

DOKUZ EYLÜL UNIVERSITY
GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES

**COST DOWN STUDY ON REFRIGERATOR
COMPONENTS BY MAINTAINING THE
SYSTEM PERFORMANCE**



by
Seda KARAGÖZ

February, 2016
İZMİR

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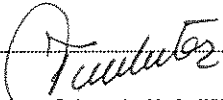
**A Thesis Submitted to the
Graduate School of Natural and Applied Sciences of Dokuz Eylül University
In Partial Fulfillment of the Requirements for the Degree of Master
of Science in Mechanical Engineering, Energy Program**

**by
Seda KARAGÖZ**

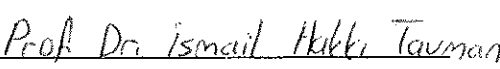
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M.Sc THESIS EXAMINATION RESULT FORM

We have read the thesis entitled "COST DOWN STUDY ON REFRIGERATOR COMPONENTS BY MAINTAINING THE SYSTEM PERFORMANCE" completed by SEDA KARAGÖZ under supervision of PROF. DR. DİLEK KUMLUTAŞ and we certify that in our opinion it is fully adequate, in scope and in quality, as a thesis for the degree of Master of Science.

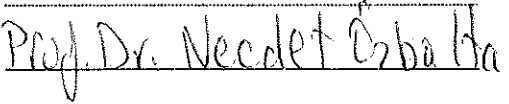

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COST DOWN STUDY ON REFRIGERATOR COMPONENTS BY MAINTAINING THE SYSTEM PERFORMANCE

ABSTRACT

The aim of this study is to determine inner design parameters of the domestic refrigerators that are important to store for long time without deterioration of the quality of food product and investigate the optimum values of these parameters. Thus, the interior air volume of a single door static type refrigerator was modeled as 3D real-like using the Computational Fluid Dynamics and Heat Transfer (CFDHT) method and analyses were made. The numerical results were validated by comparing with the experimental results and then inner design parameters were determined by using the numerical temperature distribution and flow fields of internal volume results. The calculated computational results show that the blowing angle of fan, fan position and the back wall geometry play an important role in temperature distributions and flow fields.

Keywords: Domestic refrigerators, computational fluids dynamics and heat transfer, design parameter.

MEVCUT PERFORMANS KORUNARAK BUZDOLABI BİLEŞENLERİNDE MALİYETİN DÜŞÜRÜLMESİ ÇALIŞMASI

ÖZ

Bu çalışmanın amacı gıdaların kalitesinden ödün vermeden daha uzun süre saklanması için önemli olan buzdolabı iç tasarım parametrelerinin belirlenmesi ve bu parametrelerin optimum değerlerinin incelenmesidir. Bunun için tek kapılı, statik tip bir buzdolabı Hesaplamalı Akışkanlar Dinamiği ve Isı Transferi (HADIT) yöntemi kullanılarak 3 boyutlu olarak gerçeğe yakın bir şekilde modellenmiştir. Sayısal çalışma sonuçları deneysel çalışma sonuçlarıyla kıyaslanarak doğrulanmıştır ve iç hacmin sıcaklık dağılımları ve hız dağılımları sonuçlarına bakılarak iç tasarım parametrelerine karar verilmiştir. Sayısal sonuçlar göstermektedir ki fan açısı, fan konumu ve arka duvar geometridi sıcaklık dağılımı ve hız dağılımında önemli rol oynamaktadır.

Anahtar kelimeler: Buzdolabı, hesaplamalı akışkanlar dinamiği ve ısı transferi, tasarım parametresi

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CHAPTER ONE

INTRODUCTION

1.1 Literature

Air recirculation inside a refrigerator plays an important role in keeping the consumables fresh and healthy. Since it needs a constant uniformly distributed temperature and flow field inside the cavity and this uniformity is achieved with the optimum inner design of a refrigerator.

Several studies were carried out to analyse temperature distribution in the cavity. Laguerre, Amara & Flick (2005) do an experiment on a refrigerator model in which heat is transferred by natural convection between a cold vertical wall and the other walls, which are exposed to heat losses. The air temperature profile in the boundary layers and in the central zone of the empty refrigerator model was researched. Temperature stratification in the vertical direction was observed with the cold zone at the bottom and warm zone at the top of the cavity. The impacts of temperature and the surface area of the cold wall were investigated. In order to analyse the influence of obstacles on temperature profiles, the refrigerator model was filled with 4 blocks of hollow spheres. Temperature profiles in this case were compared with the results without blocks. The air temperature is lower almost everywhere in the model containing blocks. The presence of the blocks seems to increase heat transfer particularly near the cold wall. Ding, Qiao & Lu (2004) investigated two types of refrigerators by the computational fluid dynamics (CFD) in order to enhance the temperature uniformity inside the chamber. The calculated results show that the clearance between shelves and the backwall and the clearance between the shelves and door play an important role in temperature distributions for the fully natural convective refrigerator. Temperature variation will decrease and more uniform temperature distributions will be achieved when reducing both clearances. In order to further develop the temperature uniformity, another type of directly cooled refrigerator is thrown out for consideration, in which a disturbing axial fan and an air duct are added. How to develop the airfield in this type of refrigerator is investigated, and the

calculated results show that the flow direction of air from the air duct to the chamber has apparent impacts on the temperature field and flow field. Comparison of the two types of refrigerators shows that temperature uniformity in the new type of refrigerator is better than that of the first type of refrigerator. Laguerre, Amara, Moureh & D. Flick (2007) researched heat transfer by natural convection in domestic refrigerators without ventilation. Only the refrigerating compartment was investigated for three configurations: empty refrigerator, refrigerator equipped with glass shelves and refrigerator loaded by product. Both experimental and numerical approaches were used. The simulations were carried out using CFD (computational fluid dynamic) software by taking into account or by neglecting radiation heat transfer. The following conditions were presumed: constant evaporator temperature, three-directional laminar air flow. Numerical results show temperature stratification in the refrigerating compartment (warm zone at the top and cold zone on the bottom) for all configurations. A comparison of the calculated air temperature and the experimental values shows good agreement when radiation is taken into account.

Currently, refrigeration research focuses on improving energy efficiency, reducing manufacturing cost and introducing innovative designs. The principal objective of these studies is to optimize energy consumption. Ghadiri & Rasti (2014) do an experimental study which has results of hot-wall condenser removal, condenser air cooling increment, capillary tube diameter impact, compressor cooling capacity impact, changes of R134a charge amount and ambient temperature impact on a commercial bottom mounted freezer to decrease energy consumption and production cost. The improvement of refrigerator energy consumption was investigated experimentally as a result. Investigations made under controlled conditions have shown that the more the heat removed from the condenser, the higher the efficiency can be. Results showed that by hot-wall condenser removal, condenser air cooling increment and decreasing compressor cooling capacity, energy consumption and refrigerant charge amount reduced by 23.6% and 19.3%, respectively. Bassiouny (2009) investigated the impact of the space surrounding the condenser of a household refrigerator on the rejected heat. The driving force for rejecting the heat carried by the refrigerant from the interior of a refrigerator is the temperature difference between the

condenser outer surface and surrounding air. The variation of this difference is of interest to quantify its effect because of having an insufficient space, increasing the room air temperature, or blocking this space. The results showed that having an enough surrounding space width ($s > 200$ mm) leads to a decrease in the temperature of the air flowing vertically around the condenser coil. Accordingly, this would significantly increase the amount of heat rejected. Furthermore, blocking this space retards the buoyant flow up the condenser surface, and hence increases the air temperature around the condenser. This would also decrease the heat rejected from the condenser. Sarıçay & Erbay (2008) emphasize the importance of the influences of the fin spacing and air velocity on heat transfer coefficient, cooling capacity and pressure drop of airside of the evaporator in order to increase the energy efficiency of refrigerator. The evaporator is modeled numerically and several simulations have been done to find airside thermohydraulic performance. From the numerical results, when the fin spacing is increased, the airside heat transfer coefficient is increased but heat transfer is decreased because of less total area. It has been found that as the air inlet velocity increases the heat transfer coefficient increases. The distance between fins of the evaporator is found to have considerable impact on pressure drop. The fin spacing is decreased pressure drop is increased. The most suitable fin spacing is found as 5 mm at velocity of 0,5 m/s for provide the required cooling capacity. Bansal & Chin (2002) investigated wire-and-tube condensers that are commonly used in vapour compression cycle based domestic refrigerators. A condenser was experimentally tested in a real refrigerator for some operating conditions. A simulation model was improved using the finite element and variable conductance approach, along with a combination of thermodynamic correlations. The condenser capacity per unit weight was optimised using a variety of wire and tube pitches and diameters. An optimisation factor, f_o was defined as ratio of the condenser capacity per unit weight of the optimised design and the current design. The application of this factor led to an developed design with 3% gain in capacity and 6% reduced condenser weight. Bansal, Wich & Browne (2001) carried out a parametric heat transfer study of the new “egg-crate” type evaporators in domestic refrigerators/freezers by varying fin density, fin height and fin thickness, using a steady state simulation model. Simulations were carried out over a wide range of fin parameters with finning factors ranging from 3,5 to 9. The simulation model estimated

the evaporator capacity, UA and temperature drop of air across the evaporator for varying parameters. A geometrically developed evaporator is proposed having an optimum heat transfer performance, in terms of maximum evaporator capacity per unit weight. Waltrich, Barbosa & Hermes (2011) made an investigation on the effect of an alternative evaporator design, the so-called Accelerated Flow Evaporator (AFE), on the performance of household refrigerators. In this unusual evaporator concept, the air-side cross-section area decreases with the distance from the air flow inlet, accelerating the air as it flows across the tubes and hence improving the air-side local heat transfer coefficient. An AFE heat transfer and pressure drop calculation method suggested elsewhere has been incorporated into an overall system simulation model to evaluate the effect of the evaporator geometry on the system COP. The results were compared with experimental data obtained in a topmount refrigerator. The predictions of working pressures, power consumption, cooling capacity and COP agreed with the experimental data to within 10% error bands.

Several studies were carried out to analyse fan effects in the cavity. Avcioglu (2012) carried out a literature review on this subject, which presents the experimental and modelling results (2D and 3D) to make the fan go out of use by changing design of tube on sheet evaporator. Two different evaporator was designed which has one more loop and one less loop. Results show that the design which has one more loop has similar temperature distribution with the model has no fan and the design which has one less loop has similar temperature distribution with the model has fan. Alfonso (2013) emphasize the importance of the compressor compartment, which has relatively high temperatures, due to the heat generated while the compressor is running. This indicates a larger heat transfer to the interior air volume and, as a result, more energy consumption. One way to decrease this temperature is to ventilate the base of the refrigerator with a small fan. In the research carried out, a decrease of 11 °C in the temperature of the surfaces close to the compressor was recorded, when the fan was running. The inferences of this impact can be seen at two levels: changes in the temperature distribution at the bottom of the refrigerator envelope and the resulting dynamic effects, and a decrease in energy consumption of the entire system. As a consequences, there are lower CO₂ emissions to the environment, making household

refrigerators more efficient and sustainable. Huang, Hsieh, Ke & Wang (2002) carried out a numerical study concerning the impact of frost thickness on the heat transfer performance of a four rows plate finned tube heat exchanger. Calculations are made under constant air volume and variable air volume conditions. It is found that the initial surge of heat transfer rate in the frosted finned tube heat exchanger is mainly related to the critical radius effect rather than the surface roughness. The frost thermal conductivity plays an important role in the surge phenomenon. There is hardly any initial surge when frost thermal conductivity is below $0.1 \text{ W m}^{-1} \text{ K}^{-1}$. The calculations also reveal that a centrifugal fan is recommended with a small fin-pitch heat exchanger. Nevertheless, if a long term operation at a thick frost situation is unavoidable, an axial fan should be selected. There is no great difference between selection of an axial fan or centrifugal fan for a larger fin pitch heat exchanger.

CHAPTER TWO

REFRIGERATORS

2.1 History of Refrigerators Development

The domestic refrigerator using natural ice (domestic ice box) was invented in 1803 and was used for almost 150 years without much alteration. The domestic ice box used to be made of wood with suitable insulation. Ice used to be kept at the top of the box, and low temperatures are produced in the box due to heat transfer from ice by natural convection. A drip pan is used to collect the water formed due to the melting of ice. The box has to be replenished with fresh ice once all the ice melts. Though the concept is quite simple, the domestic ice box suffered from several disadvantages. The user has to replenish the ice as soon as it is consumed, and the lowest temperatures that could be produced inside the compartment are limited. In addition, it appears that warm winters caused severe shortage of natural ice in USA. Hence, efforts, starting from 1887 have been made to develop domestic refrigerators using mechanical systems. The initial domestic mechanical refrigerators were costly, not completely automatic and were not very reliable. However, the development of mechanical household refrigerators on a large scale was made possible by the development of small compressors, automatic refrigerant controls, better shaft seals, developments in electrical power systems and induction motors. General Electric Company introduced the first domestic refrigerator in 1911, followed by Frigidaire in 1915. Kelvinator launched the domestic mechanical refrigerator in 1918 in USA. In 1925, USA had about 25 million domestic refrigerators of which only 75000 were mechanical. However, the manufacture of domestic refrigerators grew very rapidly, and by 1949 about 7 million domestic refrigerators were produced annually. With the production volumes increasing the price fell sharply (the price was 600 dollars in 1920 and 155 dollars in 1940). The initial domestic refrigerators used mainly sulphur dioxide as refrigerant. Some units used methyl chloride and methylene chloride. These refrigerants were replaced by Freon-12 in 1930s. In the beginning these refrigerators were equipped with open type compressors driven by belt drive. General Electric Company introduced the first refrigerator with a hermetic compressor in 1926. Soon

the open type compressors were completely replaced by the hermetic compressors. First refrigerators used water-cooled condensers, which were soon replaced by air cooled-condensers. Though the development of mechanical domestic refrigerators was very rapid in USA, it was still rarely used in other countries. In 1930 only rich families used domestic refrigerators in Europe. The domestic refrigerator based on absorption principle as proposed by Platen and Munters, was first made by Electrolux Company in 1931 in Sweden. In Japan the first mechanical domestic refrigerator was made in 1924. The first dual temperature (freezer-refrigerator) domestic refrigerator was introduced in 1939. The use of mechanical domestic refrigerators grew rapidly all over the world after the Second World War. Today, a domestic refrigerator has become an essential kitchen appliance not only in highly developed countries but also in countries such as India. Except very few almost all the present day domestic refrigerators are mechanical refrigerators that use a hermetic compressor and an air cooled condenser. The modern refrigerators use either HFC-134a (hydro-fluoro-carbon) or iso-butane as refrigerant.

Domestic refrigerators are widely used in industrialized countries. Three types of domestic refrigerators are available in the market: static, brewed, and no-frost. The static type (Figure 2.1a) is widely used in Europe. In this case, heat is transferred principally by natural convection and airflow is due to variations in air density. These variations are related principally to the temperature and humidity gradients. The vertical force which results from air weight and buoyancy is ascendant if air is locally lighter than the average and descendant where the opposite is true (hot/humid air is lighter than cold/dry air). There is a combination of heat transfer and airflow inside refrigerators i.e., heat transfer from air to the evaporator (vertical plate), to the other walls (cavity) and to products (various forms). Due to the principal of heat transfer, temperature heterogeneity is often observed in this type of refrigerator. The position of the evaporator (horizontal/vertical, top/bottom of the compartment) determines the location of cold and warm zones. The brewed type is a static refrigerator equipped with a fan (Figure 2.1b). It allows air circulation and the temperature decreases rapidly after door opening. Air temperature is more homogeneous in this case than in the static type but the energy consumption is higher due to the fan. In a no-frost refrigerator (Figure 2.1c), a fan (embedded in the back wall) pushes air to flow over the evaporator before

entering into the refrigerating compartment. Air temperature is more homogeneous compared to the two other refrigerator types. Disadvantages of no-frost type are noise, energy consumption, drying on food surface and high price.

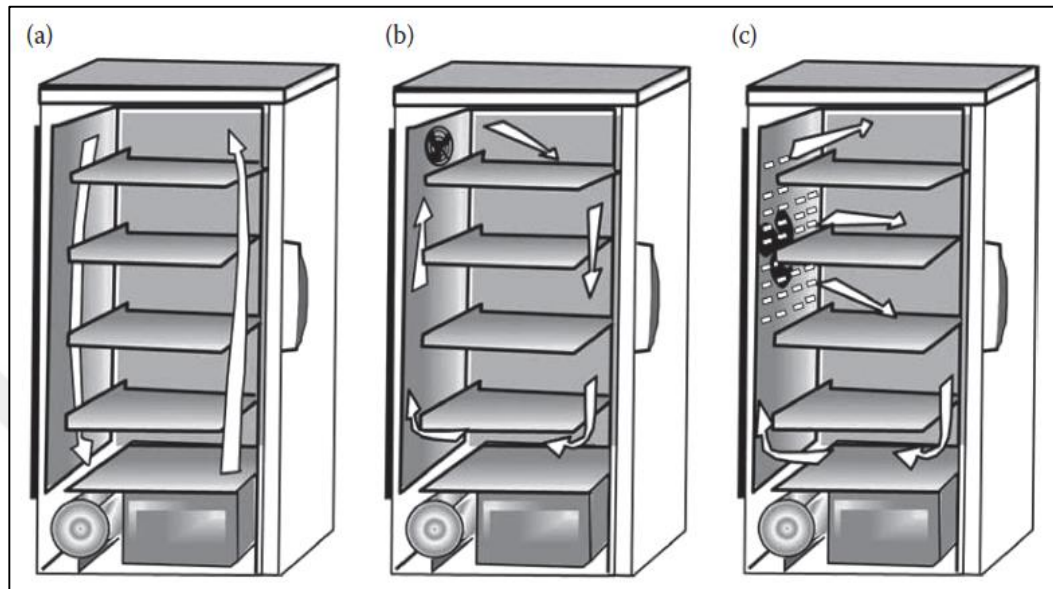


Figure 2.1 Three types of refrigerator: (a) static, (b) brewed, (c) no-frost (Farid, 2010)

Knowledge of air temperature and velocity profiles in a refrigerator is important for food quality control. In fact, if the consumer knows the position of warm and cold zones in the refrigerator, the product can be placed correctly. Knowledge of thermal and hydrodynamic boundary layers near the evaporator and the other walls is also important. If the product is too close to the evaporator wall, freezing can occur, and if it is too close to the other walls, there may be health risks.

2.2 Air Flow Recirculation in a Refrigerator

Air recirculation inside a refrigerator plays an important role in keeping the consumables fresh and healthy. Since it needs a constant temperature inside the cavity, the temperature needs to be uniformly distributed. Air is the ultimate fluid which is used as source to keep the consumables cold. Hence airflow plays a crucial role in determining the efficiency and cooling of the refrigerators.

Whenever an air flows it generates a resistance. The resistance results into two different forms of flows:

Laminar Flow: Flow of the fluid when each particle of the fluid follows a smooth path.

Turbulent Flow: Irregular flow that is characterised by tiny whirlpool regions is called turbulent flow.

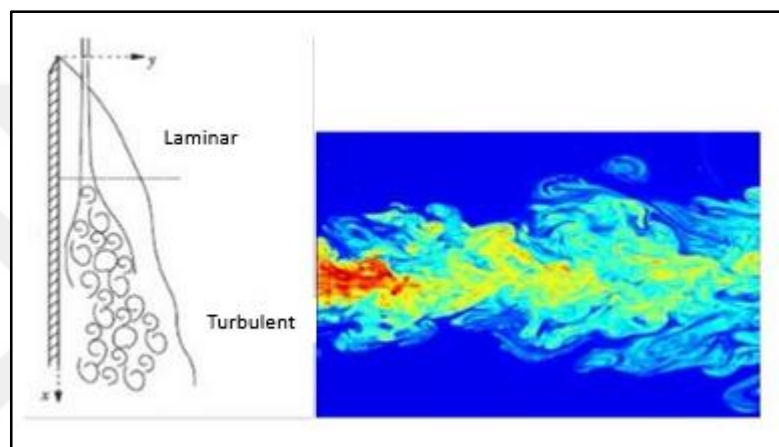


Figure 2.2 The laminar and turbulent flow of air (Farid, 2010)

As shown in the Figure 2.2, flow for vertical plate (either cold or warm) the region of the laminar flow is close to the region of inlet and the boundary. However as we move further from the inlet and the boundary the flow starts to become turbulent. Similar case is within the refrigerator. Depending on the wall being cold the flow is laminar near the walls and turbulent around the region of high velocity air flow where cold air tries to replace the hot air.

2.3 Cold Production System in Domestic Refrigerators/Freezers

The most common refrigerators and freezers have four major parts in their refrigeration system a compressor, a condenser, an expansion valve and an evaporator (Figure 2.3). In the evaporator section, a refrigerant (commonly R600a and R134; still in use R12 and ammonia) is vaporized to absorb heat added into the refrigerator due

to heat transfer across the refrigerators walls and infiltration through the door and seals and during door opening. The refrigerant boils at -18 to -20° C when pressurized at 0.9–1 bar, so the evaporator temperature is maintained at or near that temperature if the appliance is working correctly. In the next stage, an electric motor runs a small piston compressor and the refrigerant is pressurized. This raises the temperature of the refrigerant and the resulting superheated, high-pressure gas (it is still a gas at this point) is then condensed to a liquid in an aircooled condenser. In most refrigerators and freezers, the compressor is in the base and the condenser coils are at the rear of the appliance. From the condenser, the liquid refrigerant flows through an expansion valve (almost always a capillary tube), in which its pressure and temperature are reduced and these conditions are maintained in the evaporator. The whole process operates continuously, by transferring heat from the evaporator section (inside the refrigerator) to the condenser section (outside the refrigerator), by pumping refrigerant continuously through the system described above. When the desired temperature is reached, the pump stops and so does heat transfer.

The refrigerator/freezers may be equipped with one or two compressors. In the case of one compressor, the operating cycle is both controlled by the air temperature in the refrigerating compartment and in freezer. In the case of two compressors, each operating cycle is independently controlled by the air temperature in the refrigerating compartment and in the freezer. The temperature in each compartment is, therefore, better regulated but the price is higher.

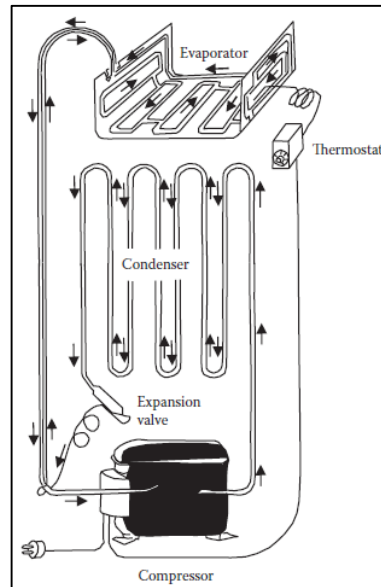


Figure 2.3 Cold production system in a domestic refrigerator/freezer (Farid, 2010)

2.4 Refrigerator Components

2.4.1 Refrigeration Compressors

A refrigeration compressor is the heart of a vapor compression system. It raises the pressure of refrigerant so that it can be condensed into liquid, throttled, and evaporated into vapor to produce the refrigeration effect. It also provides the motive force to circulate the refrigerant through condenser, expansion valve, and evaporator.

According to the compression process, refrigeration compressors can be divided into positive displacement and nonpositive displacement compressors. A positive displacement compressor increases the pressure of the refrigerant by reducing the internal volume of the compression chamber. Reciprocating, scroll, rotary, and screw compressors are all positive displacement compressors. The centrifugal compressor is the only type of nonpositive displacement refrigeration compressor widely used in refrigeration systems today.

Based on the sealing of the refrigerant, refrigeration compressors can be classified as

- Hermetic compressors, in which the motor and the compressor are sealed or welded in the same housing to minimize leakage of refrigerant and to cool the motor windings by using suction vapour

- Semihermetic compressors, in which motor and compressor are enclosed in the same housing but are accessible from the cylinder head for repair and maintenance

- Open compressors, in which compressor and motor are enclosed in two separate housings

Currently used refrigeration compressors are reciprocating, scroll, screw, rotary, and centrifugal compressors.

2.4.1.1 Reciprocating Compressors

In a reciprocating compressor, as shown in Figure 2.4a, a crankshaft connected to the motor shaft drives 2, 3, 4, or 6 single-acting pistons moving reciprocally in the cylinders via a connecting rod.

The refrigeration capacity of a reciprocating compressor is a fraction of a ton to about 200 tons. Refrigerants R-22 and R-134a are widely used in comfort and processing systems and sometimes R-717 in industrial applications. The maximum compression ratio R_{com} for a single-stage reciprocating compressor is about 7. Volumetric efficiency η_v drops from 0.92 to 0.65 when R_{com} is raised from 1 to 6. Capacity control of reciprocating compressor including: on-off and cylinder unloader in which discharge gas is in short cut and return to the suction chamber.

Although reciprocating compressors are still widely used today in small and medium-sized refrigeration systems, they have little room for significant improvement and will be gradually replaced by scroll and screw compressors.

2.4.1.2 Scroll Compressors

A scroll compressor consists of two identical spiral scrolls assembled opposite to each other, as shown in Figure 2.4b. One of the scrolls is fixed, and the other moves in an orbit around the motor shaft whose amplitude equals the radius of the orbit. The two scrolls are in contact at several points and therefore form a series of pockets.

Vapor refrigerant enters the space between two scrolls through lateral openings. The lateral openings are then sealed and the formation of the two trapped vapor pockets indicates the end of the suction process. The vapor is compressed and the discharge process begins when the trapped gaseous pockets open to the discharge port. Compressed hot gas is then discharged through this opening to the discharge line. In a scroll compressor, the scrolls touch each other with sufficient force to form a seal but not enough to cause wear.

The upper limit of the refrigeration capacity of currently manufactured scroll compressors is 60 tons. A scroll compressor has $\eta_v > 95\%$ at $R_{com} = 4$ and $\eta_{isen} = 80\%$. A scroll compressor also has only about half as many parts as a reciprocating compressor at the same refrigeration capacity. Few components result in higher reliability and efficiency. Power input to the scroll compressor is about 5 to 10% less than to the reciprocating compressor. A scroll compressor also operates more smoothly and is quieter.

2.4.1.3 Rotary Compressors

Small rotary compressors for room air conditioners and refrigerators have a capacity up to 4 tons. There are two types of rotary compressors: rolling piston and rotating vane. A typical rolling piston rotary compressor is shown in Figure 2.4c. A rolling piston mounted on an eccentric shaft is kept in contact with a fixed vane that slides in a slot. Vapor refrigerant enters the compression chamber and is compressed by the eccentric motion of the roller. When the rolling piston contacts the top housing, hot gas is squeezed out from the discharge valve.

2.4.1.4 Screw Compressors

These are also called helical rotary compressors. Screw compressors can be classified into single-screw compressors, in which there is a single helical rotor and two star wheels, and twin-screw compressors. Twin-screw compressors are widely used.

A typical twin-screw compressor, as shown in Figure 2.4d consists of a four-lobe male rotor and a six-lobe female rotor, a housing with suction and discharge ports, and a sliding valve to adjust the capacity during part load. Normally, the male rotor is the driver. Twin-screw compressors are often direct driven and of hermetic type.

Vapor refrigerant is extracted into the interlobe space when the lobes are separated at the suction port. During the successive rotations of the rotor, the volume of the trapped vapor is compressed. When the interlobe space is in contact with the discharge port, the compressed hot gas discharges through the outlet. Oil injection effectively cools the rotors and results in a lower discharge temperature. Oil also provides a sealing effect and lubrication. A small clearance of 0.0005 in. as well as the oil sealing minimizes leakage of the refrigerant.

The refrigeration capacity of twin-screw compressors is 50 to 1500 tons. The compression ratio of a twin-screw compressor can be up to 20:1. R-22 and R-134a are the most widely used refrigerants in comfort systems. In a typical twin-screw compressor, h_v decreases from 0.92 to 0.87 and h_{isen} drops from 0.82 to 0.67 when R_{com} increases from 2 to 10. Continuous and stepless capacity control is provided by moving a sliding valve toward the discharge port, which opens a shortcut recirculating passage to the suction port.

Twin-screw compressors are more efficient than reciprocating compressors. The low noise and vibration of the twin-screw compressor together with its positive displacement compression results in more applications today.

2.4.1.5 Centrifugal Compressors

A centrifugal compressor is a turbomachine and is similar to a centrifugal fan. A hermetic centrifugal compressor has an outer casing with one, two, or even three impellers internally connected in series and is driven by a motor directly or by a gear train. At the entrance to the first-stage impeller are inlet guide vanes positioned at a specific opening to adjust refrigerant flow and therefore the capacity of the centrifugal compressor.

Figure 2.4e shows a two-stage hermetic centrifugal compressor. The total pressure rise in a centrifugal compressor, often called head lift, in psi, is due to the conversion of the velocity pressure into static pressure. Although the compression ratio R_{com} of a single-stage centrifugal compressor using R-123 and R-22 seldom exceeds 4, two or three impellers connected in series satisfy most of the requirements in comfort systems.

Because of the high head lift to raise the evaporating pressure to condensing pressure, the discharge velocity at the exit of the second-stage impeller approaches the acoustic velocity of saturated vapor v_{ac} of R-123, 420 ft/sec at atmospheric pressure and a temperature of 80°F. Centrifugal compressors need high peripheral velocity and rotating speeds (up to 50,000 rpm) to produce such a discharge velocity. It is not economical to manufacture small centrifugal compressors. The available refrigeration capacity for centrifugal compressors ranges from 100 to 10,000 tons. Centrifugal compressors have higher volume flow per unit refrigeration capacity output than positive displacement compressors. Centrifugal compressors are efficient and reliable. Their volumetric efficiency almost equals 1. At design conditions, their η_{isen} may reach 0.83, and it drops to 0.6 during part-load operation. They are the most widely used refrigeration compressors in large air-conditioning systems.

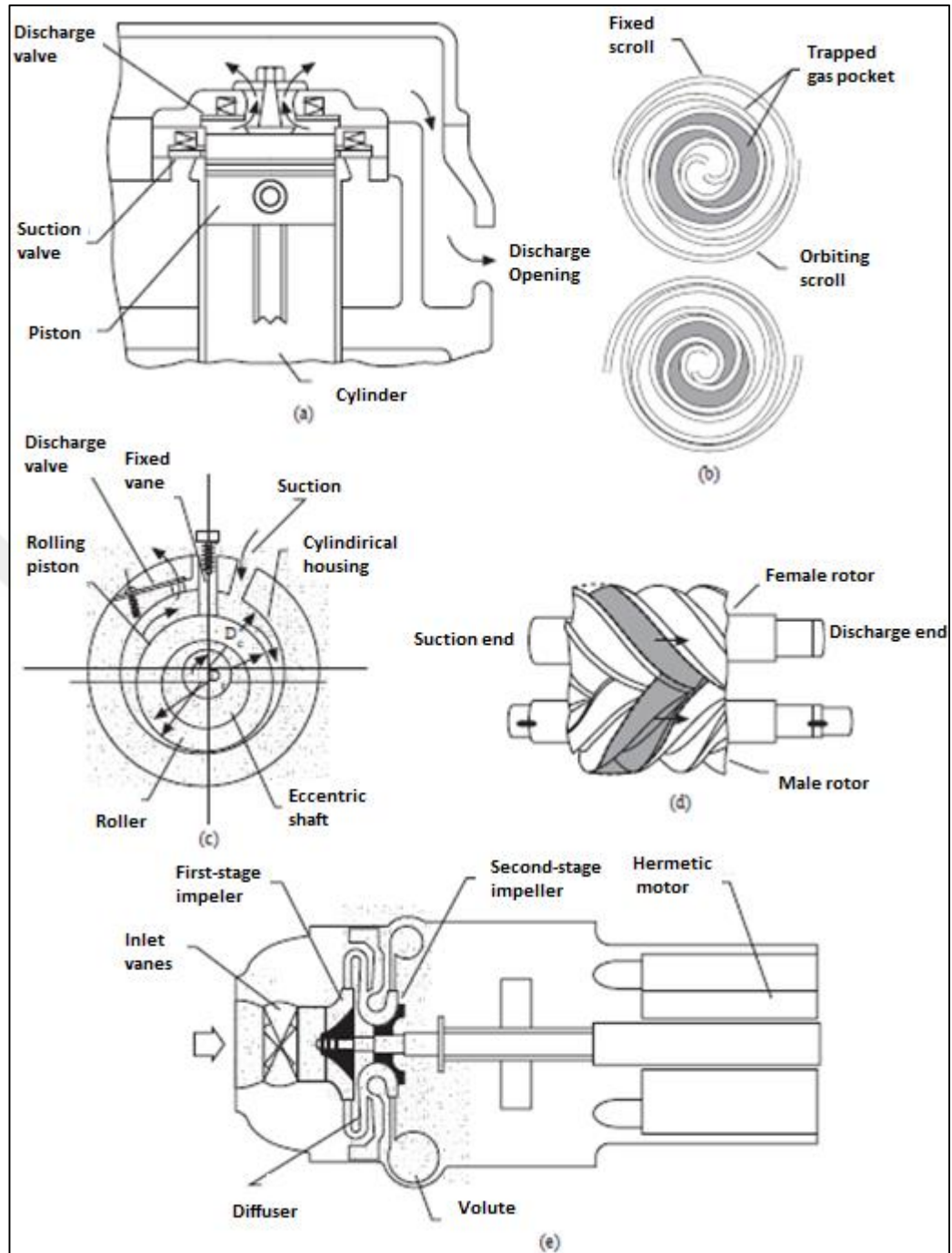


Figure 2.4 Various types of refrigeration compressors: (a) reciprocating, (b) scroll, (c) rotary, (d) twin-screw, and (e) centrifugal (Wang & Lavan, 1999)

2.4.2 Refrigeration Condensers

A refrigeration condenser or simply a condenser is a heat exchanger in which hot gaseous refrigerant is condensed into liquid and the latent heat of condensation is rejected to the atmospheric air, surface water, or well water. In a condenser, hot gas is first desuperheated, then condensed into liquid, and finally subcooled.

2.4.2.1 Air-Cooled Condensers

In an air-cooled condenser, air is used to absorb the latent heat of condensation released during desuperheating, condensation, and subcooling.

An air-cooled condenser consists of a condenser coil, a subcooling coil, condenser fans, dampers, and controls as shown in Figure 2.5a. There are refrigeration circuits in the condensing coil. Condensing coils are usually made of copper tubes and aluminum fins. The diameter of the tubes is 1/4 to 3/4 in., typically 3/8 in., and the fin density is 8 to 20 fins/in. On the inner surface of the copper tubes, microfins, typically 60 fins/in. with a height of 0.008 in., are used. A condensing coil usually has only two to three rows due to the low pressure drop of the propeller-type condenser fans. A subcooling coil is located at a lower level and is connected to the condensing coil.

Hot gas from the compressor enters the condensing coil from the top. When the condensate increases, part of the condensing area can be used as a subcooling area. A receiver is necessary only when the liquid refrigerant cannot all be stored in the condensing and subcooling coils during the shut-down period in winter.

Cooling air is drawn through the coils by a condenser fan(s) for even distribution. Condenser fans are often propeller fans for their low pressure and large volume flow rate. A damper(s) may be installed to adjust the volume flow of cooling air.

In air-cooled condensers, the volume flow of cooling air per unit of total heat rejection is 600 to 1200 cfm/ton of refrigeration capacity at the evaporator, and the

optimum value is about 900 cfm/ton. The corresponding cooling air temperature difference — cooling air leaving temperature minus outdoor temperature ($T_{ca,l} - T_o$) — is around 13°F.

The condenser temperature difference (CTD) for an air-cooled condenser is defined as the difference between the saturated condensing temperature corresponding to the pressure at the inlet and the air intake temperature, or ($T_{con,i} - T_o$). Air-cooled condensers are rated at a specific CTD, depending on the evaporating temperature of the refrigeration system T_{ev} in which the air-cooled condenser is installed. For a refrigeration system having a lower T_{ev} , it is more economical to equip a larger condenser with a smaller CTD. For a comfort air-conditioning system having a T_{ev} of 45°F, CTD = 20 to 30°F.

A higher condensing temperature T_{con} , a higher condensing pressure p_{con} , and a higher compressor power input may be due to an undersized air-cooled condenser, lack of cooling air or low value, a high entering cooling air temperature at the roof, a dirty condensing coil, warm air circulation because of insufficient clearance between the condenser and the wall, or a combination of these. The clearance should not be less than the width of the condensing coil.

If p_{con} drops below a certain value because of a lower outdoor temperature, the expansion valve in a reciprocating vapor compression system may not operate properly. At a low ambient temperature T_o , the following controls are often used:

Duty cycling, turning the condenser fans on and off until all of them are shut down, to reduce cooling air volume flow

Modulating the air dampers to reduce the volume flow

Reducing the fan speed

Some manufacturers' catalogs start low ambient control at $T_o = 65^\circ\text{F}$ and some specify a minimum operating temperature at $T_o = 0^\circ\text{F}$.

2.4.2.2 Water-Cooled Condensers

In a water-cooled condenser, latent heat of condensation released from the refrigerant during condensation is extracted by water. This cooling water, often called condenser water, is taken directly from river, lake, sea, underground well water or a cooling tower.

Two types of water-cooled condensers are widely used for air-conditioning and refrigeration: doubletube condensers and horizontal shell-and-tube condensers.

A double-tube condenser consists of two tubes, one inside the other. Condenser water is pumped through the inner tube and refrigerant flows within the space between the inner and outer tubes in a counterflow arrangement. Because of its limited condensing area, the double-tube condenser is used only in small refrigeration systems.

A horizontal shell-and-tube water-cooled condenser using halocarbon refrigerant usually has an outer shell in which copper tubes typically 5/8 to 3/4 in. in diameter are fixed in position by tube sheets as shown in Figure 2.5b. Integral external fins of 19 to 35 fins/in. and a height of 0.006 in. and spiral internal grooves are used for copper tubes to increase both the external and the inner surface area and their heat transfer coefficients.

Hot gas from the compressor enters the top inlet and is distributed along the baffle to fill the shell. Hot gas is then desuperheated, condensed, subcooled into liquid, and discharged into the liquid line at the bottom outlet. Usually one sixth of the volume is filled with subcooled liquid refrigerant. Subcooling depends on the entering temperature of condenser water T_{ce} , in $^\circ\text{F}$, and usually varies between 2 and 8 $^\circ\text{F}$.

Condenser water enters the condenser from the bottom for effective subcooling. After extracting heat from the gaseous refrigerant, condenser water is discharged at a higher level. Two-pass or three-pass water flow arrangements are usually used in shell-and-tube water-cooled condensers. The two-pass arrangement means that water flows from one end to the opposite end and returns to the original end. Two-pass is the standard setup. In a shell-and-tube water-cooled condenser, the condensing temperature T_{con} depends mainly on the entering temperature of condenser water T_{ce} , the condenser area, the fouling factor, and the configuration of the copper tube.

2.4.2.3 Evaporative Condenser

An evaporative condenser uses the evaporation of water spray on the outer surface of the condensing tubes to remove the latent heat of condensation of refrigerant during condensation.

An evaporative condenser consists of a condensing coil, a subcooling coil, a water spray, an induced draft or sometimes forced draft fan, a circulating water pump, a water eliminator, a water basin, an outer casing, and controls as shown in Figure 2.5c. The condensing coil is usually made of bare copper, steel, or sometimes stainless steel tubing.

Water is sprayed over the outside surface of the tubing. The evaporation of a fraction of condenser water from the saturated air film removes the sensible and latent heat rejected by the refrigerant. The wetted outer surface heat transfer coefficient h_{wet} is about four or five times greater than the dry surface heat transfer coefficient h_o , in Btu/hr.ft².°F. The rest of the spray falls and is collected by the basin. Air enters from the inlet just above the basin. It flows through the condensing coil at a face velocity of 400 to 700 fpm, the water spray bank, and the eliminator. After air absorbs the evaporated water vapor, it is extracted by the fan and discharged at the top outlet. The water circulation rate is about 1.6 to 2 gpm/ton, which is far less than that of the cooling tower.

An evaporative condenser is actually a combination of a water-cooled condenser and a cooling tower. It is usually located on the rooftop and should be as near the compressor as possible. Clean tube surface and good maintenance are critical factors for evaporative condensers. An evaporative condenser also needs low ambient control similar as in an air-cooled condenser.

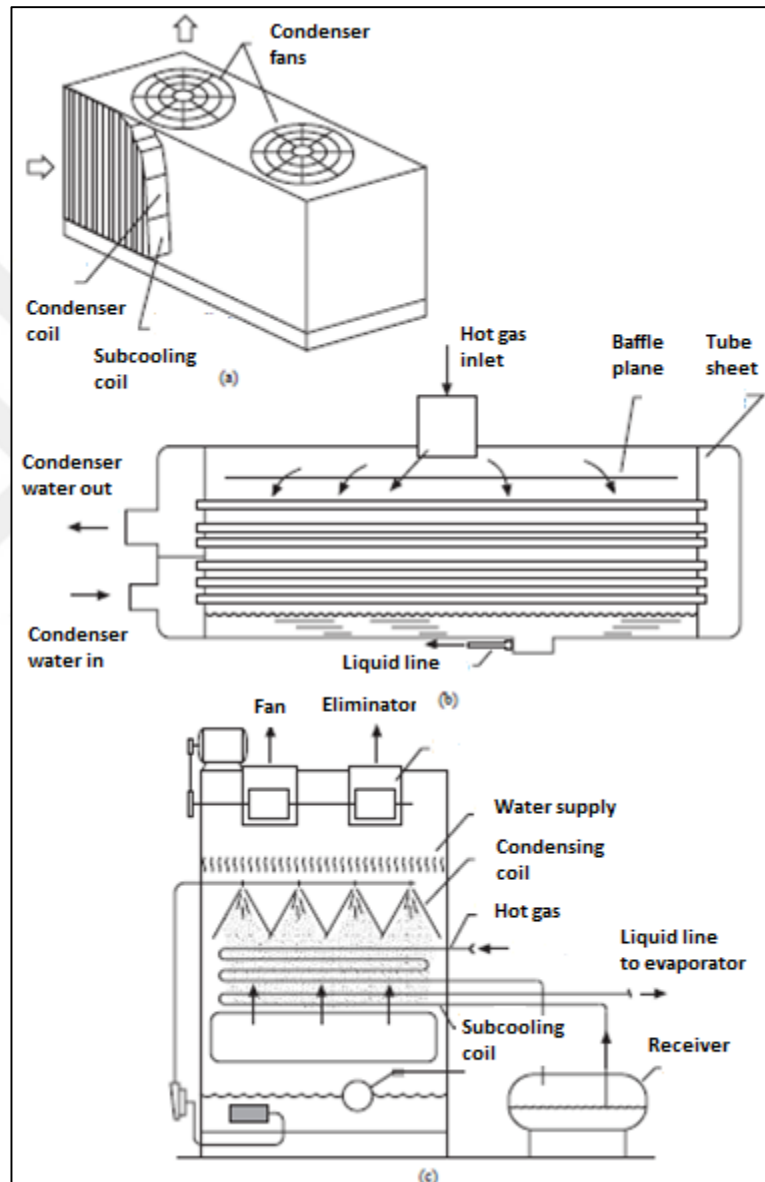


Figure 2.5 Various types of refrigeration condensers: (a) air-cooled, (b) shell-and-tube water cooled, and (c) evaporative cooled (Wang & Lavan, 1999)

2.4.3 Evaporator

The evaporator is one of the main components of a refrigeration system, in which refrigerant evaporates for the purpose of extracting heat from the surrounding air, chilled water, or other substances. In vapor compression refrigeration systems, the evaporator is also an indirect-contact heat exchanger.

Evaporators can be classified into three categories, depending on the medium or substance to be cooled:

- An air cooler is an evaporator that cools the air directly in a refrigerated space or piece of equipment (such as a packaged unit). Conditioned air is then distributed through air distribution systems. In an air cooler, the refrigerant flows inside the metal tubes or finned tubes while air flows over them.
- In a liquid cooler, chilled water is cooled to a lower temperature and is pumped to remote airhandling units, fan coils, or other terminals for air conditioning or other applications.
- An evaporator can be used to produce ice directly, such as an ice maker in an ice harvester of an ice storage system.

A liquid cooler is different from a chiller. A liquid cooler is an evaporator, a component of a refrigeration system, whereas a chiller is a refrigeration package to produce chilled water.

Refrigerant feed for air and liquid coolers and ice makers can be classified mainly into the following three types:

- Dry-expansion or direct-expansion (DX). In evaporators with dry- or direct expansion refrigerant feed, liquid refrigerant is fed through an expansion valve and a distributor, flows inside the tubes in a finned coil, the evaporator, and is

completely vaporized and superheated to a certain degree before reaching the exit of the evaporator.

- Flooded refrigerant feed. In evaporators with flooded refrigerant feeds, liquid refrigerant is fed through a throttling device and vaporizes outside the tubes within a shell. The refrigerant-side surface area is always wetted by the liquid refrigerant, which results in a higher surface heat-transfer coefficient.
- Liquid overfeed. In liquid overfeed evaporators, liquid refrigerant is fed by a mechanical or gas pump and is then overfed to each evaporator. The inner surface in an overfeed evaporator is also wetted by liquid refrigerant.

The various combinations of cooler types and refrigerant feeds make up the following evaporator groups:

- DX cooler, which is a DX coil, acts as an air cooler with direct-expansion refrigerant feed
- Flooded shell-and-tube liquid cooler, or simply flooded liquid cooler
- Shell-and-tube liquid cooler with direct-expansion refrigerant feed, or simply direct-expansion liquid cooler
- Liquid overfeed cooler
- Direct-expansion ice maker

Among these, DX coils are most widely used. Flooded liquid coolers are widely used in large chillers, and direct-expansion liquid coolers are often used in reciprocating or screw chillers. Liquid overfeed coolers are sometimes adopted for processing air conditioning systems. Ice makers are usually used in thermal storage systems and for industrial applications.

2.4.4 Throttling Devices

We have seen that refrigeration is brought about by evaporator of the liquid refrigeration in the evaporator. The latent heat for vaporization of the liquid is supplied by the substance (air or liquid that to be cooled) in contact with the evaporator surface. The substance falls down more and more, a lesser amount of refrigerant need be vaporized in the evaporator. In other words, as the temperature of the substance falls down and nears the designed level, the load on the refrigeration system decreases. While for full load, sufficient refrigerant liquid should be fed into the evaporator, the rate of flow should get reduced as the load comes down to prevent overfeeding of the evaporator and liquid carry over (or liquid flood back) to the compressor. Thus the rate of flow of the fluid refrigerant to the evaporator has to be varied according to the load on the system.

Further the pressure of the liquid refrigerant at the high pressure liquid refrigerant falls in passing through the orifice, due to friction.

These two functions are performed by the throttling device. The throttling device is fixed at the end of the liquid line and the outlet of the device is at the inlet of the evaporator.

Every throttling device has an orifice (small opening). The pressure of the high pressure liquid refrigerant falls in passing through the orifice, due to friction.

The rate of flow of liquid through the throttling device depends on the size and opening of the orifice. It also depends on the pressure difference between the liquid line and the evaporator pressure. For the same orifice, the flow-rate will increase with an increase in the pressure difference and vice versa.

As the temperature of the substance being cooled falls (on reduction of load), the evaporator refrigerant pressure will also come down; and with the same condensing pressure, the pressure difference will increase. As a result, the flow rate will increase

if the orifice opening is kept constant. Therefore, the opening of the orifice has to be reduced to reduce the rate of flow of refrigerant in accordance with the load.

There are four types of expansion device widely used in commercial and industrial refrigeration:

- 1) Capillary tubes
- 2) Thermostatic, electronic or balanced port expansion valves
- 3) Float valves (high and low side)
- 4) Hand expansion valve and level switch

2.4.5 Refrigerants

A refrigerant is the primary working fluid used for absorbing and transmitting heat in a refrigeration system. All substances that exist in liquid and vapor states absorb heat during evaporation and could therefore be used as refrigerants. Water, for instance, could even be used, but its boiling point is too high to be of practical use. A refrigerant should evaporate at the required cooling temperature at a reasonable pressure, and must be able to be condensed by a readily available cooling medium (usually ambient air) at a practical pressure.

Refrigerants used in refrigeration and air-conditioning are usually one of:

- 1) CFC – chlorofluorocarbons
- 2) HCFCs – hydro chlorofluorocarbons
- 3) HFCs- hydro fluorocarbons

4) HCs – hydrocarbons

5) NH₃ – ammonia

CFCs deplete stratospheric zone and following the Montreal Protocol, are no longer produced. They were very widely used in the past, and are therefore still in older systems. HCFCs also deplete ozone, but to a lesser extent HFCs have zero ozone depletion potential.

Ammonia NH₃ is used extensively in large industrial refrigeration plant. Ammonia has a characteristic smell even in small concentrations in air. It cannot burn, but is explosive when mixed with air in a volume percentage 13-28. Because of corrosion, copper or copper alloys must not be used in ammonia plant.

The temperatures of the refrigerant in the evaporator and condenser are the temperatures of the cold and warm regions, respectively, with which the system interacts thermally. This, in turn, determines the operating pressures in the evaporator and condenser. Consequently, the selection of a refrigerant is based partly on the suitability of its pressure-temperature relationship in the range of the particular application. It is generally desirable to avoid excessively low pressures in the evaporator and excessively high pressures in the condenser. Other considerations in refrigerant selection include chemical stability, toxicity, corrosiveness, and cost. The type of compressor used also affects the choice of refrigerant. Centrifugal compressors are best suited for low evaporator pressures and refrigerants with large specific volumes at low pressure. Reciprocating compressors perform better over large pressure ranges and are better able to handle low specific volume refrigerants. Other considerations in refrigerant selection include chemical stability, toxicity, corrosiveness, and cost.

2.4.5.1 Types of Refrigerants

A refrigerant can either be a single chemical compound or a mixture (blend) of multiple compounds.

1) Azeotropic - These are blends of multiple components of volatilities (refrigerants) that evaporate and condense as a single substance and do not change their volumetric composition or saturation temperature when they evaporate or condense at a constant pressure. The word "azeotropic" means that the refrigerant will be found in the same concentration over the whole plant. Components in a mixture of azeotropes cannot be separated from their constituents by distillation. Properties of azeotropic refrigerants are entirely different from those of their components and may be conveniently treated as a single chemical compound. R 502: An azeotropic mixture of refrigerants R 22 and R 115 (CClF₂CF₃).

2) Near Azeotropic - Near-azeotropic refrigerants are blends whose characteristics are near to azeotropic. Although properties of near-azeotropic refrigerants are nearer to azeotropic than to nonazeotropic (zeotropic), near-azeotropic refrigerants are defined as zeotropic or nonazeotropic.

3) Zeotropic - These are blends of multiple components of volatilities (refrigerants) that evaporate and condense as a single substance and do change volumetric composition or saturation temperature when they evaporate or condense at a constant pressure. It is recommended to check with the manufacturers of the refrigerant before using a zeotropic blend in a flooded evaporator system as it may affect the system's performance.

4) Blends - Mixtures of refrigerants of two or more chemical compounds are blends. The advantage of a blend of multiple chemical compounds compared to a single compound is that the required properties of the blend can possibly be achieved by varying the fractional composition of the components.

5) Glide - Zeotropic mixtures, including near-azeotropic blends, show changes in composition because of the leaks, the difference between liquid and vapor phases, or the difference between the charge and circulation, or their combined effect. The shift in composition causes the change in evaporating and condensing temperature and pressure. The difference in dew point and bubble point in the temperature-

concentration diagram of a zeotropic refrigerant during evaporation and condensation is called glide, expressed in °F (°C). A near-azeotropic refrigerant has a smaller glide than a zeotropic one. The midpoint between the dew point and bubble point is usually taken as the evaporating or condensing temperature for a non-azeotropic and near-azeotropic refrigerant.

2.4.5.2 Selection of a Refrigerant

A wide variety of refrigerants are commercially available for use in vapor-compression systems. Selection of a refrigerant is based on several performance characteristics that assist in determining the refrigerant's suitability for a given system. The following is a list of important characteristics that are usually considered:

1. Latent heat of vaporization. A high latent heat of vaporization is preferred. For a given capacity, a high value of latent heat For domestic refrigerators, R-600 (n-butane) and R134a are suitable as refrigerants can be used. of vaporization indicates that a smaller amount of refrigerant will be circulated per unit of time.
2. Condensing pressure. Excessively high condensing pressure requires considerable expenditure on heavy construction of condenser and piping.
3. Freezing temperature. The freezing temperature of the refrigerant should be below the evaporator temperature.
4. Critical temperature. The refrigerant should have sufficiently high critical temperature. At temperatures above the critical temperature, the refrigerant vapor cannot be liquefied. Particularly in the case of air-cooled condensers, the critical temperature should be above the highest ambient temperature expected.
5. Toxicity. In many applications, including air conditioning systems, the refrigerant must be nontoxic.

6. Flammability. The refrigerant should be nonflammable.
7. Corrosiveness. The refrigerant should not be corrosive to the materials used in the construction of the refrigeration system.
8. Chemical stability. The refrigerant must be chemically stable.
9. Detection of leaks. If a leak develops in the refrigeration system, the detection of such a leak should be easy.
10. Cost. Low-cost refrigerant is preferred in industrial applications.
11. Environmental impact. The refrigerant released from the refrigeration systems due to leaks should not cause environmental damage.

Table 2.1 Comparison between Commonly Used Refrigerants (Performance Based on -15°C (Singh & Heldman, 1984)

Chemical formula	Freon 12 (dichlorodifluoro- methane, CCl_2F_2)	Freon 22 (monochlorodi- fluoro-methane, CHClF_2)	HFC 134a (CH_2FCF_3)	Ammonia (NH_3)
Molecular weight	120.9	86.5	102.3	17.0
Boiling point ($^{\circ}\text{C}$) at 101.3 kPa	-29.8	-40.8	-26.16	-33.3
Evaporator pressure at -15°C (kPa)	182.7	296.4	164.0	236.5
Condensing pressure at 30°C (kPa)	744.6	1203.0	770.1	1166.5
Freezing point ($^{\circ}\text{C}$) at 101.3 kPa	-157.8	-160.0	-96.6	-77.8
Critical temperature ($^{\circ}\text{C}$)	112.2	96.1	101.1	132.8
Critical pressure (kPa)	4115.7	4936.1	4060	11423.4
Compressor discharge temperature ($^{\circ}\text{C}$)	37.8	55.0	43	98.9
Compression ratio ($30^{\circ}\text{C}/-15^{\circ}\text{C}$)	4.07	5.06	4.81	4.94
Latent heat of vaporization at -15°C (kJ/kg)	161.7	217.7	209.5	1314.2
Horsepower/ton refrigerant, ideal	1.002	1.011	1.03	0.989
Refrigerant circulated/ton refrigeration (kg/s), ideal	2.8×10^{-2}	2.1×10^{-2}	2.38×10^{-2}	0.31×10^{-2}
Compressor displacement/ton refrigeration (m^3/s)	2.7×10^{-3}	1.7×10^{-3}	2.2×10^{-3}	1.6×10^{-3}
Stability (toxic decomposition products)	Yes	Yes	No (at high temperatures)	No
Flammability	None	None	None	Yes
Odor	Ethereal	Ethereal	Ethereal	Acrid

For domestic refrigerators, R-600 (n-butane) and R134a are suitable as refrigerants can be used.

2.5 Domestic Refrigerating System Basics and Theories

Temperature plays an important role in maintaining the quality of stored food products. In earlier days, a lower temperature was obtained by the use of ice. Ice was allowed to melt in an insulated chamber that contained food products (Figure 2.6). During melting, ice requires latent heat (333.2 kJ/kg) to be converted from the solid phase to liquid water. This heat was extracted from the product that was kept next to ice in an insulated chamber. Today, the cooling process is achieved by the use of a mechanical refrigeration system. Refrigeration systems allow transfer of heat from the cooling chamber to a location where the heat can easily be discarded. The transfer of

heat is accomplished by using a refrigerant, which like water changes state—from liquid to vapor.

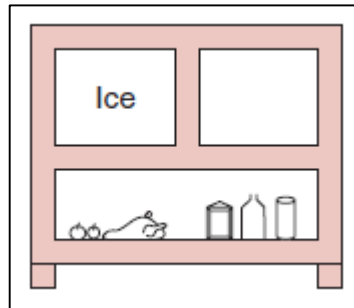


Figure 2.6 An ice box (Singh & Heldman, 1984)

2.5.1 The Basic Refrigeration Cycle

Refrigeration is defined as the process of extracting heat from a lower-temperature heat source, substance, or cooling medium and transferring it to a higher-temperature heat sink. Refrigeration maintains the temperature of the heat source below that of its surroundings while transferring the extracted heat, and any required energy input, to a heat sink, atmospheric air, or surface water.

Refrigeration is used in industry for cooling and freezing of products, condensing vapors, maintaining environmental conditions and for cold storage. The number of different applications is huge and they are a major consumer of electricity. In some sectors, particularly food, drink and chemicals it represents a significant proportion of overall site energy costs (up to 90% in the case of some cold storage facilities).

A refrigeration system is a combination of components and equipment connected in a sequential order to produce the refrigeration effect.

A refrigeration process indicates the change of thermodynamic properties of the refrigerant and the energy transfer between the refrigerant and the surroundings. The following refrigeration processes occur during the operation of a vapor compression refrigerating system:

Evaporation: In this process, the refrigerant evaporates at a lower temperature than that of its surroundings, absorbing its latent heat of vaporization.

Superheating: Saturated refrigerant vapor is usually superheated to ensure that liquid refrigerant does not flow into the compressor.

Compression: Refrigerant is compressed to a higher pressure and temperature for condensation.

Condensation: Gaseous refrigerant is condensed to liquid form by being desuperheated, then condensed, and finally subcooled, transferring its latent heat of condensation to a coolant.

Throttling and expansion: The higher-pressure

Major components of a simple mechanical vapor compression refrigeration system are shown in Figure 2.7. As the refrigerant flows through these components its phase changes from liquid to gas and then back to liquid. The flow of refrigerant can be examined by tracing the path of the refrigerant in Figure 2.7.

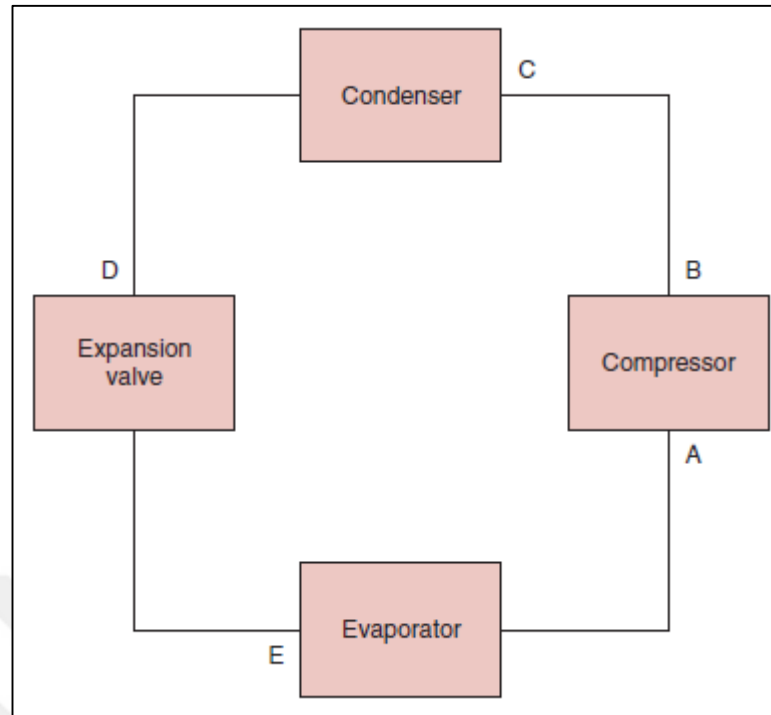


Figure 2.7 A mechanical vapour compression refrigeration system (Singh & Heldman, 1984)

At location D on Figure 2.7, just prior to the entrance to the expansion valve, the refrigerant is in a saturated liquid state. It is at or below its condensation temperature. The expansion valve separates the high pressure region from the low-pressure region. After passing through the expansion valve, the refrigerant experiences a drop in pressure accompanied by a drop in temperature. Due to the drop in pressure, some of the liquid refrigerant changes to gas. The liquid/gas mixture leaving the expansion valve is termed “flash gas”. The liquid/gas mixture enters the evaporator coils at location E. In the evaporator, the refrigerant completely vaporizes to gas by accepting heat from the media surrounding the evaporator coils. The saturated vapors may reach a superheated stage due to gain of additional heat from the surroundings.

The saturated or superheated vapors enter the compressor at location A, where the refrigerant is compressed to a high pressure. This high pressure must be below the critical pressure of the refrigerant and high enough to allow condensation of the refrigerant at a temperature slightly higher than that of commonly available heat sinks, such as ambient air or well water. Inside the compressor, the compression process of the vapors occurs at constant entropy (called an isentropic process). As the pressure of

the refrigerant increases, the temperature increases, and the refrigerant becomes superheated as shown by location B.

The superheated vapors are then conveyed to a condenser. Using either an air-cooled or a water-cooled condenser, the refrigerant discharges heat to the surrounding media. The refrigerant condenses back to the liquid state in the condenser as shown by location D. After the entire amount of refrigerant has been converted to saturated liquid, the temperature of the refrigerant may decrease below that of its condensation temperature due to additional heat discharged to the surrounding media; in other words, it may be subcooled. The subcooled or saturated liquid then enters the expansion valve and the cycle continues.

2.5.2 Refrigeration Cycles and Theories

The transfer of heat from a low-temperature region to a high-temperature one requires special devices called refrigerators.

Refrigerators are cyclic devices, and the working fluids used in the refrigeration cycles are called refrigerants. A refrigerator is shown schematically in Figure 2.8. Here Q_L is the magnitude of the heat removed from the refrigerated space at temperature T_L . Q_H is the magnitude of the heat rejected to the warm space at temperature T_H , and $W_{\text{net,in}}$ is the net work input to the refrigerator. Q_L and Q_H represent magnitudes and thus are positive quantities.

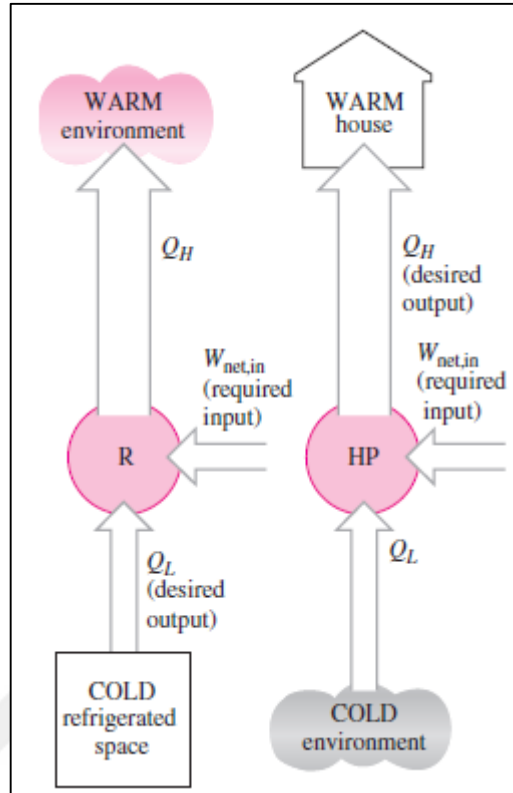


Figure 2.8 The heat transfer schema of (a) refrigerator, (b) heat pump (Cengel & Boles, 2006)

The performance of refrigerators and heat pumps is expressed in terms of the coefficient of performance (COP), defined as

$$COP_R = \frac{\text{Desired output}}{\text{Required Output}} = \frac{\text{Cooling effect}}{\text{Work input}} = \frac{Q_L}{W_{\text{net},in}} \quad (2.1)$$

2.5.2.1 The Reversed Carnot Cycle

The Carnot cycle is a totally reversible cycle that consists of two reversible isothermal and two isentropic processes. It has the maximum thermal efficiency for given temperature limits, and it serves as a standard against which actual power cycles can be compared.

Since it is a reversible cycle, all four processes that comprise the Carnot cycle can be reversed. Reversing the cycle does also reverse the directions of any heat and work interactions. The result is a cycle that operates in the counterclockwise direction on a

T-s diagram, which is called the reversed Carnot cycle. A refrigerator or heat pump that operates on the reversed Carnot cycle is called a Carnot refrigerator or a Carnot heat pump.

Consider a reversed Carnot cycle executed within the saturation dome of a refrigerant, as shown in Figure 2.9. The refrigerant absorbs heat isothermally from a low-temperature source at T_L in the amount of Q_L (process 1-2), is compressed isentropically to state 3 (temperature rises to T_H), rejects heat isothermally to a high-temperature sink at T_H in the amount of Q_H (process 3-4), and expands isentropically to state 1 (temperature drops to T_L). The refrigerant changes from a saturated vapor state to a saturated liquid state in the condenser during process 3-4.

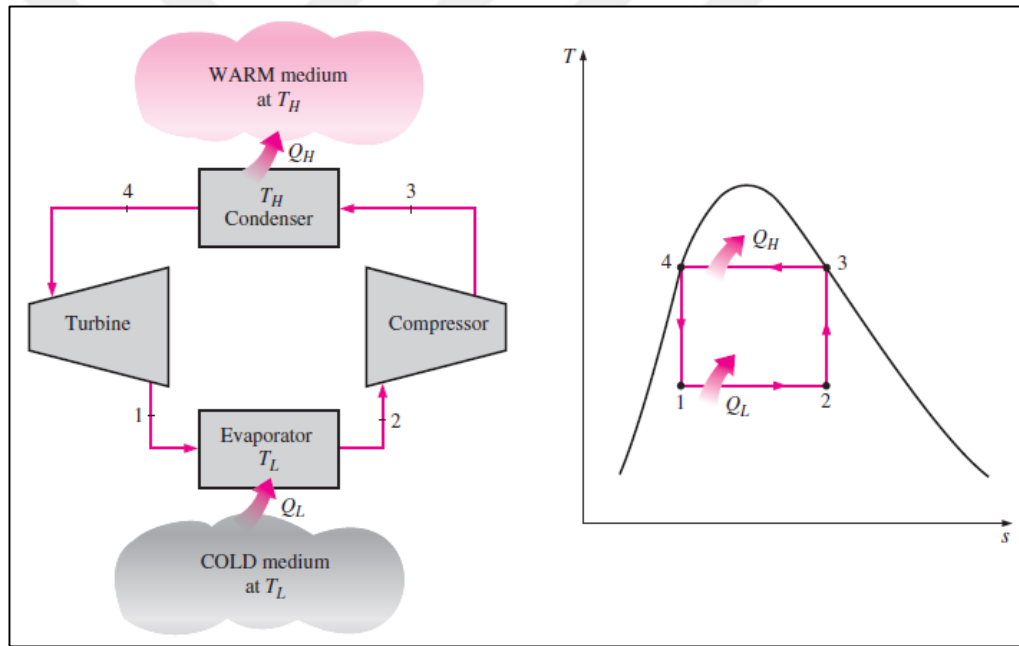


Figure 2.9 Schematic of a Carnot refrigerator and T-s diagram of the reversed Carnot cycle (Cengel & Boles, 2006)

The coefficients of performance of Carnot refrigerators is expressed in terms of temperatures as

$$COP_{R,Carnot} = \frac{1}{T_H/T_L - 1} \quad (2.2)$$

Notice that both COPs increase as the difference between the two temperatures decreases, that is, as T_L rises or T_H falls.

The reversed Carnot cycle is the most efficient refrigeration cycle operating between two specified temperature levels. Therefore, it is natural to look at it first as a prospective ideal cycle for refrigerators and heat pumps. If we could, we certainly would adapt it as the ideal cycle. As explained below, however, the reversed Carnot cycle is not a suitable model for refrigeration cycles.

The two isothermal heat transfer processes are not difficult to achieve in practice since maintaining a constant pressure automatically fixes the temperature of a two-phase mixture at the saturation value. Therefore, processes 1-2 and 3-4 can be approached closely in actual evaporators and condensers. However, processes 2-3 and 4-1 cannot be approximated closely in practice. This is because process 2-3 involves the compression of a liquid-vapor mixture, which requires a compressor that will handle two phases, and process 4-1 involves the expansion of high-moisture-content refrigerant in a turbine.

It seems as if these problems could be eliminated by executing the reversed Carnot cycle outside the saturation region. But in this case we have difficulty in maintaining isothermal conditions during the heat-absorption and heat-rejection processes. Therefore, we conclude that the reversed Carnot cycle cannot be approximated in actual devices and is not a realistic model for refrigeration cycles. However, the reversed Carnot cycle can serve as a standard against which actual refrigeration cycles are compared.

2.5.2.2 The Ideal Vapor-Compression Refrigeration Cycle

Many of the impracticalities associated with the reversed Carnot cycle can be eliminated by vaporizing the refrigerant completely before it is compressed and by replacing the turbine with a throttling device, such as an expansion valve or capillary tube. The cycle that results is called the ideal vapor-compression refrigeration cycle, and it is shown schematically and on a T-s diagram in Figure 2.10. The vapor-compression refrigeration cycle is the most widely used cycle for refrigerators, air-conditioning systems, and heat pumps. It consists of four processes:

1-2 Isentropic compression in a compressor

2-3 Constant-pressure heat rejection in a condenser

3-4 Throttling in an expansion device

4-1 Constant-pressure heat absorption in an evaporator

In an ideal vapor-compression refrigeration cycle, the refrigerant enters the compressor at state 1 as saturated vapor and is compressed isentropically to the condenser pressure. The temperature of the refrigerant increases during this isentropic compression process to well above the temperature of the surrounding medium. The refrigerant then enters the condenser as superheated vapor at state 2 and leaves as saturated liquid at state 3 as a result of heat rejection to the surroundings. The temperature of the refrigerant at this state is still above the temperature of the surroundings.

The saturated liquid refrigerant at state 3 is throttled to the evaporator pressure by passing it through an expansion valve or capillary tube. The temperature of the refrigerant drops below the temperature of the refrigerated space during this process. The refrigerant enters the evaporator at state 4 as a low-quality saturated mixture, and it completely evaporates by absorbing heat from the refrigerated space. The refrigerant leaves the evaporator as saturated vapor and reenters the compressor, completing the cycle.

1) Process “4-1” – The process ‘4 – 1’ is an internally, reversible, constant pressure heat interaction in which the working fluid (as a mixture of vapor and liquid) is evaporated to a saturated vapor at state point 1. The heat necessary for evaporation is supplied by the refrigerated space surrounding the evaporator (i.e. $Q_C = h_1 - h_4$). During this transfer of heat energy, only latent heat is absorbed resulting in the refrigerant remaining at a constant pressure and constant temperature.

2) Process “1-2” - The process ‘1 – 2’ is a reversible, adiabatic (isentropic) compression of the refrigerant from a low pressure gas to a high pressure gas. The saturated vapor at state 1 is superheated to state 2. During the compression process, the refrigerant gas absorbs additional heat known as the “Heat of Compression”. The additional heat energy caused by compression is represented by the line between points 1 and 2 and is given by equation $Q_W = h_2 - h_1$. Note that point 2 is to the right of point 1, indicating the additional enthalpy resulting from the Heat of Compression.

3) Process “2-3” - The process ‘2 – 3’ is an internally, reversible, constant pressure heat rejection in which the working substance is desuperheated and then condensed to a saturated liquid at 3. The condenser removes the heat of compression plus the latent heat of evaporation, collectively known as the “Total Heat of Rejection”, or THR. The total heat rejection is given by equation $Q_H = h_2 - h_3$ and is sum of $Q_C + Q_W$.

4) Process “3-4” - The process ‘3 – 4’ through expansion valve is an irreversible process in which the refrigerant vapor is expanded from high pressure to low pressure region. Both temperature and pressure decrease at constant enthalpy (i.e. $h_3 = h_4$).

The process of evaporation at a low pressure and corresponding low temperature, followed by compression, followed by condensation at around atmospheric temperature and corresponding high pressure, is the refrigeration cycle. Because the constant-temperature lines and constant-pressure lines in the middle two-phase (wet vapor) region are horizontal, they are closely related. The specific pressure of a refrigerant in the two-phase region determines its temperature, and vice versa. Overall, the energy side of the refrigeration cycle can therefore be summed up:

1) Heat taken in from surroundings at the (low) evaporator temperature and pressure ($Q_C = h_1 - h_4$),

2) Heat equivalent to the work done by the compressor ($Q_W = h_2 - h_1$) and

3) Heat rejected at the (high) compressor pressure and temperature ($Q_H = h_2 - h_3$).

Note that the amount of heat rejected by the condenser and absorbed by the evaporator is represented as a horizontal (constant pressure) line. The compression path is represented along a constant entropy line, and the expansion process is represented along a vertical (constant enthalpy) line.

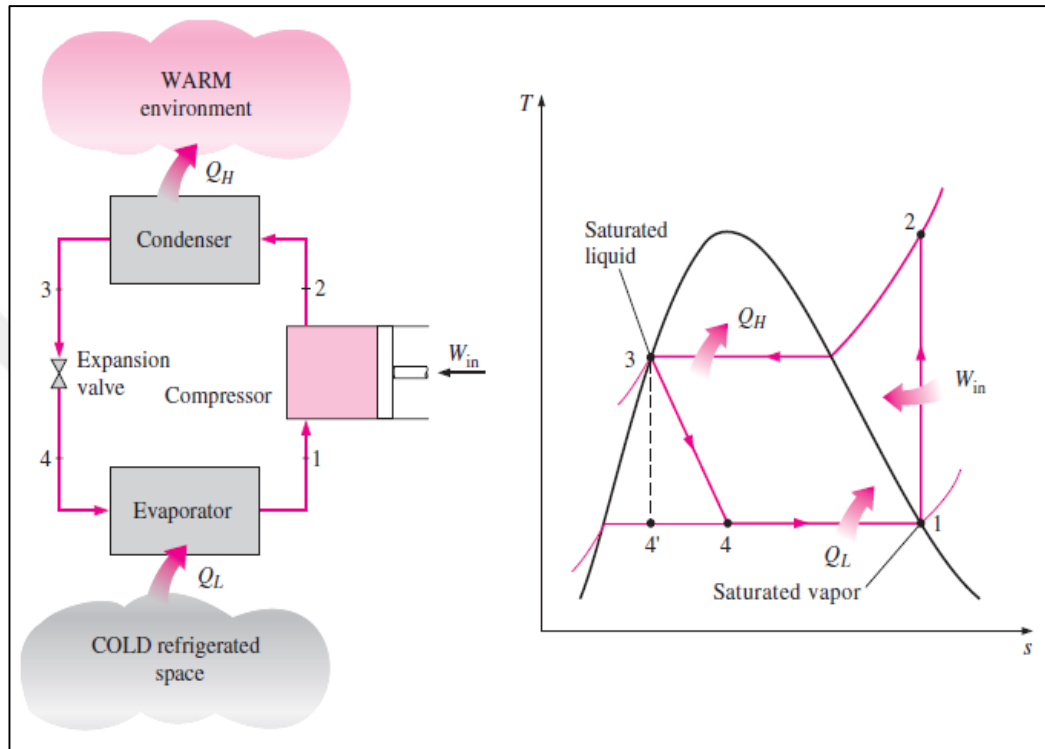


Figure 2.10 Schematic and $T-s$ diagram for the ideal vapor-compression refrigeration cycle (Cengel & Boles, 2006)

In a household refrigerator, the tubes in the freezer compartment where heat is absorbed by the refrigerant serves as the evaporator. The coils behind the refrigerator, where heat is dissipated to the kitchen air, serve as the condenser (Figure 2.11).

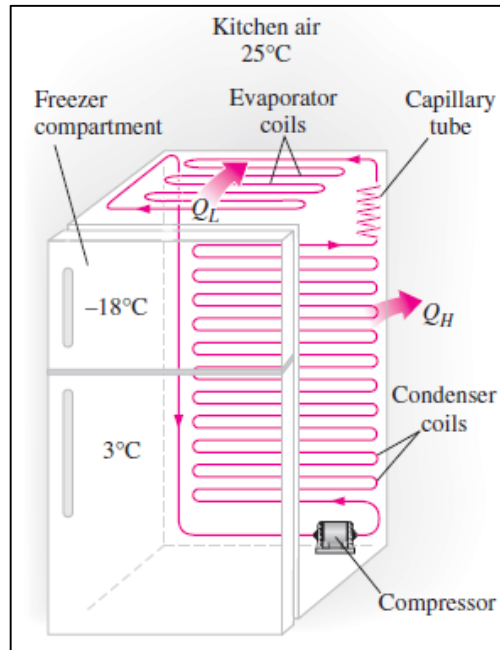


Figure 2.11 An ordinary household refrigerator (Cengel & Boles, 2006)

Remember that the area under the process curve on a T-s diagram represents the heat transfer for internally reversible processes. The area under the process curve 4-1 represents the heat absorbed by the refrigerant in the evaporator, and the area under the process curve 2-3 represents the heat rejected in the condenser. A rule of thumb is that the COP improves by 2 to 4 percent for each °C the evaporating temperature is raised or the condensing temperature is lowered.

Another diagram frequently used in the analysis of vapor-compression refrigeration cycles is the P-h diagram, as shown in Figure 2.12. On this diagram, three of the four processes appear as straight lines, and the heat transfer in the condenser and the evaporator is proportional to the lengths of the corresponding process curves.

Notice that unlike the ideal cycles discussed before, the ideal vapour compression refrigeration cycle is not an internally reversible cycle since it involves an irreversible (throttling) process. This process is maintained in the cycle to make it a more realistic model for the actual vapor-compression refrigeration cycle. If the throttling device were replaced by an isentropic turbine, the refrigerant would enter the evaporator at state 4₋ instead of state 4. As a result, the refrigeration capacity would increase (by the area under process curve 4₋-4 in Figure 2.12) and the net work input would decrease

(by the amount of work output of the turbine). Replacing the expansion valve by a turbine is not practical, however, since the added benefits cannot justify the added cost and complexity.

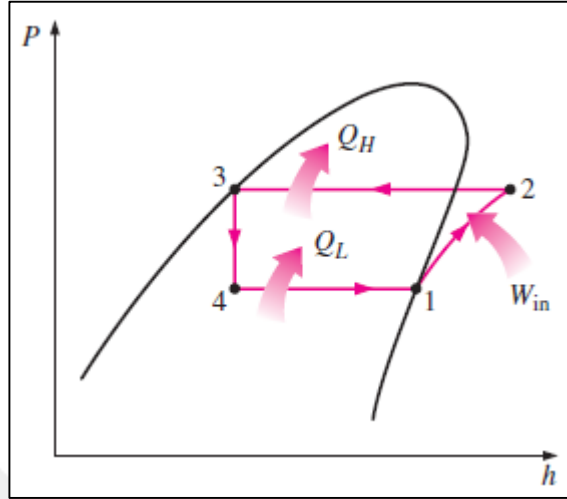


Figure 2.12 The P-h diagram of an ideal vapor-compression refrigeration cycle (Cengel & Boles, 2006)

All four components associated with the vapor-compression refrigeration cycle are steady-flow devices, and thus all four processes that make up the cycle can be analyzed as steady-flow processes. The kinetic and potential energy changes of the refrigerant are usually small relative to the work and heat transfer terms, and therefore they can be neglected. Then the steadyflow energy equation on a unit-mass basis reduces to

$$(q_{in} - q_{out}) + (w_{in} - w_{out}) = h_e - h_i \quad (2.3)$$

The condenser and the evaporator do not involve any work, and the compressor can be approximated as adiabatic. Then the COPs of refrigerators operating on the vapor-compression refrigeration cycle can be expressed as

$$COP_R = \frac{q_L}{w_{net,in}} = \frac{h_1 - h_4}{h_2 - h_1} \quad (2.3)$$

2.5.2.3 Actual Vapor–Compression Refrigeration Cycle

An actual vapor-compression refrigeration cycle differs from the ideal one in several ways, owing mostly to the irreversibilities that occur in various components. Two common sources of irreversibilities are fluid friction (causes pressure drops) and heat transfer to or from the surroundings. The T-s diagram of an actual vapor-compression refrigeration cycle is shown in Figure 2.13.

In the ideal cycle, the refrigerant leaves the evaporator and enters the compressor as saturated vapor. In practice, however, it may not be possible to control the state of the refrigerant so precisely. Instead, it is easier to design the system so that the refrigerant is slightly superheated at the compressor inlet. This slight overdesign ensures that the refrigerant is completely vaporized when it enters the compressor. Also, the line connecting the evaporator to the compressor is usually very long; thus the pressure drop caused by fluid friction and heat transfer from the surroundings to the refrigerant can be very significant. The result of superheating, heat gain in the connecting line, and pressure drops in the evaporator and the connecting line is an increase in the specific volume, thus an increase in the power input requirements to the compressor since steady-flow work is proportional to the specific volume.

The compression process in the ideal cycle is internally reversible and adiabatic, and thus isentropic. The actual compression process, however, involves frictional effects, which increase the entropy, and heat transfer, which may increase or decrease the entropy, depending on the direction. Therefore, the entropy of the refrigerant may increase (process 1-2) or decrease (process 1-2₋) during an actual compression process, depending on which effects dominate. The compression process 1-2₋ may be even more desirable than the isentropic compression process since the specific volume of the refrigerant and thus the work input requirement are smaller in this case. Therefore, the refrigerant should be cooled during the compression process whenever it is practical and economical to do so.

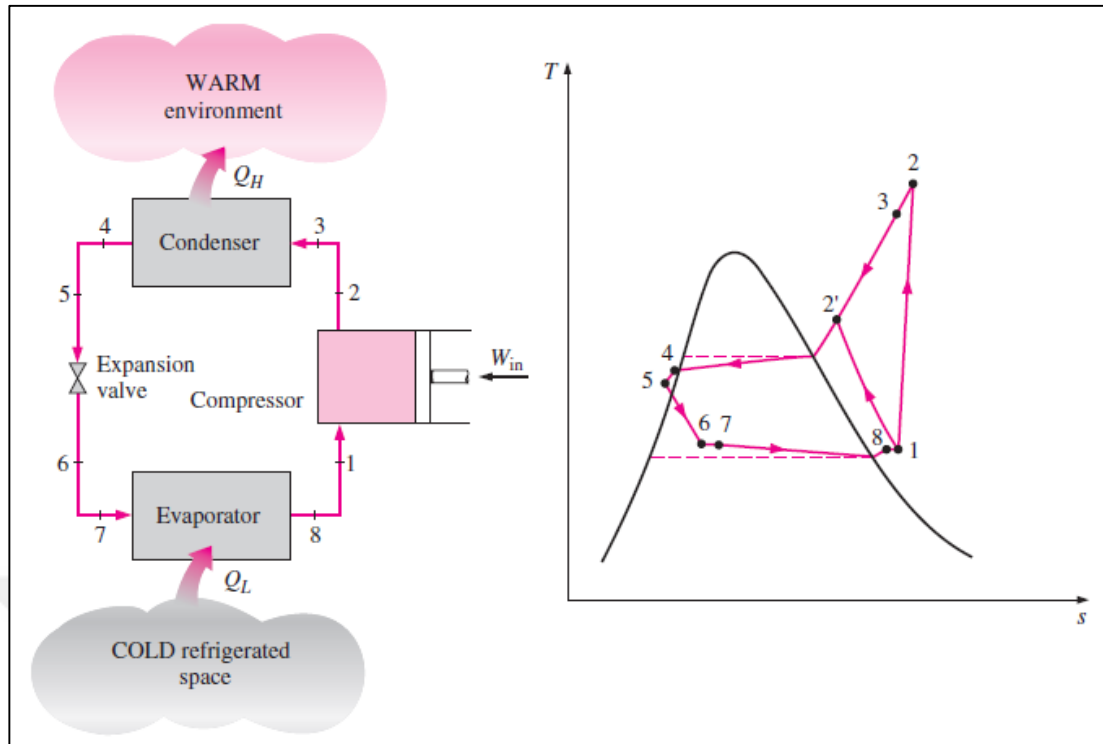


Figure 2.13 Schematic and T-s diagram for the actual vapor-compression refrigeration cycle (Cengel & Boles, 2006)

In the ideal case, the refrigerant is assumed to leave the condenser as saturated liquid at the compressor exit pressure. In reality, however, it is unavoidable to have some pressure drop in the condenser as well as in the lines connecting the condenser to the compressor and to the throttling valve. Also, it is not easy to execute the condensation process with such precision that the refrigerant is a saturated liquid at the end, and it is undesirable to route the refrigerant to the throttling valve before the refrigerant is completely condensed. Therefore, the refrigerant is subcooled somewhat before it enters the throttling valve. We do not mind this at all, however, since the refrigerant in this case enters the evaporator with a lower enthalpy and thus can absorb more heat from the refrigerated space. The throttling valve and the evaporator are usually located very close to each other, so the pressure drop in the connecting line is small.

CHAPTER THREE

NUMERICAL STUDY

3.1 Model of the Refrigerator

In this study the model has 1.7m x 0.48m x 0.45m (height x width x depth) internal dimensions, a single-door with only a fresh food cabinet (Figure 3.1.b). The cycle defrost type evaporator which has 44.5x91 cm² is located into the back wall, opposite of the door. The wire-on-tube condenser is located on the external surface of the back wall. Refrigerator has four main glass shelves, five door shelves, two vegetable compartments and their glass shelves, a light box and a fan box (Figure 3.1.b).

These inner configurations which were assumed as a geometrical obstacle against the flow were defined a cavity inside the generated air volume. Some neglectable details of inner configuration, solid fan geometry and external walls of refrigerator (inner plastic, insulation material and sheet metal) weren't modeled for to simplify the geometry and reduce the numerical analysis time.

The refrigerator model which was prepared due to the assumptions is shown in Figure 3.1.

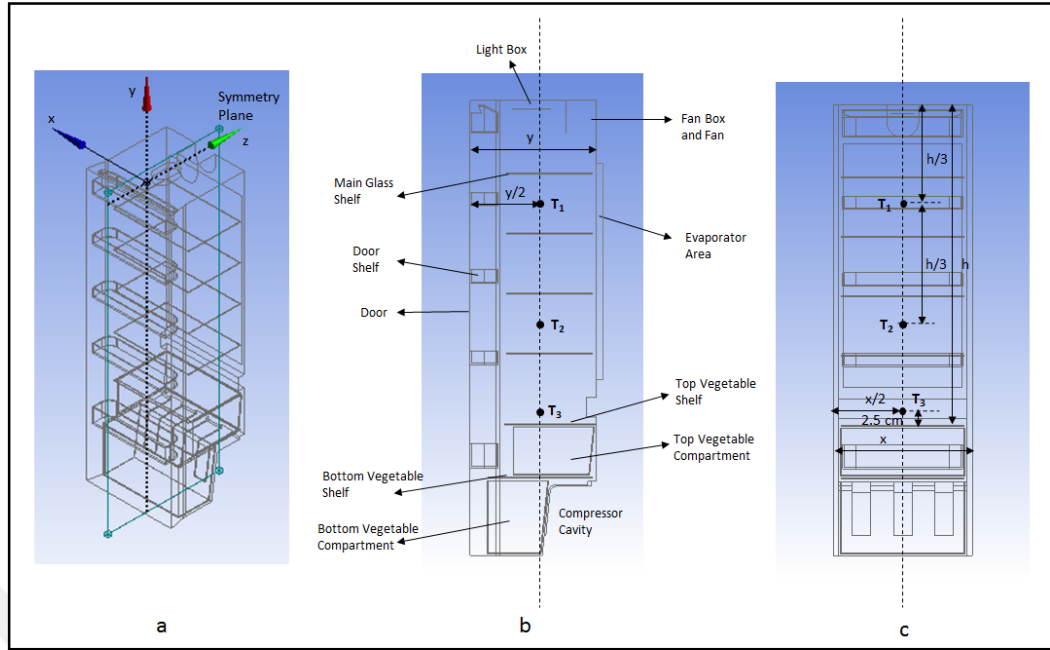


Figure 3.1 Internal air volume of refrigerator model views: (a) Isometric view, (b) Left view, (c) Front view

3.2 Meshing

In order to analyze fluid flows, flow domains are split into smaller subdomains (made up of geometric primitives like hexahedra and tetrahedra in 3D and quadrilaterals and triangles in 2D). The governing equations are then discretized and solved inside each of these subdomains. Typically, one of three methods is used to solve the approximate version of the system of equations: finite volumes, finite elements, or finite differences.

The generated mesh has 850084 tetrahedral elements and 118839 nodes. Then, the names of internal air volume model's regions were defined for the ease of definitions of boundary condition.

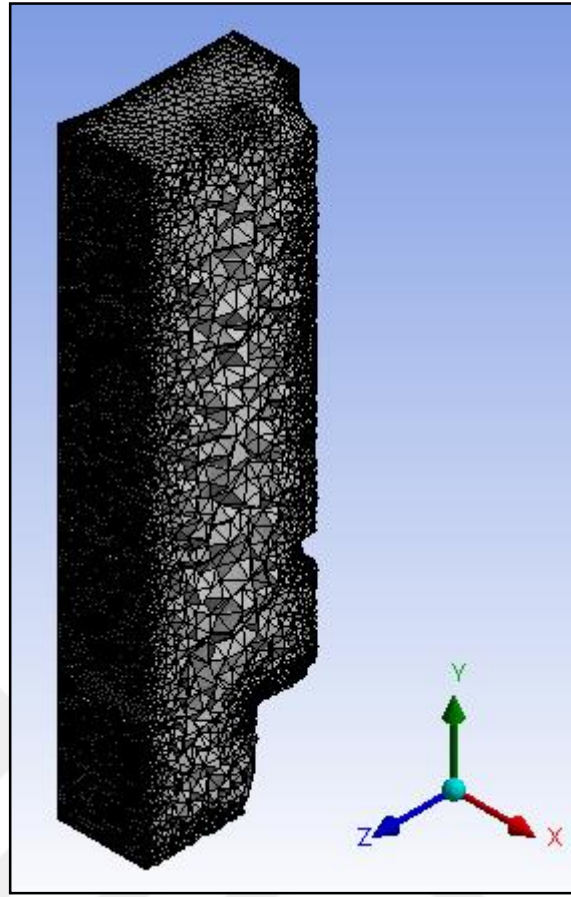


Figure 3.2 Mesh structure on the section of symmetry plane

3.3 Boundary Conditions

CFX interface was used to determine the boundary conditions.

The domestic refrigerator analysis is a transient problem because of the effect of compressor on-off cycle that occurs the oscillating of temperatures of the internal air volume. Studies on refrigerator analysis in literature were considered average temperatures during equilibrium condition of refrigerator. Hence, steady state analyses were carried out to simplify the analysis.

Internal walls of the inside air are composed of internal plastic, insulation material and sheet metal from inside to outside. Heat transfer occurs along these layers into the refrigerator. Overall heat transfer coefficient based on the thermal resistances was calculated by Equation 3.1 (Incropera, DeWitt, Bergman & Lavine, 2006, p.100). In

this equation, thermal resistance of internal plastic and sheet metal was neglected because they have smaller thickness compared to the insulation material .

$$U = \frac{1}{l_y/k_y} \quad (3.1)$$

In Equation 3.1, l_y (m) is thickness of insulation material and k_y (W/mK) is the thermal conductivity of insulation material.

The temperature of evaporator was determined fixed which calculated the average value of measurement from the thermocouples on the evaporator surfaces at the experiment according to the article.

Boundary conditions that was used for the model are shown in Table 3.1.

Table 3.1 Boundary conditions

Region	Boundary Condition
Evaporator Wall	Constant Temperature -3.16 °C
Front (Door) Wall	T=24.82 °C
	U=0.619 W/m ² K
Compressor Cavity Top Wall	T=26 °C
	U=0.416 W/m ² K
Compressor Cavity Side Wall	T=26 °C
	U=0.327 W/m ² K
Back Wall	T=25.47 °C
	U=0.331 W/m ² K
Side Walls	T=24.9 °C
	U=0.466 W/m ² K
Top Wall	T=25 °C
	U=0.353 W/m ² K
Bottom Wall and Flow Obstaclers	Adiabatic

Gravity is defined as 9,81 m/s² along y direction due to the buoyancy effects.

The internal air's thermo-physical properties according such as density (ρ), specific heat (c_p), thermal conductivity (k), thermal diffusivity (α), kinematic viscosity (ν) that were calculated according to its film temperature 3,75 °C are shown in the Table 3.2. L was defined as evaporator's height to calculate Rayleigh number that is needed for defining flow regime. Rayleigh number was calculated according to Equation 3.2 with these values and found as $7,3 \times 10^8$. This value is smaller than critical Rayleigh number (10^9). So laminar flow was accepted for flow regime.

$$Ra_L = \frac{\rho \beta (T_\infty - T_s) L^3}{\nu \alpha} \quad (3.2)$$

Table 3.2 The internal air's thermo-physical properties

T_f (K)	ρ (kg/m³)	c_p (J/kgK)	k (W/mK)	α (m²/s)	ν (m²/s)
276.9	1.269	1006.538	0.024	1.94508 x10 ⁻⁵	1.38341x10 ⁻⁵

Radiation effects were taken into between the internal surfaces due to the literature studies (Laguerre, Amara, Moureh, & Flick, 2007).

According to the article, fan effects were neglected to simplify the analysis.

Definition of boundary conditions were completed under steady state circumstances. And initial conditions were defined.

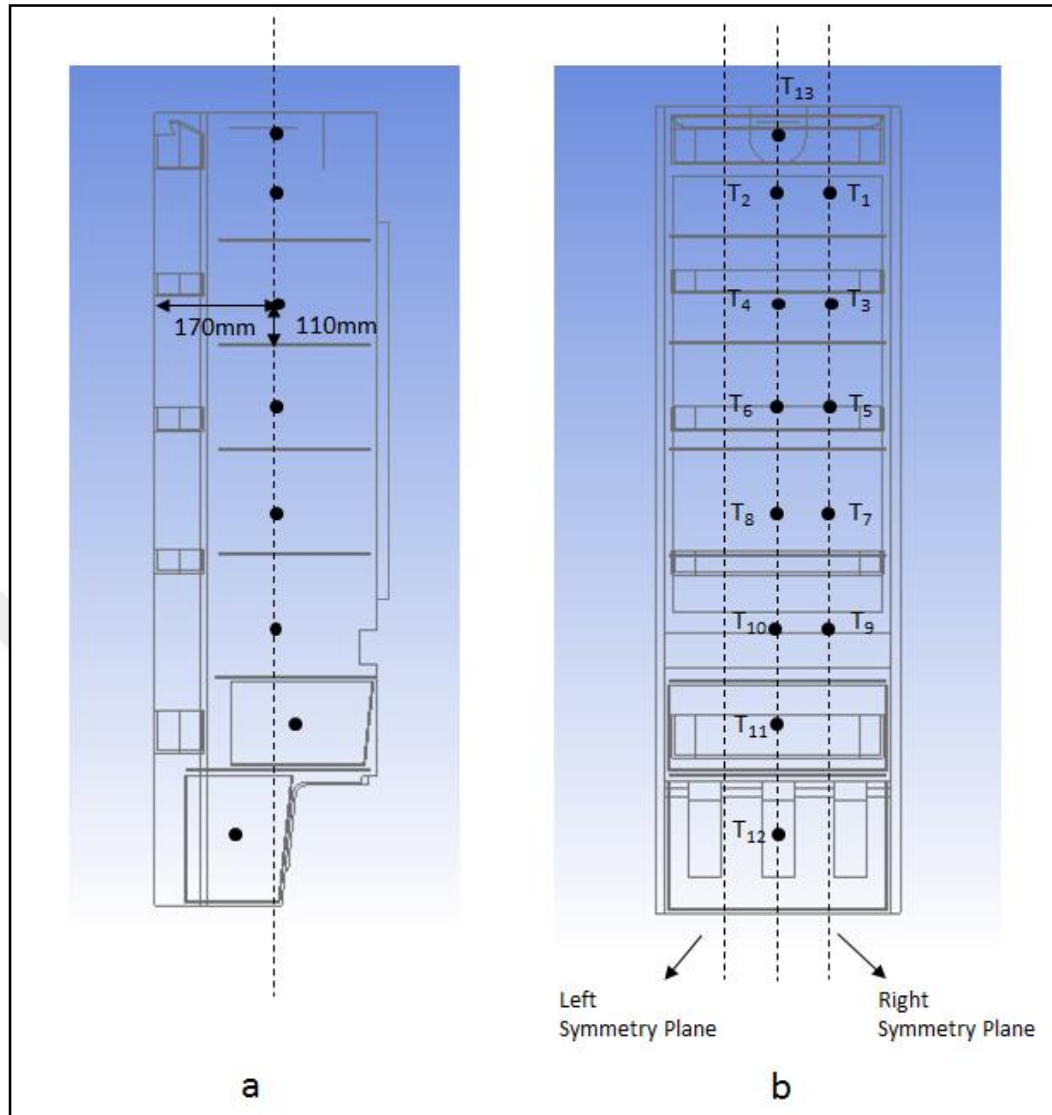


Figure 3.3 Arrangement of thermocouples: (a) On surface, (b) In internal air

3.4 Solver

Analyses' iterations have continued until residuals reach $1.0\text{e-}03$. It tooks 4 hours with 1213 iterations. Heat transfer, momentum and mass and H-energy imbalances were shown in Figure 3.4 and Figure 3.5.

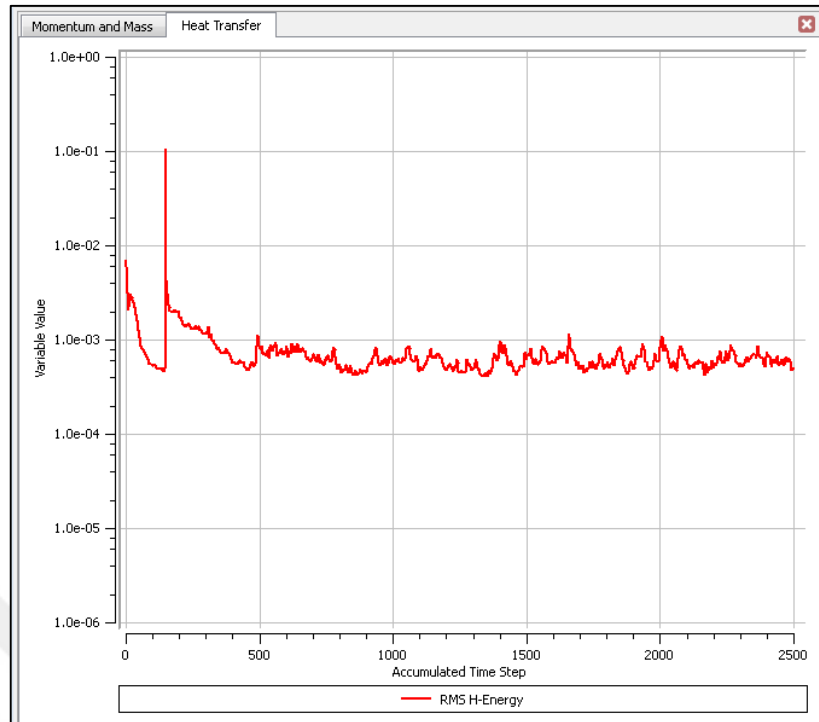


Figure 3.4 Heat transfer residuals

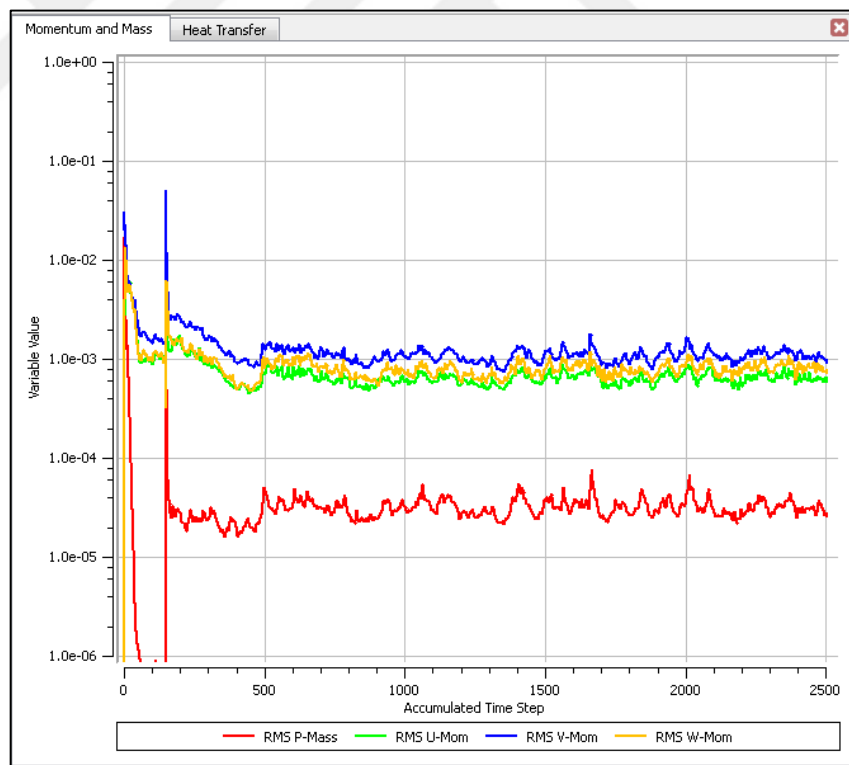


Figure 3.5 Momentum and mass residuals

3.5 Results and Validation of the Model

After solution was converged, temperature contours and velocity vectors were displayed for visual consideration of the results. The temperature distribution and flow field of the internal volume are shown in Figure 3.6.

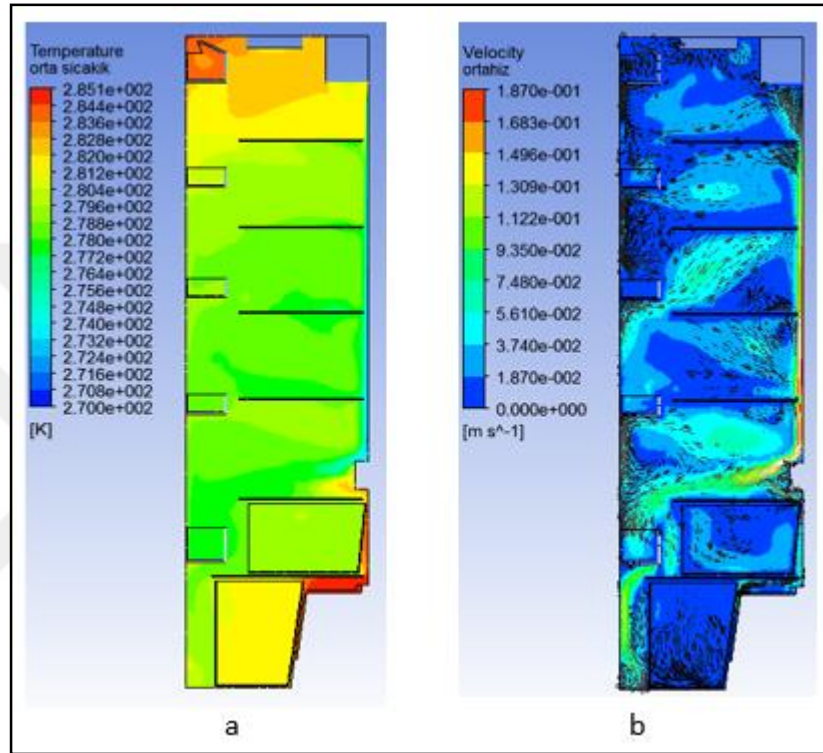


Figure 3.6 Flow field on the symmetry plane: (a) temperature distribution, (b) velocity distribution

The calculated temperatures of measurement points and experimental results are shown in Figure 3.7. As shown in Figure 3.7, it is found that the computational and experimental results qualitatively agree with each other. All computational results shown with the black dots are within this error band and so closed to desired line. The maximum absolute temperature difference is 1.98 °K and this difference occurs on the lightbox. Second biggest absolute temperature difference is 1 °K and it is within the acceptable limits according to the literature views, so that the numerical results have been concluded to be reliable. The difference between experimental and computational results may be attributed to the lack of precise data on the airflow rates and the unaccounted heat transfer rates through the door gaskets and the compressor.

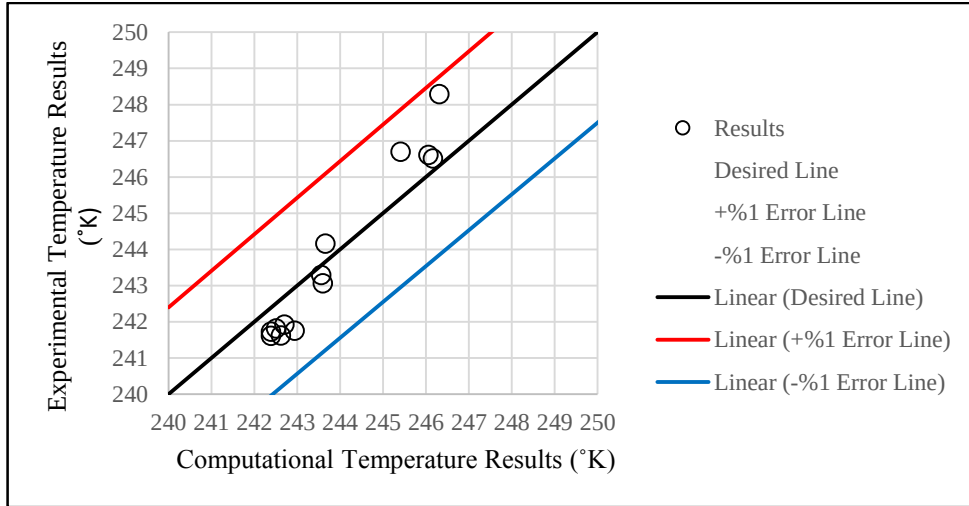


Figure 3.7 Temperature comparison between experimental and computational results

3.6 Effects of the Radiation

Radiation was taken into account for validation of the model. Radiation wasn't taken into account and all process steps were repeated to investigate effects of radiation. More warmer temperature distribution fields were acquired when radiation effects weren't taken into account as shown in Figure 3.8.

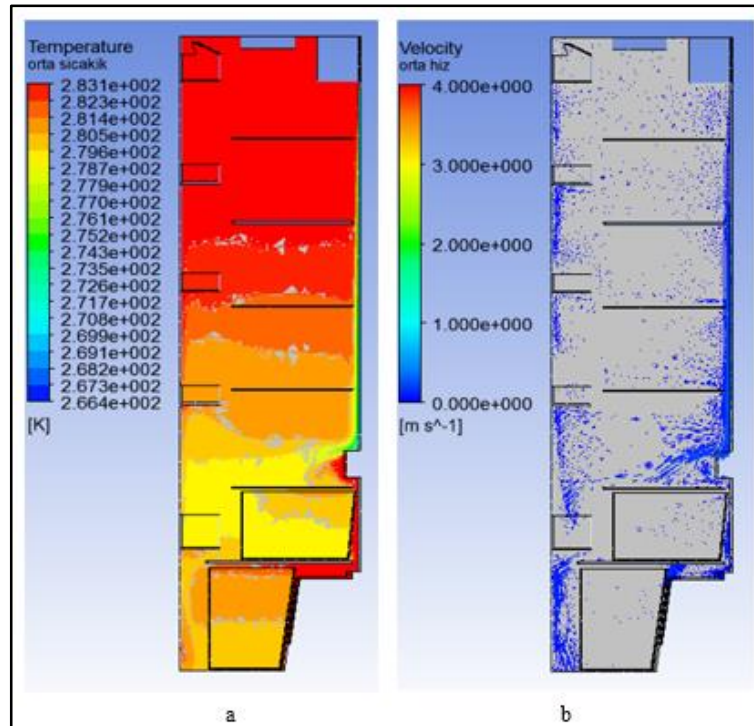


Figure 3.8 (a) Temperature distribution and (b) velocity distribution without radiation

Extremely hot temperatures were acquired when radiation effects weren't taken into account as shown in Figure 3.9.

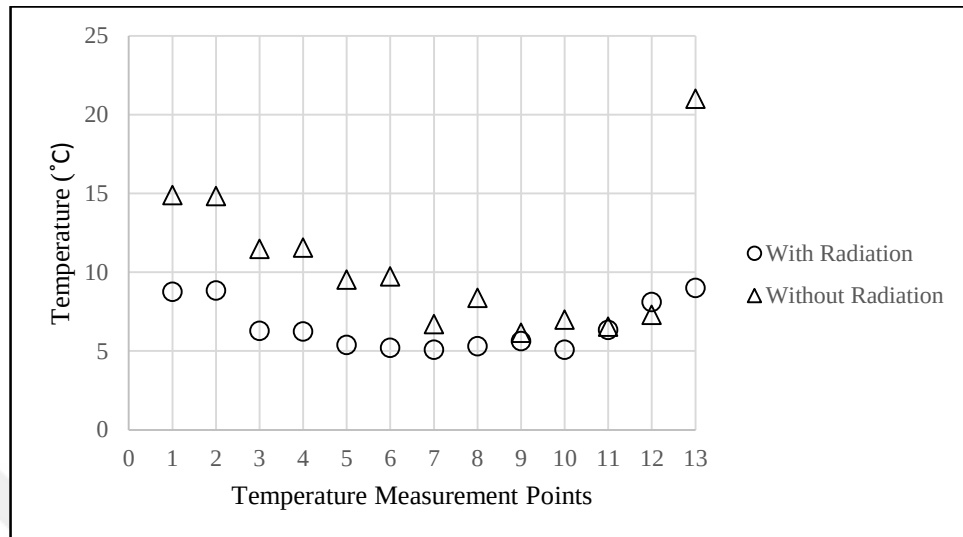


Figure 3.9 Variation of temperature due to radiation

CHAPTER FOUR

PARAMETRIC STUDY

4.1 Model Including Fan

Fan was added to the geometry and all process steps were repeated. Measured velocity values of fan are shown in Table 4.1. Average velocity is 3,72 m/s and this value was used as the program input.

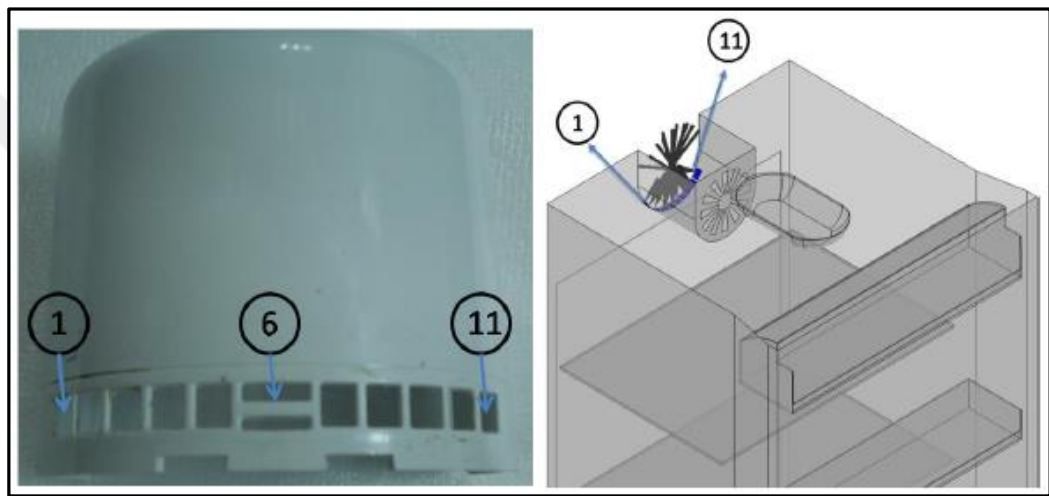


Figure 4.1 Fan geometry

Table 4.1 Velocity of fan

Exit Region of Fan	Velocity (m/s)
1	4,35
2	4,60
3	4,68
4	4,35
5	4,00
6	3,00
7	3,47
8	3,58
9	3,20
10	2,84
11	2,80

More homogeneous airflow was acquired when fan effects have taken into account as shown in Figure 4.2.

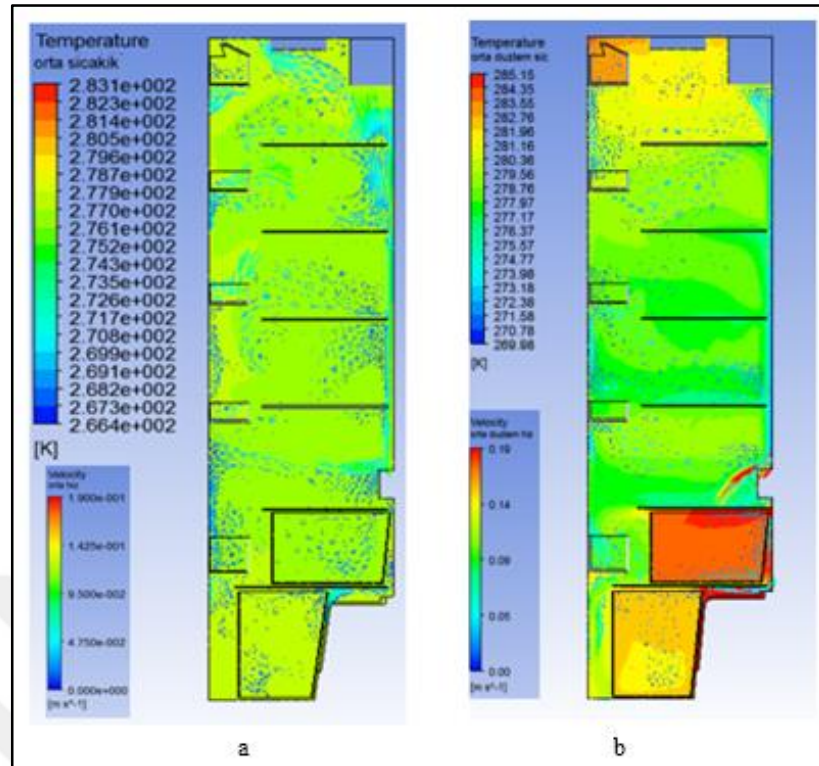


Figure 4.2 Temperature and flow field on the symmetry plane-A a) with fan b) without fan

The comparison of the temperature results of refrigerator models with and without fan are shown in Figure 4.3. It is seen that colder temperatures were acquired when fan effects were taken into account as shown in Figure 4.3.

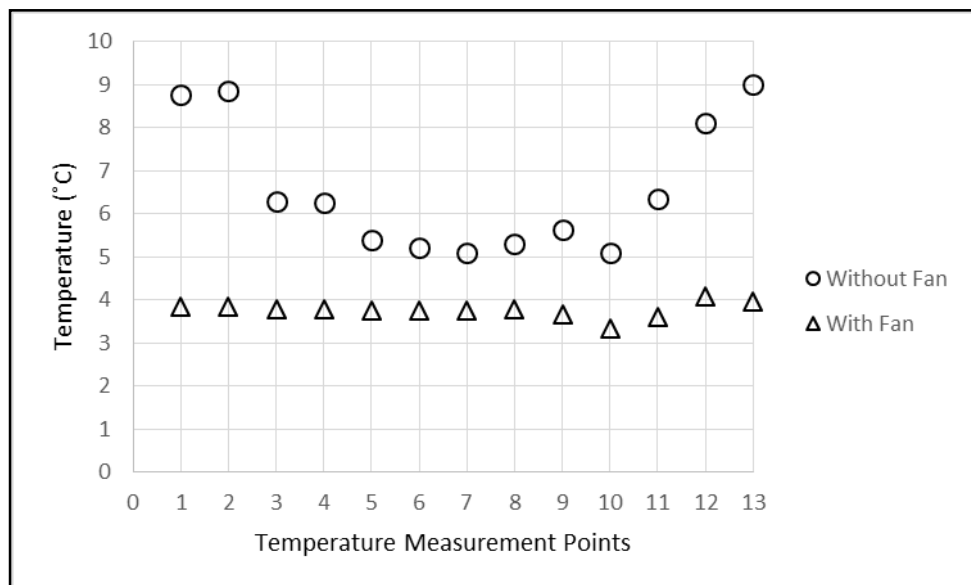


Figure 4.3 Temperature comparison between with and without fan results

4.2 Determination of Design Parameters

Design parameters were decided after validation of numerical model. The suitability of refrigerator design was emphasized to investigate by considering the distributions of temperature and flow field. Therefore, the design parameters were determined according to acquired the distributions of temperature and flow field from the numerical results.

As shown in Figure 4.2, cold zones are near the back wall because of the evaporator. If effectiveness of evaporator increase, colder temperatures will be acquired. So the effects which increase the back wall surface area were investigated. Furthermore, when colder temperatures were obtained at the back wall, necessity of extra precautions occurred for providing temperature uniformity. It was foreseen that a fan could help to provide this uniformity by increasing the air circulation inside of the refrigerator, so fan was added to the geometry to see this effect. For optimizing these effects; blowing angle of fan, velocity of fan and position of fan which shifted on x and y directions were investigated.

The investigated design input parameters and their ranges are shown in Table 3. Output parameters were defined with 3 temperature measurement points and cooling capacity of evaporator according to ISO 15502 Standards. Temperature of these points have to be between 0 and 10 °C and average temperature of these points have not to exceed 5°C according to this standart.

The investigated design parameters are shown in Table 4.3.

Table 4.3 The range of design parameters values variation

Design Parameters	Current Value	Range
Blowing Angle of Fan	1°	1° – 165°
Average Velocity of Fan	3,72 m/s	1 m/s – 6 m/s
y- Position of Fan	0	0mm – 1070mm
x- Position of Fan	180mm	0mm – 360mm
Temperature of Evaporator	-3,16 °C	-8 – 0 °C

CHAPTER FIVE

RESULTS AND DISCUSSIONS

5.1 Effects of Fan

Blowing angle of fan, average velocity of fan and position of fan were investigated.

5.1.1 Effects of Blowing Angle of Fan

Normally fan blows through the y position. Blowing angle was investigated to understand it is the best position or not. Blowing angle of the fan is input parameter and its range is $1^\circ - 165^\circ$. Output parameters are T_1 , T_2 , T_3 and cooling capacity of evaporator.

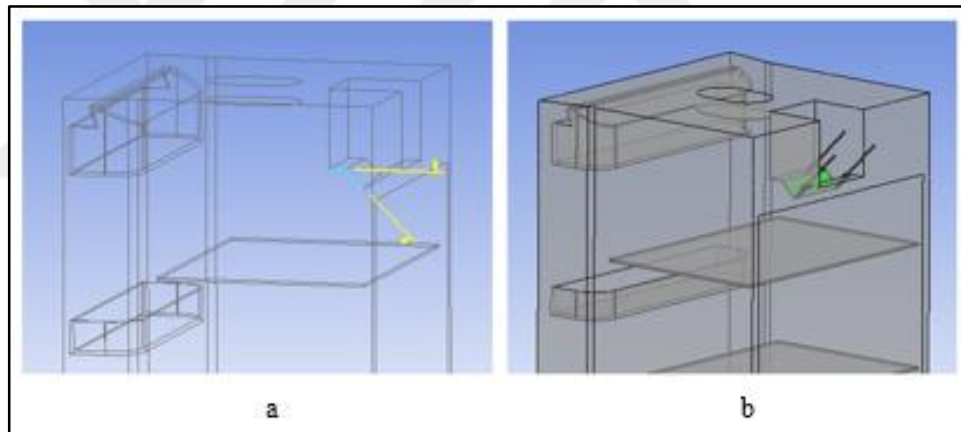


Figure 5.1 Blowing angle of fan (a) geometry, (b) set-up

The temperature and evaporator capacity results are shown in the Figure 5.2. This figure indicates that when the blowing angle is 150° , the evaporator capacity is maximum. But $T_3 > T_2$ in this situation and it's not a proper output result according to ISO 15502 Standards. The second biggest evaporator capacity was acquired when the blowing angle is 120° and $T_1 > T_2 > T_3$. It means that the optimum blowing angle is 120° and it is shown in Figure 5.3.

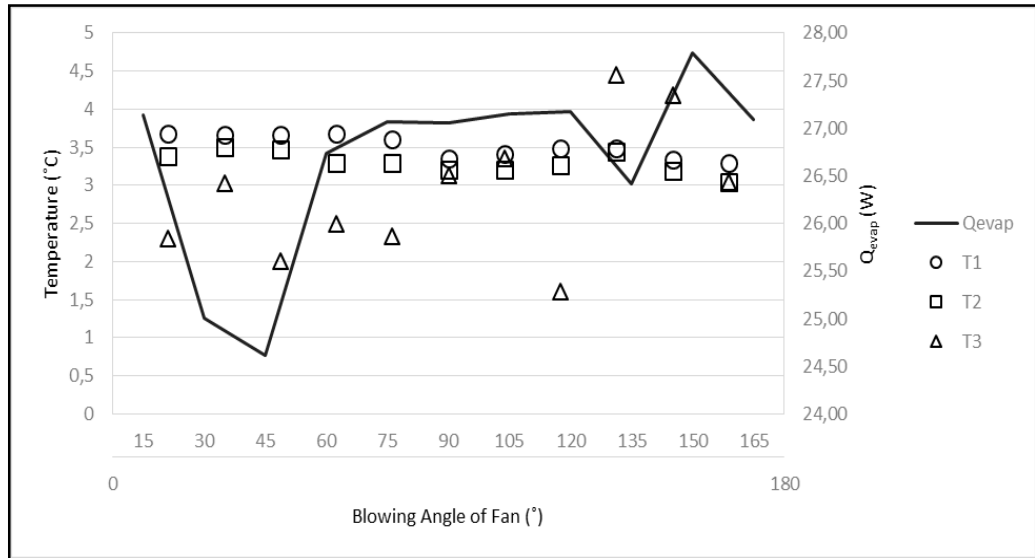


Figure 5.2 Variation of temperature due to blowing angle of fan

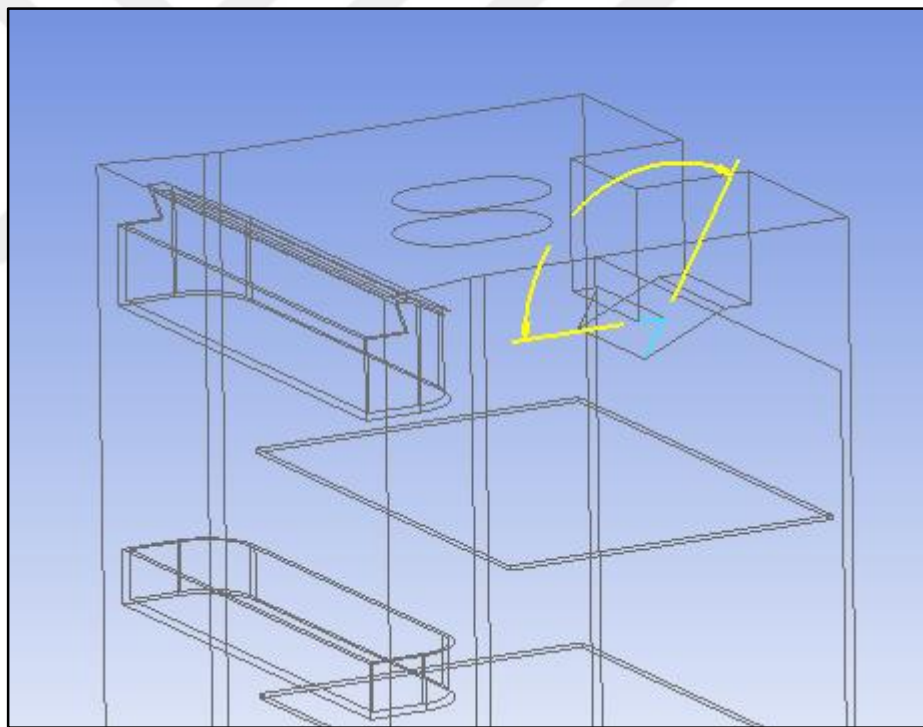


Figure 5.3 Optimum blowing angle of fan

5.1.2 Effects of the Blowing Angle of Fan and Velocity of Fan

In addition to fan's blowing angle investigation, both velocity and blowing angle of fan were investigated together. Blowing angle of the fan is input parameter and its

range is $0^\circ - 165^\circ$. Average velocity of fan is also input parameter and its range is 1 m/s – 6 m/s. As mentioned current average velocity of fan is 3.72 m/s. Output parameters are T_1 , T_2 , T_3 and cooling capacity of evaporator.

The temperature and evaporator capacity results are shown in the Figure 5.4. Figure 5.4a, Figure 5.4b, Figure 5.4c, Figure 5.4d, Figure 5.4e, Figure 5.4f show the acquired results when average velocity of fan is 1 m/s, 2 m/s, 3 m/s, 4 m/s, 5 m/s, 6 m/s respectively. The maximum evaporator capacity was acquired in the Fig. 5.4f by comparing the evaporator capacity results. The average velocity is 6 m/s in this situation. If T_1 , T_2 , T_3 results were compared, it's seen that maximum evaporator capacity was acquired when the blowing angle of fan is 0° . It means the optimum situation is the combination of 0° blowing angle and 6 m/s velocity.

It shouldn't be forgotten that high velocity values cause the foods to be dry in the refrigerator. So it is more suitable to investigate every graphic individually. All T_2 values are bigger than all T_1 values for every blowing angle in the Figure 5.4a. Thus 1 m/s is not suitable as average velocity of fan according to ISO 15502 Standards. The maximum evaporator capacity was acquired when the blowing angle of fan was 0° and $T_1 > T_2 > T_3$ in Figure 5.4b. The maximum evaporator capacity was acquired when the blowing angle of fan was 0° and $T_1 > T_2 > T_3$ in Figure 5.4c. The maximum evaporator capacity was acquired when the blowing angle of fan was 0° , but $T_2 > T_1$ in Figure 5.4d. The second biggest maximum evaporator capacity was acquired when the blowing angle of fan was 15° and $T_1 > T_2 > T_3$. It means that the optimum blowing angle is 15° when average velocity of fan was 4 m/s. The maximum evaporator capacity was acquired when the blowing angle of fan was 0° and $T_1 > T_2 > T_3$ in Figure 5.4e.

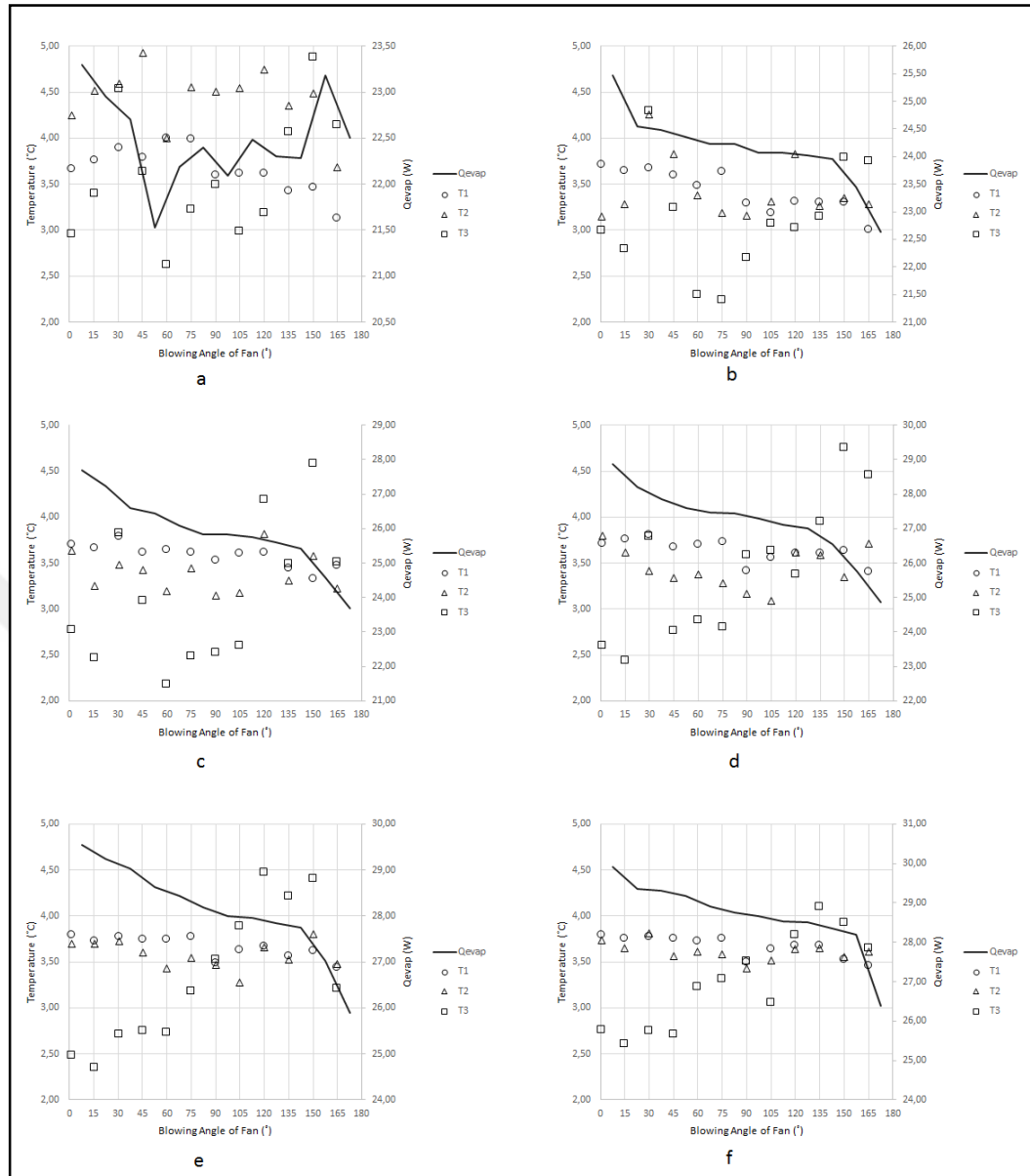


Figure 5.4 Variation of temperature due to blowing angle and velocity of fan: (a) velocity of fan is 1 m/s, (b) velocity of fan is 2 m/s, (c) velocity of fan is 3 m/s, (d) velocity of fan is 4 m/s, (e) velocity of fan is 5 m/s, (f) velocity of fan is 6 m/s

5.1.3 Effects of the Position of Fan

Figure 5.5 shows the x-position and y-position of fan box. Fan box is situated in the symmetry plane and top of the refrigerator. Fan position was investigated to see that it's the best position or not and it's effects on the heat transfer performance of evaporator.

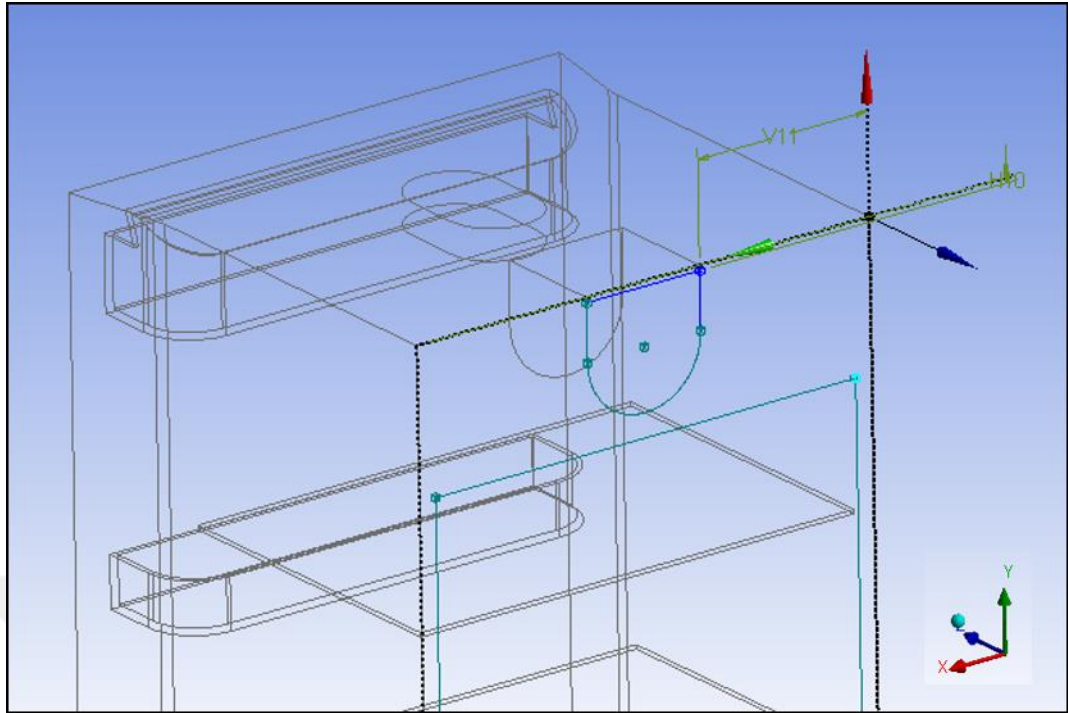


Figure 5.5 Position of fan

5.1.3.1 Effects of the y-Position of Fan

In the first instance y position of the fan is input parameter and its range is 0mm – 1070mm. Output parameters are T_1 , T_2 , T_3 , and cooling capacity of evaporator.

The results are shown in the Figure 5.6. The current situation has the maximum evaporator capacity and $T_1 > T_2 > T_3$. Because air flows down when it cools.

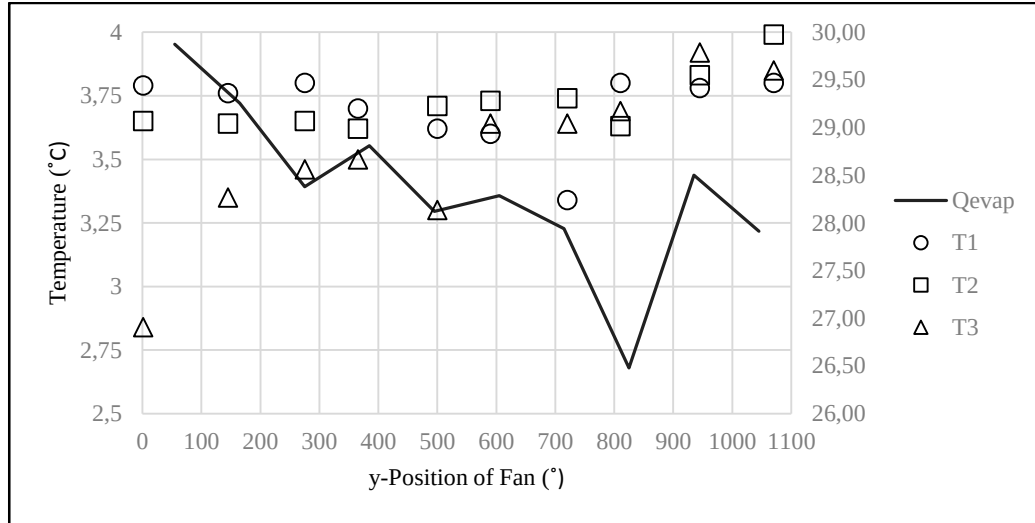


Figure 5.6 Variation of temperature due to y-position of fan

5.1.3.2 Effects of the xy-Position of Fan

In the second place, input parameters are x-position and y-position of the fan and their ranges are 0mm – 1070mm and 0mm – 360mm respectively. Output parameters are T_1 , T_2 , T_3 , and cooling capacity of evaporator.

The temperature and evaporator capacity results are shown in the Figure. 5.7. Figure 5.7a, Figure 5.7b, Figure 5.7c, Figure 5.7d, Figure 5.7e show the acquired results when fan box is on the left side of the refrigerator (x-position of fan is 0mm), between left side and symmetry plane of the refrigerator (x-position of fan is 90mm), symmetry plane of the refrigerator (x-position of fan is 180mm), between right side and symmetry plane of the refrigerator (x-position of fan is 270mm), right side of the refrigerator (x-position of fan is 360mm) respectively.

As shown in Figure 5.7e, maximum evaporator capacity was acquired when fan box was top and right side of the refrigerator and $T_1 > T_2 > T_3$. It means the optimum situation is the combination of these fan's positions. But there is no noticeable effects on evaporator capacity by changing x-position of fan because of the model is symmetric.

The maximum evaporator capacity was acquired when fan box was on top of the refrigerator (y-position of fan was 0mm) and $T_1 > T_2 > T_3$ in Figure 5.7a. The maximum evaporator capacity was acquired when fan box was on top of the refrigerator and $T_1 > T_2 > T_3$ in Figure 5.7b. The maximum evaporator capacity was acquired when fan box was on top of the refrigerator, but $T_2 > T_1$ in Figure 5.7c. The second biggest maximum evaporator capacity was acquired y-position of fan was 145mm and $T_1 > T_2 > T_3$. It means that the optimum y-position of fan is 145mm when fan box was on the symmetry plane. The maximum evaporator capacity was acquired when fan box was on top of the refrigerator, but $T_2 > T_1$ in Figure 5.7d. The second biggest maximum evaporator capacity was acquired y-position of fan was 145mm, but $T_3 > T_2$. The third biggest maximum evaporator capacity was acquired y-position of fan was 275mm and $T_1 > T_2 > T_3$. It means that the optimum y-position of fan is 275mm when fan box was between right side and symmetry plane of the refrigerator.

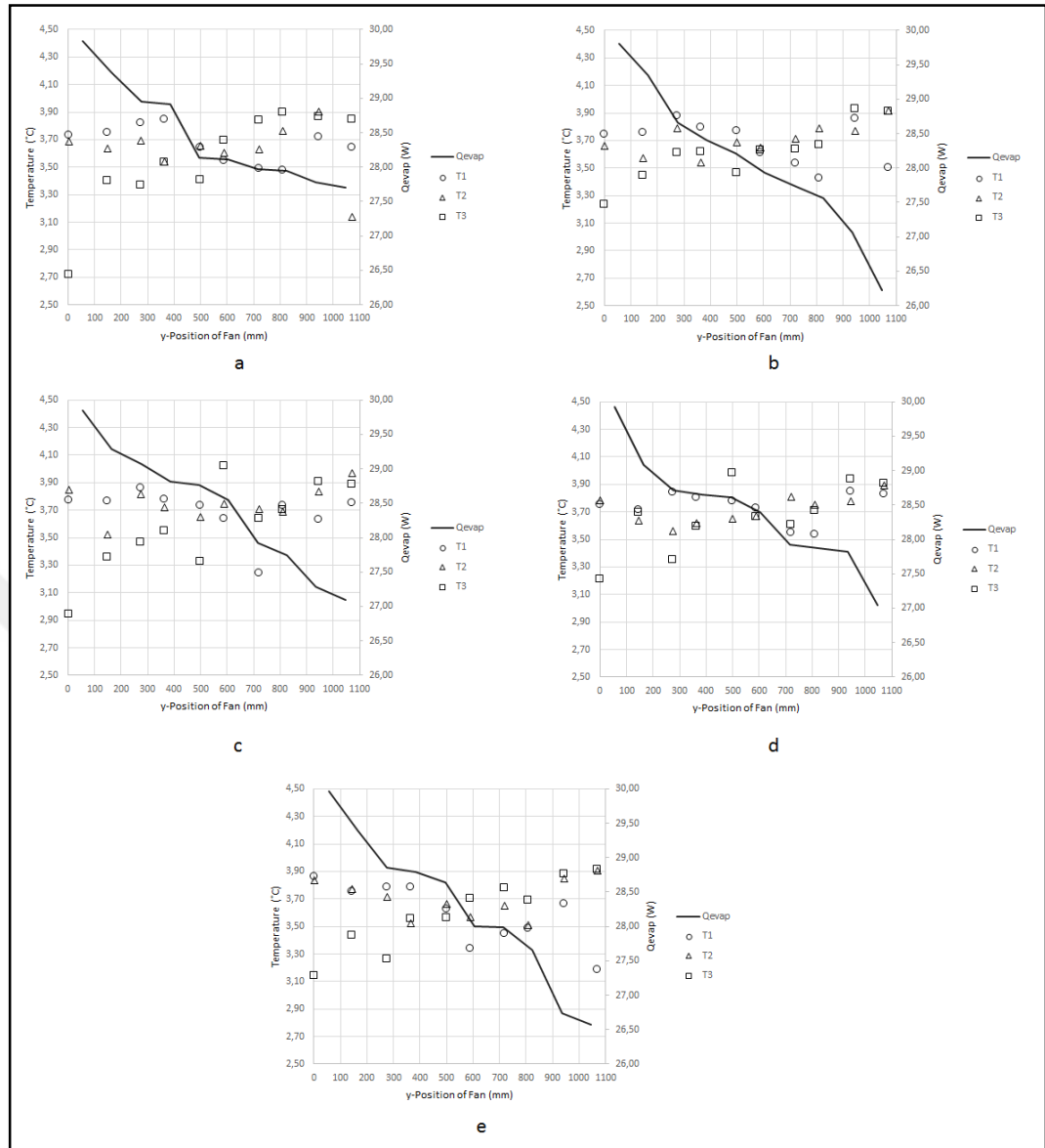


Figure 5.7 Variation of temperature due to x-position and y-position of fan: (a) x-Position of fan is 0mm, (b) x-Position of fan is 90mm, (c) x-Position of fan is 180mm, (d) x-Position of fan is 270mm, (e) x-Position of fan is 360mm

5.1.3.3 Effects of the y-Position of Fan and Evaporator Temperature

Y position of the fan is input parameter and its range is 1mm – 1070mm. Evaporator temperature is input parameter and its range -8 °C – 0 °C. Output parameters are T_1 , T_2 , T_3 and cooling capacity of evaporator.

The results are shown in the Table 5.1. The underlined situation is the current situation. The table indicates that the optimum position is the current position when evaporator temperature is -8 °C. The first situation has the maximum evaporator capacity and $T_1 > T_2 > T_3$. Because air flows down when it cools.

Table 5.1 Variation of temperature due to y-position of fan and evaporator temperature

y-Position of Fan (mm)	T _{evap} (°C)	T ₁ (°C)	T ₂ (°C)	T ₃ (°C)	Q _{evap} (W)
1	-8	3.66	3.54	1.91	49.72
1	-7	3.67	3.60	2.12	45.61
1	-6	3.68	3.66	2.33	41.51
1	-5	3.70	3.72	2.53	37.43
1	-4	3.71	3.76	2.72	33.38
<u>1</u>	<u>-3</u>	<u>3.73</u>	<u>3.81</u>	<u>2.90</u>	<u>29.35</u>
1	-2	3.76	3.85	3.07	25.27
1	-1	3.78	3.88	3.23	21.05
1	0	3.82	3.92	3.38	16.57
145	-8	3.58	3.54	2.27	48.65
145	-7	3.59	3.60	2.44	44.61
145	-6	3.60	3.66	2.60	40.59
145	-5	3.62	3.72	2.77	36.59
145	-4	3.64	3.76	2.93	32.63
145	-3	3.66	3.81	3.08	28.71
145	-2	3.69	3.85	3.22	24.75
145	-1	3.73	3.88	3.35	20.67
145	0	3.77	3.92	3.48	16.35
275	-8	3.52	3.54	2.58	47.86
275	-7	3.53	3.60	2.70	43.86
275	-6	3.55	3.66	2.84	39.90
275	-5	3.57	3.72	2.97	35.97
275	-4	3.59	3.76	3.10	32.08
275	-3	3.62	3.81	3.22	28.23
275	-2	3.66	3.85	3.34	24.36
275	-1	3.70	3.88	3.45	20.38
275	0	3.75	3.92	3.56	16.19
365	-8	3.49	3.54	2.77	47.40
365	-7	3.51	3.60	2.87	43.44
365	-6	3.52	3.66	2.98	39.51
365	-5	3.54	3.72	3.10	35.61
365	-4	3.57	3.76	3.21	31.76
365	-3	3.60	3.81	3.32	27.96
365	-2	3.64	3.85	3.42	24.13
365	-1	3.69	3.88	3.52	20.21
365	0	3.74	3.92	3.62	16.10
500	-8	3.46	3.54	3.03	46.85
500	-7	3.48	3.60	3.11	42.92
500	-6	3.50	3.66	3.19	39.03
500	-5	3.53	3.72	3.27	35.18
500	-4	3.55	3.76	3.36	31.38
500	-3	3.59	3.81	3.45	27.63
500	-2	3.64	3.85	3.53	23.86
500	-1	3.69	3.88	3.61	20.02

Table 5.1 Variation of temperature due to y-position of fan and evaporator temperature (continue)

500	0	3.75	3.92	3.69	15.99
590	-8	3.46	3.54	3.19	46.56
590	-7	3.48	3.60	3.25	42.66
590	-6	3.50	3.66	3.31	38.79
590	-5	3.52	3.72	3.38	34.96
590	-4	3.56	3.76	3.45	31.18
590	-3	3.60	3.81	3.52	27.46
590	-2	3.65	3.85	3.59	23.73
590	-1	3.70	3.88	3.66	19.92
590	0	3.77	3.92	3.74	15.94
720	-8	3.46	3.54	3.40	46.25
720	-7	3.48	3.60	3.43	42.36
720	-6	3.51	3.66	3.47	38.52
720	-5	3.54	3.72	3.52	34.72
720	-4	3.57	3.76	3.57	30.97
720	-3	3.62	3.81	3.62	27.27
720	-2	3.67	3.85	3.68	23.57
720	-1	3.74	3.88	3.74	19.80
720	0	3.81	3.92	3.79	15.87
810	-8	3.48	3.54	3.54	46.07
810	-7	3.50	3.60	3.55	42.20
810	-6	3.53	3.66	3.57	38.37
810	-5	3.56	3.72	3.61	34.58
810	-4	3.60	3.76	3.65	30.84
810	-3	3.64	3.81	3.69	27.17
810	-2	3.70	3.85	3.73	23.49
810	-1	3.77	3.88	3.78	19.74
810	0	3.85	3.92	3.83	15.84
945	-8	3.51	3.54	3.72	45.86
945	-7	3.54	3.60	3.71	42.00
945	-6	3.57	3.66	3.71	38.18
945	-5	3.60	3.72	3.73	34.41
945	-4	3.65	3.76	3.75	30.69
945	-3	3.70	3.81	3.78	27.04
945	-2	3.76	3.85	3.81	23.38
945	-1	3.84	3.88	3.84	19.66
945	0	3.93	3.92	3.88	15.80
1070	-8	3.57	3.54	3.88	45.69
1070	-7	3.60	3.60	3.85	41.84
1070	-6	3.63	3.66	3.84	38.04
1070	-5	3.66	3.72	3.83	34.28
1070	-4	3.71	3.76	3.84	30.58
1070	-3	3.77	3.81	3.86	26.94
1070	-2	3.84	3.85	3.88	23.30
1070	-1	3.92	3.88	3.90	19.61
1070	0	4.02	3.92	3.92	15.76

5.2 Effects of the Wavy Back Wall

Back wall area was increased to see the effects of evaporator area on evaporator capacity besides the temperature distribution and flow field besides. The aim of this investigation is to increase the heat transfer surface on evaporator by changing the

back wall geometry. Thus three types of wavy back wall geometry were investigated as back wall with horizontal channels, vertical channels and embosses.

5.2.1 Effects of the Wavy Back Wall with the Horizontal Channels

The area of back wall was increased by making horizontal channels on it as shown in Figure 5.8. It helps to increase the heat transfer surface area considerably. Thus it is expected that evaporator capacity is comparatively higher than current situation. The evaporator capacity was calculated as 87.87 W by numerical simulation.

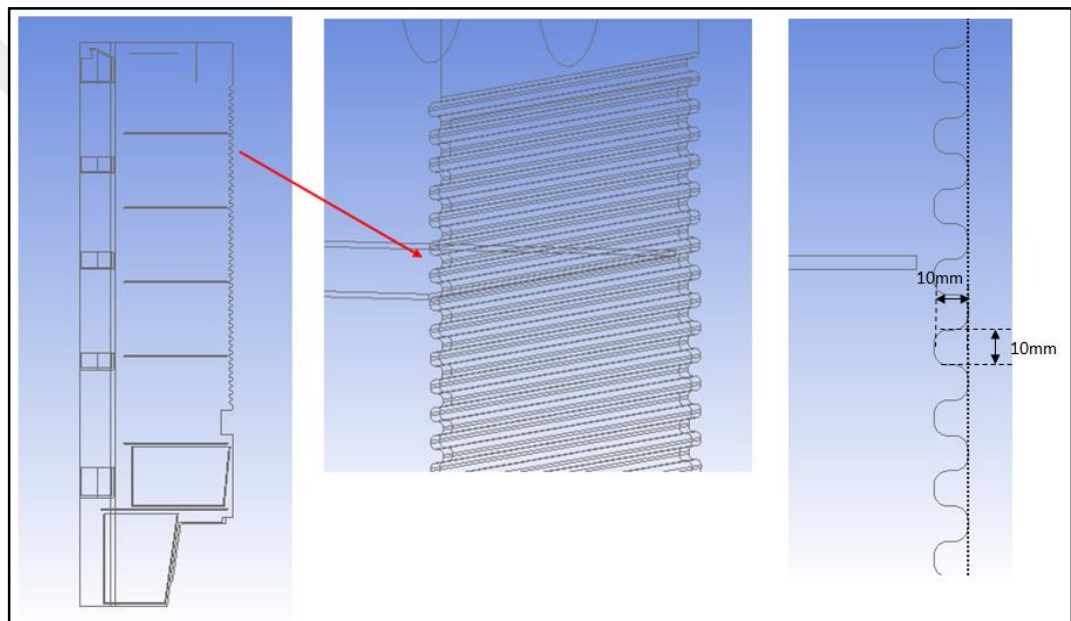


Figure 5.8 Wavy back wall geometry with the horizontal channel

Temperature distribution and flow field of the internal volume are shown in Figure 5.9. This new back wall geometry provides air circulation to reach more between the shelves.

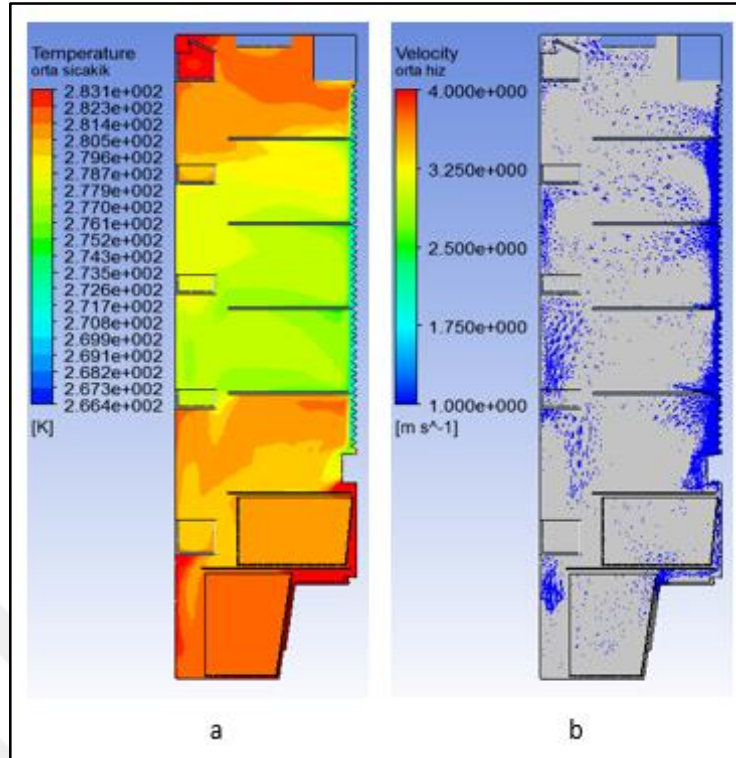


Figure 5.9 (a) Temperature distribution and (b) velocity distribution with back wall channel horizontally

There is no noticeable temperature differences between current model and new model with the horizontally wavy back wall as shown in Figure 5.10.

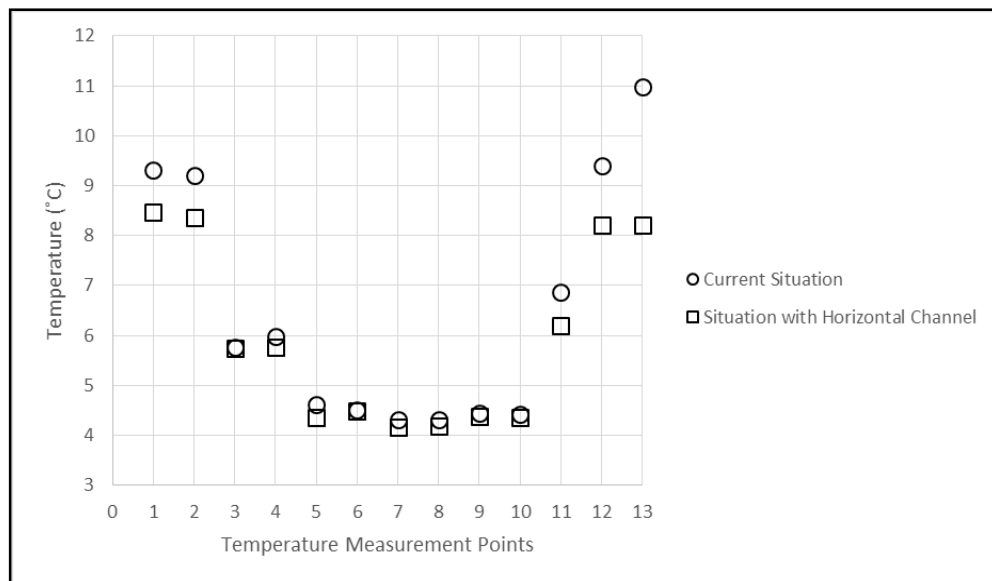


Figure 5.10 Variation of temperature due to wavy back wall geometry with horizontal channel

5.2.2 Effects of the Wavy Back Wall with the Vertical Channels

The area of back wall was increased by making vertical channels on it as shown in Figure 5.11. It helps to increase the heat transfer surface area considerably. Thus it is expected that evaporator capacity is comparatively higher than current situation. The evaporator capacity was calculated as 79.07 W by numerical simulation.

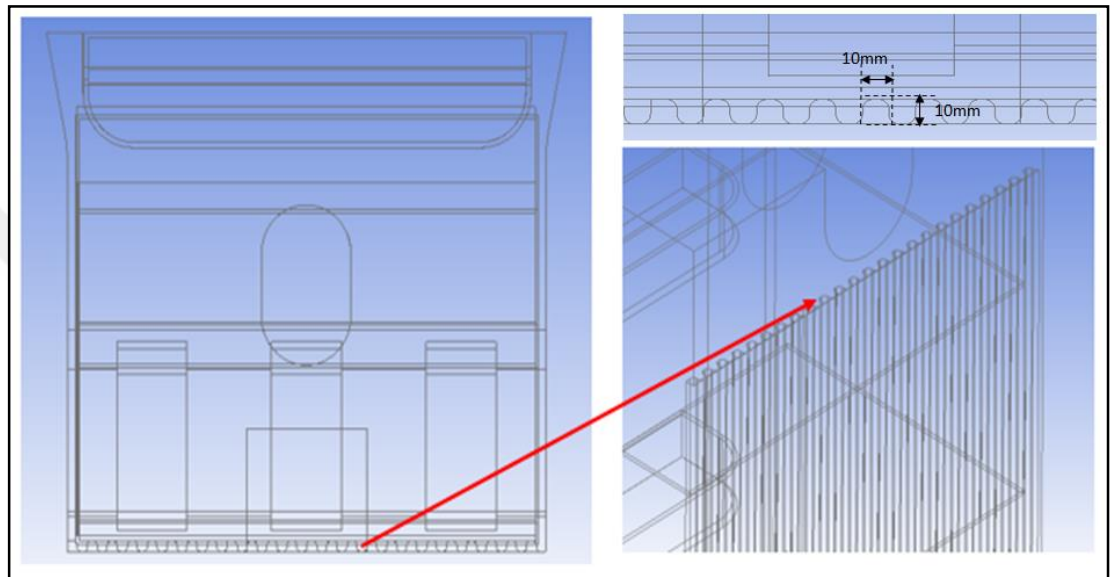


Figure 5.11 Wavy back wall geometry with the vertical channel

Temperature distribution and flow field of the internal volume are shown in Figure 5.12. This new back wall geometry provides air circulation to reach more between the shelves.

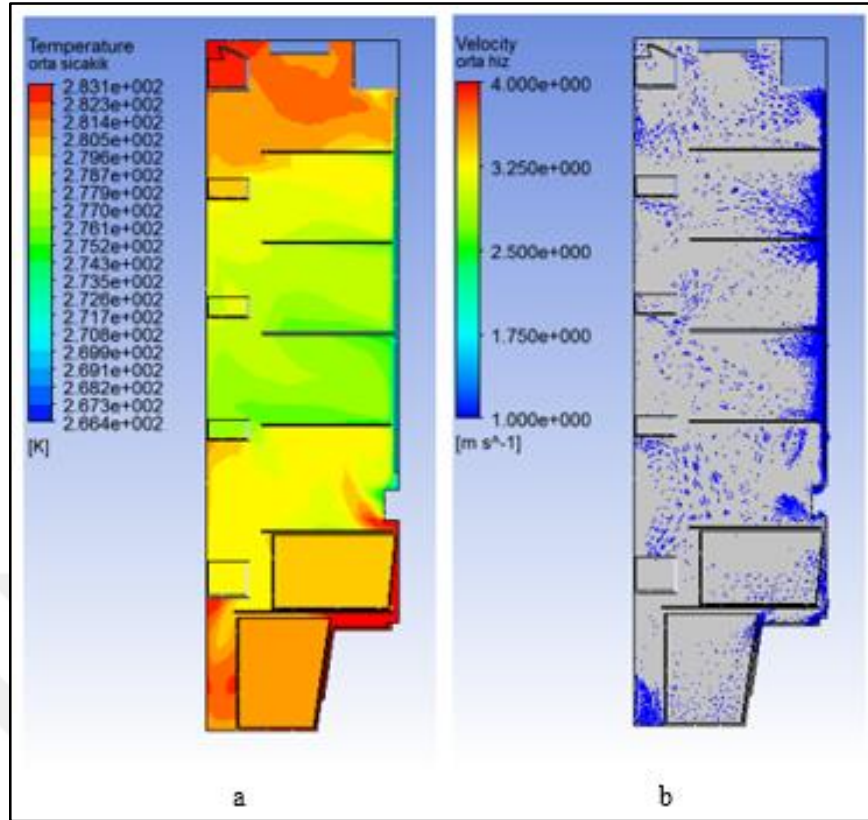


Figure 5.12 (a) Temperature distribution and (b) velocity distribution with back wall channel vertically

There is no noticeable temperature differences between current model and new model with the vertically wavy back wall as shown in Figure 5.13.

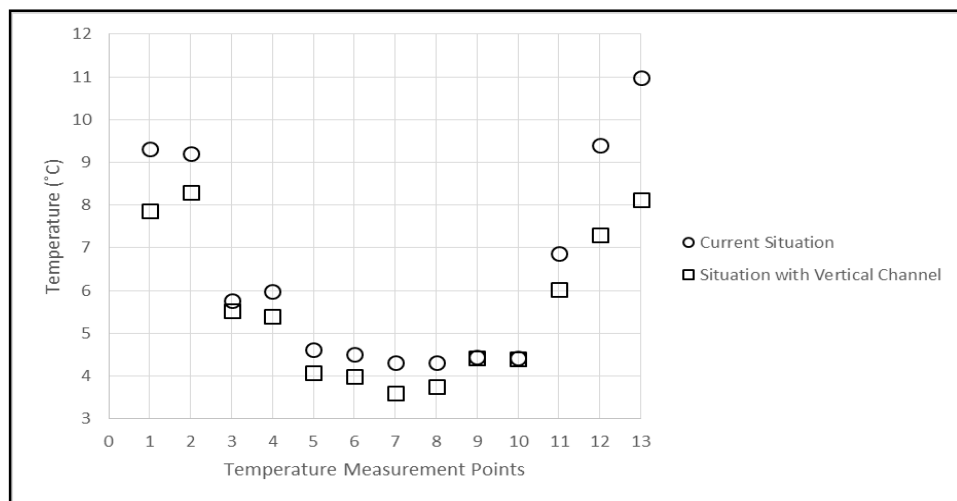


Figure 5.13 Variation of temperature due to wavy back wall geometry with vertical channel

5.2.3 Effects of the Wavy Back Wall with Emboss

The area of back wall was increased by making embosses on it as shown in Figure 5.14. It helps to increase the heat transfer surface area considerably. Thus it is expected that evaporator capacity is comparatively higher than current situation. The evaporator capacity was calculated as 63.31 W by numerical simulation.

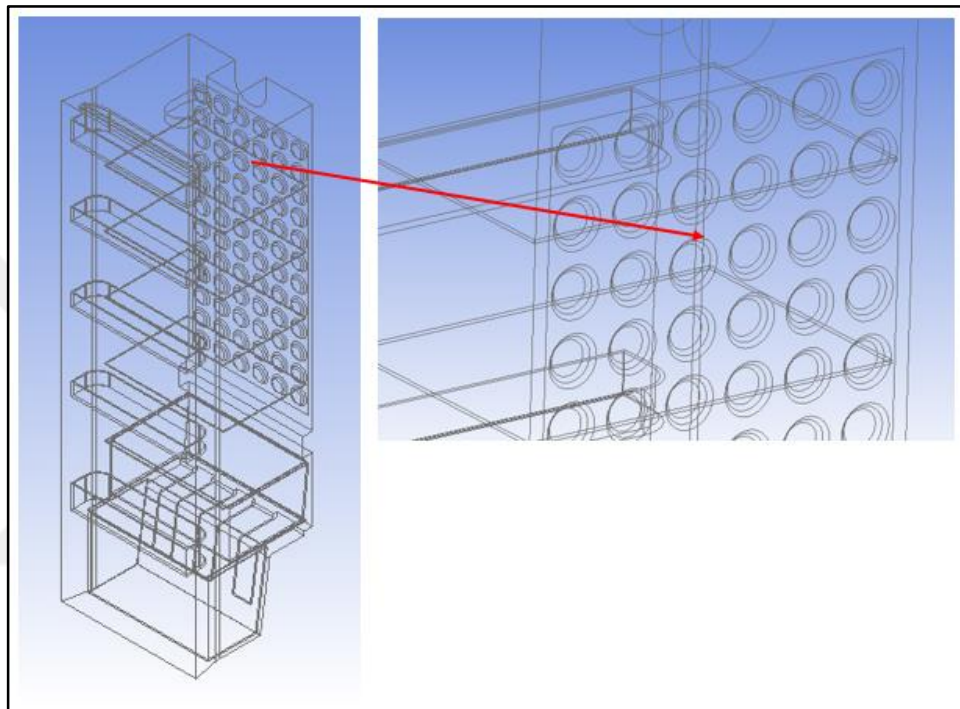


Figure 5.14 Wavy back wall geometry with emboss

Temperature distribution and flow field of the internal volume are shown in Figure 5.15. This new back wall geometry provides air circulation to reach more between the shelves.

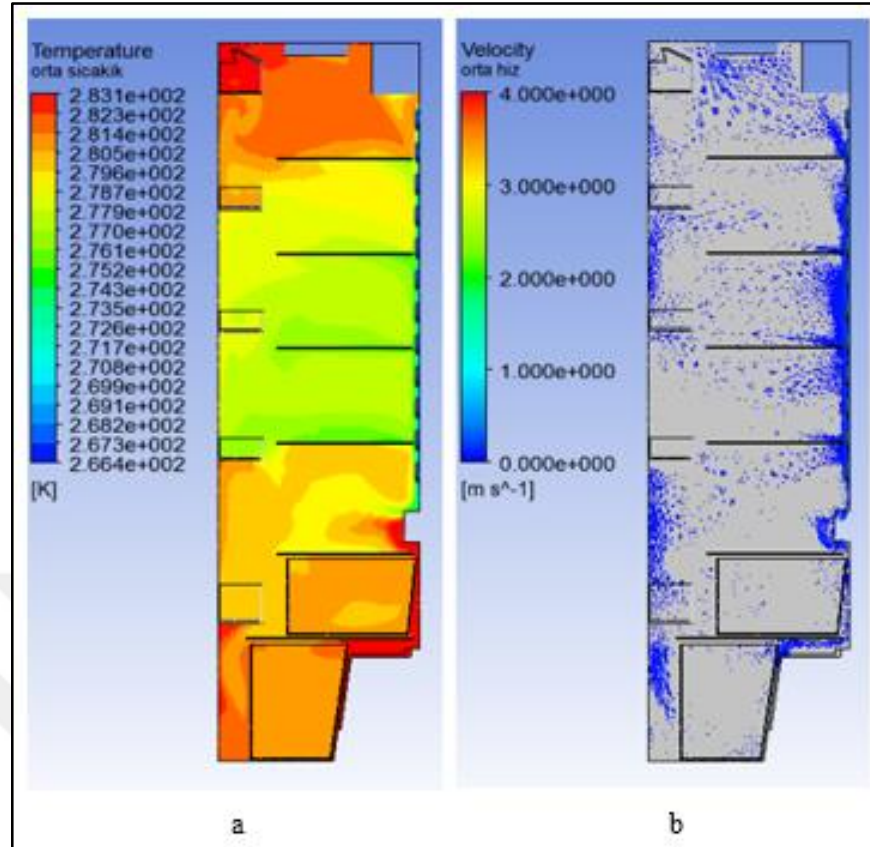


Figure 5.15 (a) Temperature distribution and (b) velocity distribution with back wall emboss

There is no noticable temperature differences between current model and new model with the embossed wavy back wall as shown in Figure 5.16.

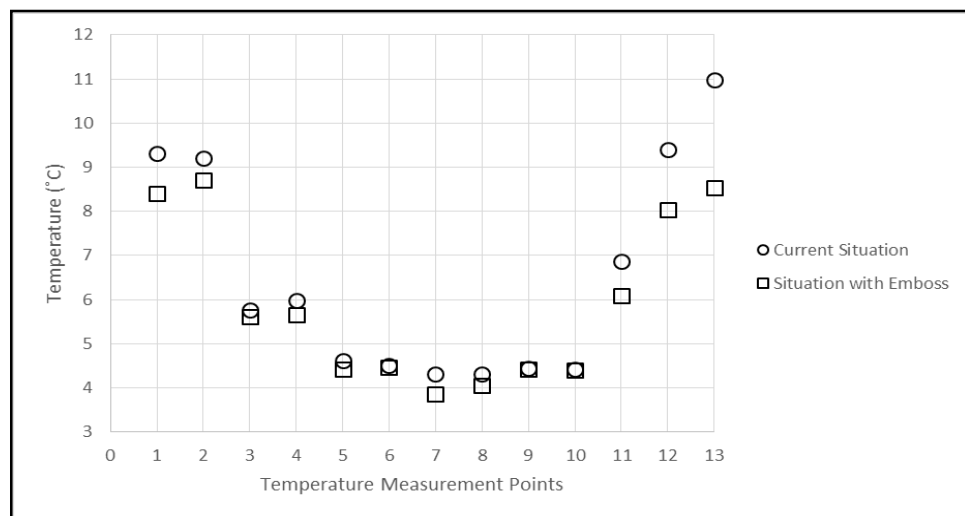


Figure 5.16 Variation of temperature due to wavy back wall geometry with emboss

CHAPTER SIX

CONCLUSIONS

The aim of this study is to determine inner design parameters of the domestic refrigerators that are important to store for long time without deterioration of the quality of food product and investigate the optimum values of these parameters. Thus, the interior air volume of a single door static type refrigerator was modeled as 3D real-like using the Computational Fluid Dynamics and Heat Transfer (CFDHT) method and analyses were made. The numerical results were validated by comparing with the experimental results and then inner design parameters were determined by using the numerical temperature distribution and flow fields of internal volume results.

As a contribution to the literature, blowing angle of fan was investigated besides the position of fan and average velocity of fan. Optimum angle values were determined according to different average velocity values of fan. Also x-position and y-position of fan box investigated together and optimum positions designated.

Different types of wavy back wall geometries as back wall with horizontal channels, vertical channels and embosses studied to increase the heat transfer surface on evaporator. It is seen that there is extremely improvement on evaporator capacity.

As a result, the calculated computational results show that the blowing angle of fan, fan position and the back wall geometry play an important role in temperature distributions and flow fields.

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