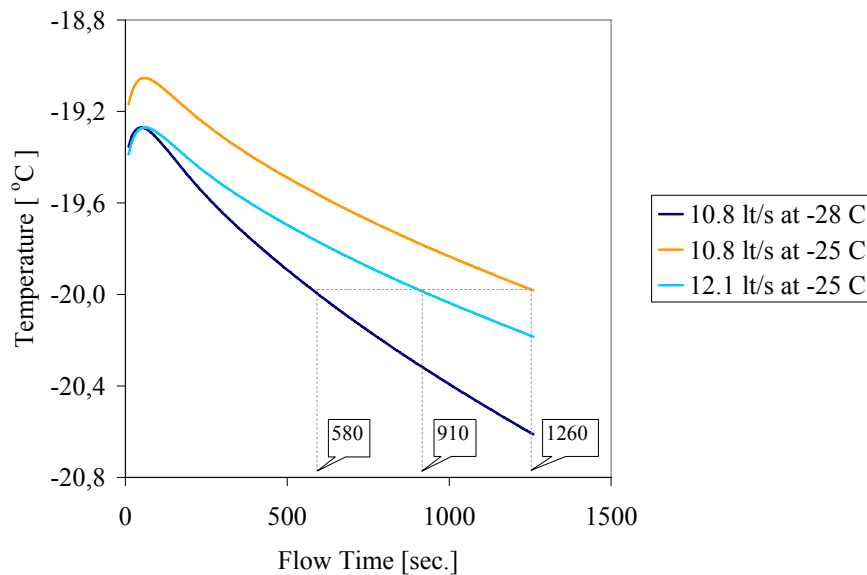


Figure 7.1. The effect of flow rate and air temperature on cooling time

8. CONCLUSION AND RECOMMENDATIONS

This study focuses on the airflow behavior and the temperature distribution inside the freezer compartment of a domestic no-frost refrigerator because uniformity of temperature distribution in a loaded freezer cabinet is one of the most important factors affecting energy consumption of a refrigerator. Energy consumption increases in a freezer cabinet if the temperature difference between the warmest load package and average of all packages is high. The objective is to keep food fresh by providing a uniform temperature distribution and concurrently to reduce the energy consumption. The cooling time (on-time) and off-time are the other parameters which determine the energy consumption of the refrigerator. The goal is decreasing the on-time period. On the other hand the longer off-time period is demanded. A study which focuses to improve the insulation efficiency is necessary to elongate the off-time period.

8.1 Application of The Work

In this study, the 3D transient CFD analyses have been performed to model the heat transfer by forced convection during the run-time and by natural convection during off-time period with an appropriate loading plan. The initial and boundary conditions were provided from temperature controlled room and 3D PIV measurements. After the validation of the CFD analyses results a DOE study is performed to find the optimum condition for the temperature uniformity inside the refrigerator cabinet. The optimum design is obtained by DOE study. The new design of the air supply channel is necessary to provide the optimum condition for the position of the air blowing holes and distribution percentage of air flow. The design stage of the air supply channel was performed by CFD analyses. A prototype of the new design is produced and tested with PIV measurements. The CFD analysis obtained are shown to be in a reasonable accuracy by experiments. Then the prototype is tested by temperature controlled room measurements and the desired temperature uniformity inside freezer cabinet is obtained. Moreover, the parameters such as flow rate and air temperature blown inside the freezer cabinet which effect the cooling time were examined by CFD analysis.

8.2 Recommendations and Future Work

The temperature distribution inside refrigerator is effected by its type and geometry. For instance, a no-frost refrigerator was choosen for this study which has just one fan for cooling process. The distribution of cold air to the freezer and fresh-food cabinets is maintained by air supply channels. Then the air which became warmer inside cabinets is absorbed by air suction channels. The fresh-food air suction channel is located below the bottom wall of freezer cabinet. This channel carries relatively warm air than freezer air suction channels. Consequently, the mid and the bottom part of the freezer becomes warmer. This situation is not observed in other type of no-frost refrigerators which perform the cooling process of freezer and fresh-food compartment separately by the help of two fans. The effect of this situation is not major but even so a similar study should have been revised for the other type of refrigerators. Although, determined optimum conditions for the position of air supply holes in this study can be acceptable for all type of refrigerators.

Moreover, the parameters such as flow rate and air temperature blown inside the freezer cabinet which effect the cooling time were examined by CFD analysis. As a result, the air temperature sent inside the freezer cabinet has a more dominant effect on cooling time than the flow rate. Nonetheless, a detailed study which surveying the optimum air temperature and flow rate blown inside the cabinets can be pursued. Furthermore, a similar parametric study can be carried out for refrigerators which have different volumes. Moreover, a study which focuses to improve the insulation efficiency is necessary to elongate the off-time period.

Moreover, a CFD based numerical optimization technology can be applied to automate the design process of the position and percent distribution of air blowing holes to make the optimal results more accurate and more efficient.

Thus, the energy consumption of a refrigerator can be predicted by the outputs of these studies even at the beginning of the design stage.

REFERENCES

- [1] **Kazuhiro, F., Taichi, T., Haruko, A.**, 2003: Thermal uniformity and rapid cooling inside refrigerators, *International Journal of Refrigeration*, **26**, 249-255.
- [2] **Laguerre, O., Amara S. B., Moureh, J., and Flick, D.**, 2007: Numerical simulation of air flow and heat transfer in domestic refrigerators, *Journal of Food Engineering*, **81**, 144-156.
- [3] **Guo-Liang, D., Hong-Tao, Q., Zhi-Li, L.**, 2004: Ways to improve thermal uniformity inside a refrigerator, *Applied Thermal Engineering*, **24**, 1827-1840.
- [4] **Gupta, J.K., Gopal, M. R., and Chakraborty, S.**, 2007: Modeling of a domestic frostfree refrigerator, *International Journal of Refrigeration*, **30**, 311-322.
- [5] **Cho, S.H., Lee, I.S., Choi, J.H., and Nam, Y.S.**, PIV Measurement and Numerical Analysis of a new Refrigeration Compartment of the Refrigerator, *Core Technology Team, Digital Appliance Research Laboratory, LG Electronics Inc.*, 327-23, Gasan-Dong, Keumchen-Gu, Seoul, Korea
- [6] **Lacerda, V.T., Melo, C., Barbosa, J.R., and Duarte, P.O.O.**, 2005: Measurements of the air flow field in the freezer compartment of a top-mount no-frost refrigerator: the effect of temperature, *International Journal of Refrigeration*, **28**, 774-783.
- [7] **Lee, I.S., Baek, S.J., Chung, M.K., Rhee, D.I., Kaga, A., and Yamaguchi, K.**, 1999: A study of air flow characteristics in the refrigerator using PIV and computational simulation, *J Flow Visualization Image Process*, **6**, 333-342.
- [8] **Hoffmann, K. A., and Chiang S. T.**, Computational Fluid Dynamics, Volume 1, 4th Edition (August 2000), *A publication of engineering education system, Wichita, Kansas, USA*
- [9] **Kurt, K., Soysal, F. A.**, Arçelik Araştırma Notu, ANN-787, 2005
- [10] **Bejan, A., and Kraus, A. D.**, Heat Transfer Handbook, *John Wiley & Sons, inc.*, 2003, p531.

APPENDICES

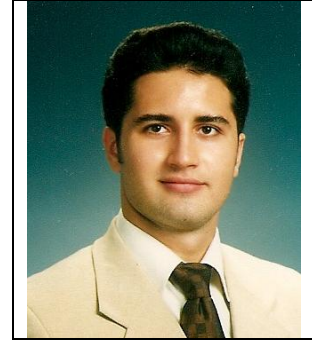
APPENDIX A.1 : Design Table for the First Row of Air Supply Holes

APPENDIX A.1

Table A.1 : Design table for the first row of air supply holes

Run Order	Row of air supply holes	Position of air supply holes & Angle of air flow	% Distribution of flow rate
1	# 1	from side holes with 25° to the sides	25
2	# 1	from side holes with 25° to the sides	40
3	# 1	from side holes with 25° to the sides	60
4	# 1	from side holes with 25° to the sides	80
5	# 1	from side holes with 25° to the sides	100
6	# 1	from side holes with 40° to the sides	25
7	# 1	from side holes with 40° to the sides	40
8	# 1	from side holes with 40° to the sides	60
9	# 1	from side holes with 40° to the sides	80
10	# 1	from side holes with 40° to the sides	100
11	# 1	from mid holes with 10° to down	25
12	# 1	from mid holes with 10° to down	40
13	# 1	from mid holes with 10° to down	60
14	# 1	from mid holes with 10° to down	80
15	# 1	from mid holes with 10° to down	100
16	# 1	from mid holes with normal to surface	25
17	# 1	from mid holes with normal to surface	40
18	# 1	from mid holes with normal to surface	60
19	# 1	from mid holes with normal to surface	80
20	# 1	from mid holes with normal to surface	100
21	# 1	from mid holes with 10° to up	25
22	# 1	from mid holes with 10° to up	40
23	# 1	from mid holes with 10° to up	60
24	# 1	from mid holes with 10° to up	80
25	# 1	from mid holes with 10° to up	100
26	# 1	from side holes with 25° to sides and from mid holes with 10° to down	25
27	# 1	from side holes with 25° to sides and from mid holes with 10° to down	40
28	# 1	from side holes with 25° to sides and from mid holes with 10° to down	60
29	# 1	from side holes with 25° to sides and from mid holes with 10° to down	80
30	# 1	from side holes with 25° to sides and from mid holes with 10° to down	100
31	# 1	from side holes with 25° to sides and from mid holes with normal to surface	25
32	# 1	from side holes with 25° to sides and from mid holes with normal to surface	40
33	# 1	from side holes with 25° to sides and from mid holes with normal to surface	60
34	# 1	from side holes with 25° to sides and from mid holes with normal to surface	80
35	# 1	from side holes with 25° to sides and from mid holes with normal to surface	100
36	# 1	from side holes with 25° to sides and from mid holes with 10° to up	25
37	# 1	from side holes with 25° to sides and from mid holes with 10° to up	40
38	# 1	from side holes with 25° to sides and from mid holes with 10° to up	60
39	# 1	from side holes with 25° to sides and from mid holes with 10° to up	80
40	# 1	from side holes with 25° to sides and from mid holes with 10° to up	100
41	# 1	from side holes with 40° to sides and from mid holes with 10° to down	25
42	# 1	from side holes with 40° to sides and from mid holes with 10° to down	40
43	# 1	from side holes with 40° to sides and from mid holes with 10° to down	60
44	# 1	from side holes with 40° to sides and from mid holes with 10° to down	80
45	# 1	from side holes with 40° to sides and from mid holes with 10° to down	100
46	# 1	from side holes with 40° to sides and from mid holes with normal to surface	25
47	# 1	from side holes with 40° to sides and from mid holes with normal to surface	40
48	# 1	from side holes with 40° to sides and from mid holes with normal to surface	60
49	# 1	from side holes with 40° to sides and from mid holes with normal to surface	80
50	# 1	from side holes with 40° to sides and from mid holes with normal to surface	100
51	# 1	from side holes with 40° to sides and from mid holes with 10° to up	25
52	# 1	from side holes with 40° to sides and from mid holes with 10° to up	40
53	# 1	from side holes with 40° to sides and from mid holes with 10° to up	60
54	# 1	from side holes with 40° to sides and from mid holes with 10° to up	80
55	# 1	from side holes with 40° to sides and from mid holes with 10° to up	100

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