MARMARA UNIVERSITY
INSTITUTE FOR GRADUATE STUDIES
IN PURE AND APPLIED SCIENCES

THE DESIGN, DEVELOPMENT AND ANALYSIS OF A HYBRID HOUSEHOLD REFRIGERATOR COMBINING THERMOELECTRIC AND VAPOUR COMPRESSION COOLING SYSTEMS

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Ph.D. THESIS
Department of Mechanical Engineering

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İSTANBUL, 2018

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Engin SÖYLEMEZ, a Doctor of Philosophy student of Marmara University Institute for Graduate Studies in Pure and Applied Sciences, defended his/her thesis entitled "**THE DESIGN, DEVELOPMENT AND ANALYSIS OF A HYBRID HOUSEHOLD REFRIGERATOR COMBINING THERMOELECTRIC AND VAPOUR COMPRESSION COOLING SYSTEMS**", on October 8, 2018 and has been found to be satisfactory by the jury members.

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ÖZET

TERMOELEKTRİK VE BUHAR SIKIŞTIRMALI SOĞUTMA SİSTEMLERİNDEN OLUŞAN HİBRİD EV TİPİ BUZDOLABININ DİZAYNI, GELİŞTİRİLMESİ VE ANALİZİ

Termoelektrik (TE) ve buhar sıkıştırmalı soğutma sistemlerinden oluşan hibrit ev tipi buzdolapları deneysel ve nümerik olarak incelenmiştir. Bunun için, ticari olarak uygun termoelektrik soğutucular ticari buzdoplarının taze gıda ve chill bölmelerine uygulanmıştır. Analizler termoelektrik soğutma teknolojilerinin uygulama fırsatlarını arttırmak için termoelektrik soğutucuların faydalarına odaklanmıştır.

Bu çalışmanın iki aşaması bulunmaktadır. İlk aşamada, çalışmanın amaçları belirlenmiş ve takip eden hazırlıklar yapıldı: termoelektrik soğutucuların belirlenmesi için 0 °C (chill) ve taze gıda bölmeleri için soğutma yükü hesapları yapılmıştır. Daha sonra, iki referans (seri) ev tipi buzdolabının yanı sıra iki de farklı hibrit ev tipi buzdolabı üretilmiş ve deneysel olarak incelenmiştir. Daha fazla değerlendirme için, bilgisayar destekli akışkan dinamiği analizi (CFD) verileri için gerekli ölçümler yapılmıştır. İkinci safhada ise, en iyi test sonuçlarına sahip hibrit ev tipi buzdolabı sonraki değerlendirmeler için seçilmiştir. Hibrit buzdolabının nümerik analizleri sonrası, CFD modeli geliştirilmiş ve bu modelle önerilen konfigürasyonların analizleri de yapılmıştır. CFD analizinin amacı daha düzgün hava akış ve sıcaklık dağılımı elde etmede termoelektrik soğutucu için en uygun yeri belirlemektir. CFD tahminleri dikkate alınarak, en az sıcaklık tabakalaşmasına sahip konfigürasyon temel alınarak iki yeni hibrit buzdolabı yapılmıştır. CFD tahminleri test sonuçlarıyla daha sonra teyit edilmiştir.

Bu çalışmanın sonucu önemli çıktılar içermektedir. Enerji tüketiminin azaltılmasında sağlanan iyileştirmelere rağmen, hibrit buzdolaplarının enerji tüketimi, seri buzdolaplarına kıyasla hala üç kat daha yüksektir. Bu, termoelektrik soğutma teknolojisinin, ölçümlerin işaret ettiği gibi, buhar sıkıştırmalı teknolojiye alternatif olması için Performans Katsayısı'nda (COP) iyileştirmelere ihtiyaç duyduğunu vurgular. Termoelektrik soğutucuların COP'si testlerde 0.3 ila 0.5 W/W arasındaydı. Alternatif olabilmesi için, termoelektrik soğutucuların COP değerleri yaklaşık 2 W/W olmalıdır. Bununla birlikte, hibrit ev tipi buzdolapları, soğutma kapasitesi testlerinde kanıtlandığı üzere, sıcak gıdaların soğutulmasını hızlandırmak açısından muazzam bir potansiyele

sahiptir. Yani, sıcak gıda maddeleri, gıdaların güvenli bir şekilde korunabileceği hedef sıcaklık bölgelerine kadar dakikalar içinde soğutulabilir. Ayrıca, buzdolaplarının bölmesinde çok düzgün sıcaklık dağılımı ve daha az sıcaklık dalgalanmaları termoelektrik soğutucular tarafından sağlanabilir. Bu, sıcaklık farklılıklarına çok duyarlı olan bozulmaya karşı hassas gıdaların bozulmadan daha uzun bir süre tutulması için çok önemlidir.

TE teknolojisindeki gelişmeler ve yenilikler önemli ölçüde artmaktadır ve bu nedenle TE soğutma teknolojilerinin verimliliği artacaktır. Bu durum ev tipi buzdolaplarına uygulanan buhar sıkıştırmalı soğutma teknolojisine güçlü bir alternatif olmalarına yardımcı olacaktır. Bu kesinlikle biraz zaman alacak olsa da, ev tipi buzdolaplarında son kullanıcıların bazı özel ihtiyaçları için TE soğutucuları kullanmak mümkün olacaktır.

Ekim 2018 **Engin SÖYLEMEZ**

ABSTRACT

THE DESIGN, DEVELOPMENT AND ANALYSIS OF A HYBRID HOUSEHOLD REFRIGERATOR COMBINING THERMOELECTRIC AND VAPOUR COMPRESSION COOLING SYSTEMS

Hybrid household refrigerators which combine vapour compression (VC) and thermoelectric (TE) cooling technologies were experimentally and numerically investigated. In order to realise this investigation, commercially available thermoelectric coolers (TEC) were applied to the fresh food (FFC) and chill (CC) compartments of commercial refrigerators. Analysis focused on the benefits of TECs to increase opportunities for the application of the thermoelectric cooling technologies.

There were two phases of this work. In the first phase, the objectives of the work were set, and the following preparations were made: the reference refrigerators were selected and the target compartments, CC and FFC, for the implementation of the TECs were determined. The cooling loads for them were calculated to determine the suitable TECs. Another necessary test equipment was also collected. Then, two different hybrid household refrigerators were built along with two reference household refrigerators (serial refrigerators) and they were investigated experimentally. For further evaluations, necessary measurements were made to collect data for Computational Fluid Dynamics (CFD) analysis. In the second phase, the hybrid household refrigerator (HHR) with better test results was selected for further evaluations. After numerical analysis of hybrid refrigerator, CFD model was developed and then proposed configurations were also analysed. The purpose of the CFD analysis was to determine the optimum location for the TEC to achieve more uniform air flow and temperature fields. Regarding the predictions, two new hybrid refrigerators were built based on the configurations which have less temperature stratification. CFD predictions were validated with the experimental results, later on.

The conclusion of this work has pivotal outputs as follows. Although enhancements achieved in reducing energy consumption, HHRs' energy consumption is still three times higher compared to serial refrigerators. This highlights that TE cooling technology needs improvements in Coefficient of Performance (COP) to become an alternative to VC technology as measurements indicated. The COP of TECs were between 0.3 to 0.5 W/W.

To become an alternative, TEC's COP values must be around 2 W/W. Nevertheless, HHRs have an enormous potential in terms of speeding up the warm foodstuffs' cooling as proved in cooling capacity tests. That is, the warm foodstuffs can be cooled down in minutes to the target temperature regions in where foods can be safely preserved. Moreover, highly uniform temperature distribution and less temperature fluctuations in the compartment of the refrigerators can be provided by the TECs. This is crucial for keeping the perishable foods, which are very sensible to temperature differences, for a longer time without deterioration.

The developments and innovations in TE technology are significantly increasing and therefore the efficiency of TE cooling technologies will likely increase which help them to be a strong alternative to VC cooling technology applied to household refrigerators. Although this will certainly take some time, it will possible to use TECs for some specific needs of end-users in household refrigerators.

October 2018 **Engin SÖYLEMEZ**

CLAIM FOR ORIGINALITY

THE DESIGN, DEVELOPMENT AND ANALYSIS OF A HYBRID HOUSEHOLD REFRIGERATOR COMBINING THERMOELECTRIC AND VAPOUR COMPRESSION COOLING SYSTEMS

The originality of this work can be summarised as follows:

1. It includes a whole, key performance analysis of the hybrid household refrigerators to gain a complete understanding of them. The author did not find any prior experimental work on the same test unit addressing the various aspects of the type of refrigerators studied here.
2. It measures the primary components of household refrigerators' performance, such as (i) different ambient temperatures (ii) operating conditions (iii) loading conditions. Several types of tests were carried out to comply with different standards to measure its performance in various conditions.
3. It provides numerical (CFD) and experimental data of a household refrigerator including VC and TE cooling technologies. To the author's knowledge, there has been no such work so far which presents air temperature fields and flow distributions inside the refrigerator compartment cooled by a TEC.
4. This work experimentally proved that despite the higher energy consumptions observed, rapid cooling can be achieved by implementing thermoelectric coolers.

October 2018 **Assoc. Prof. Dr. Emre ALPMAN** **Engin SÖYLEMEZ**

SYMBOLS

A	Area (m^2)
c_p	Specific heat in constant pressure (kJ/kg°C)
c_{pack}	Specific heat of the packages (kJ/kg°C)
D	Depth of the compartment (mm)
d_F	Fan diameter (mm)
G	Acceleration due to gravity (9.81 m/s^2)
H	Height of the compartment (mm)
H_T	Height of the TEC
I	Electric current (A)
K	Thermal conductivity (W/m°C)
L	Thickness (m)
M	Mass (kg)
Q	Electrical power (W)
Q_A	Active heat load (W)
$Q_{A,TEC}$	Thermoelectric cooler active heat load (W)
Q_C	Cooling effect (W)
$Q_{C,TEC}$	Thermoelectric cooler cooling effect (W)
Q_H	Heating effect (W)
$Q_{H,TEC}$	Thermoelectric cooler heating effect (W)
Q_P	Passive heat load (W)
Q_T	Total heat load (W)
T	Temperature (K)
T_a	Ambient temperature (K)
T_{cc}	Target temperature of chill compartment (°C)
ΔT	Temperature difference (°C)
UA	Overall transmission losses (W/°C)
V	Voltage (V)
$\vec{V}$	Velocity vector (m/s)
∇	Gradient operator
W	Width of the compartment (mm)

W_T	Width of the TEC (mm)
W_i	Width of the TEC inlet (mm)
$W_{comp,in}$	Compressor power (W)
X_s	The mass fraction of the solids in the food (%)

Greek symbols

P	Air density (kg/m^3)
μ	Dynamic viscosity (Pa s)
Φ	Viscous-dissipation function

ABBREVIATIONS

CC	Chill compartment
CFD	Computational fluid dynamics
Comp	Compressor
COP	Coefficient of performance (W/W)
CRO	Chemical reaction optimisation
CRTR	Compressor run time ratio (%)
Cu	Copper
DC	Direct current (A)
EEC	Energy efficiency class
EN	European norm
FC	Freezer compartment
FEM	Finite element method
FFC	Fresh food compartment
GWP	Global warming potential
HC	Hydrocarbon
HFC	Hydrofluorocarbon
ICN	Institute of Child Nutrition
ICRT	International Consumer Research Testing
IEC	International Electrotechnical Commission
IV	Idling voltage (V)
Ma	Mach number
ODP	Ozone depletion potential
POD	Proper orthogonal decomposition
RPM	Revolutions per minute
RTR	Run time ratio (%)
SLHX	Suction line heat exchanger
TE	Thermoelectric
TEC	Thermoelectric cooler
VC	Vapour compression

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1. INTRODUCTION

1.1. General Information

Compressor based vapour compression (VC) technology, which has a higher coefficient of performance (COP) and lower cost in comparison with its alternative technologies, has the highest share of usage in domestic refrigerators. In this technology, a refrigerant gas is used as coolant. The most widely used refrigerants are R134a and R600a. R134a is an HFC refrigerant that needs to be reduced because of its high global warming potential (GWP) in the coming years. On the other hand, R600a is a natural refrigerant, a hydrocarbon (HC) and getting more attention due to its low GWP and the ozone depleting potential (ODP) of zero. It is used in more than 700 million household refrigerators worldwide, and it is estimated that by 2020 that R600a will have a 75% share of global production [1]. However, it has a drawback of being flammable; hence it faces with some restrictions in use.

In addition to using a refrigerant, one of the disadvantages of VC systems is that it is difficult to control the temperature of the target compartment when a single speed compressor is used in the system. Only the VC systems equipped with variable speed compressors and electronic expansion valves are capable of precise temperature control. The temperature control is especially important for preserving food products which are sensitive to frequent temperature changes.

Due to the drawbacks of VC technology, a great deal of effort has recently been given to alternative cooling technologies. Thermoelectric/Peltier (TE) cooling is one of those alternatives and it is studied in this work because it is compact and has fewer moving parts. More importantly, there is no refrigerant used. The technology is simply based on the principle of Peltier effect that "*the passage of an electric current through the junction between two dissimilar conductors in a certain direction produces a cooling effect*" [2]. Moreover, it is also possible to control the inner temperature of target compartment with a higher degree of accuracy. However, thermoelectric coolers (TECs) have lower COP values and higher costs. Although there are no moving parts in its core system, fans can be installed on the heat sinks in order to increase heat transfer over its surfaces. This can cause high noise levels.

TE cooling has been studied extensively since the introduction of semiconductors in the 1950s [3] which led to the development of more efficient TECs. The studies up to now focused on design optimisation, performance improvement, and its materials. TE cooling and its comparison with other cooling technologies have also been investigated [4, 5]. Hermes & Barbosa (2012) [4] compared the thermodynamic performance of the portable coolers that employ TE, Stirling, and VC cooling systems. In their study, the Stirling and the reciprocating VC refrigeration systems presented overall thermodynamic efficiencies of around 14%, whereas the TEC presented approximately 1%. Bansal & Martin (2000) conducted a similar study [5] that investigated the performance characteristics of three domestic refrigerators, namely the vapour compression, the TE and the absorption refrigeration. Their study found that the vapour compression refrigerator had the highest COP (2.59) followed by the TE (0.69) and the absorption refrigerator (0.47), respectively.

Besides, significant efforts have been given to the application of thermoelectric cooling technology to the different cooling systems. The technology has a potential to meet the refrigeration demands in aerospace applications [6, 7] and low temperature applications by means of transverse Peltier coolers [8, 9]. One of the most popular applications of the technology is electronic cooling due to its compact size, low mass and vibration-free operation [10-15]. Space and automotive air-conditioning systems are another potential field of application for thermoelectric cooling cited by the researchers [16-22].

The application of this technology to the household refrigerators has also attracted the researchers' attention [23-30]. Vian & Astrain (2009a) [23] and Astrain et al. (2012) [24] conducted experimental studies to apply thermoelectric cooling technology to household refrigerators. Those studies concentrated on the electric power consumption and efficiency of hybrid refrigerators combining thermoelectric and compressor-based cooling systems. Development and validation of a computational model for the simulation of the complete hybrid refrigerator was another focus in their study. Astrain et al. (2016) [25] investigated the effect of different heat exchange systems (water-air heat exchanger, finned heat sink, and heat pipes) on the efficiency of thermoelectric refrigerator. Their study showed that the higher COP values for TE refrigerators can be achieved through the optimal design of the heat exchangers. Vian & Astrain (2009b) [26] developed a domestic refrigerator, without moving parts, with only the fresh food

compartment cooled by TECs. To increase heat transfer, a thermosyphon with two-phase and capillary lift design was applied on the cold side of TEC, whereas another thermosyphon, with two-phase and without fan was applied to the hot side of TEC. The experimental results presented that the compressor-based refrigerator had half the electric power consumption of the TE refrigerator. In another work, Martinez et al. (2013) [27] studied experimentally the influence of different temperature control systems on the electric power consumption and COP of a TEC. Systems studied included: an on/off controller, a proportional-integral-derivative controller, and a novel operating system based on idling voltages. Their study proved that the controller based on idling voltages decreases the electric power consumption of the refrigerator by 32% and increases the COP by 64%, compared with the on/off controller. Riffat et al. (2001) [28] investigated the application of heat pipes and phase change materials for TE refrigeration. They designed and constructed two different TE refrigerators: the first one employed a conventional heat sink system (bonded fin heat sink) on the cold side of TEC. The other had an encapsulated phase change material in place of the conventional heat sink. Both refrigerators employed heat pipe embedded fins as the heat sink on the hot side. In their study it was found that application of encapsulated phase change material improved the performance of TE refrigerator. Jugsujinda et al. (2011) [29] analyzed a TE refrigerator performance. The study showed that the TE refrigerator's maximum COP was 0.65 W/W. Min & Rowe (2006) [30] developed prototypes TE domestic refrigerator and they evaluated them regarding the COP, heat-pumping capacity and cooling-down rate. They found that the COP of a TE refrigerator was around 0.3-0.5 W/W for the operating temperature of 5 °C with ambient temperature at 25 °C.

Modelling and numerical analysis have been developed and adopted for the TECs in different studies. The chief parameters marked mainly in those studies are Seebeck, Peltier, Thomson and Joule effects. Yamanashi (1996) [31] analysed TEC's optimum design conditions in a system with heat exchangers. The aim was to achieve the best performance under a constant temperature difference and heat load. Heat balance equations were presented as well as dimensionless entropy flow equations, exergy flow and exergetic efficiency. He concluded that the hot side heat exchanger of the TEC has more considerable effect on degrading the overall TEC system than does the cold side

heat exchanger. Fraisse et al. (2013) [32] focused on TE leg and proposed to compare different simplified models with the more accurate ones. These models are standard and improved simplified, analytical, electrical analogy, and numerical models based on finite element method (FEM) software. Their study presented that the electrical analogy model, which gives almost the same results with FEM, is more efficient. Perez-Aparicio et al. (2012) [33], combined FEM with Monte Carlo technique to simulate a commercial TE cell. Kim et al. (2015) [34] developed computer software to predict the Seebeck and Peltier effects for TE devices. A novel approach for optimisation of two-stage TE refrigeration systems using a chemical reaction optimisation (CRO) algorithm was proposed by Hadidi [35].

The numerical simulation studies for household refrigerators through computational fluid dynamics (CFD) have drawn attention because the method successfully predicts the air temperature and velocity distribution in the compartment to be simulated. It is critical to comprehend the air temperature and flow characteristics inside the compartments because it provides insight into the cooling performance of the refrigerator. Evenly distributed cooled air inside the compartment averts temperature stratification, that is, it prevents the formation of warmer regions which cause the deterioration of the perishable foodstuff [36-38]. There are works conducting the comparison of various simulation tools, such as between CFD and Artificial Neural Network (ANN) [39-41], between CFD and Proper Orthogonal Decomposition (POD) [42]. In these works, it is shown that the ANN and POD methods produced satisfactory results along with CFD.

The CFD method also provides insights about the heat leakages through the door gaskets of the refrigerators as presented by Miller et al. [43]. Researchers are interested in the impact of radiation on the cooling performance of household refrigerators by means of CFD; they indicated that the radiation affects air circulation, natural convection and heat transfer inside the refrigerators [44-46]. Numerical and mathematical models have been developed and verified through the experimental tests for different types of household refrigerators under different conditions [47-53].

Limited simulation analysis has been performed on the refrigerators in which a TEC is installed. A computational model was developed and advanced to analyse the thermal and electric performance of a Peltier based thermoelectric refrigerator [54, 55]. This

model's results were validated with the experimental test results and thus, it was proved that using TE cooling technology can be beneficial to household refrigerators.

This work has differences in terms of the investigation of a TEC application to household refrigerators and its performance analysis. The originality of this work can be summarised as follows:

* It includes a whole, key performance analysis of the hybrid household refrigerators to gain a complete understanding of them. The author did not find any prior experimental work on the same test unit addressing the various aspects of the type of refrigerators studied here.
* It measures the primary components of household refrigerators' performance, such as (ⅰ) different ambient temperatures (ⅱ) operating conditions (ⅲ) loading conditions. Several types of tests were carried out to comply with different standards to measure its performance in various conditions. The tests in this study were performed according to valid household test standards EN 62552:2013 [56], new IEC 62552:2015 [57], IC 15068 [58], and IEC 60704-2-14 [59]. Experimental studies involving these three standards were not found in the literature. As will be discussed in the following sections, even though energy consumption is of prime importance to the measurement of refrigerator's performance, there are other key aspects such as cooling and freezing capacity, and noise level which have direct effect on the satisfaction of end-users.
* It provides numerical (CFD) and experimental data of a household refrigerator including VC and TE cooling technologies. To the author's knowledge, there has been no such work so far which presents air temperature fields and flow distributions inside the refrigerator compartment cooled by a TEC.
* This work experimentally proved that despite the higher energy consumptions observed, rapid cooling can be achieved by implementing thermoelectric coolers.

1.2. Objectives of the Work

The principal objectives of this experimental and numerical work are:

* To build a hybrid household refrigerator (HHR) like the one explained by Vian & Astrain (2009a) [23], combining both vapour compression and thermoelectric cooling technologies.
* To measure the influence of the TECs applied not only to chill compartment (CC) but also the fresh food compartment (FFC), which has different cooling power requirements due to the different target temperatures and volumes of the respective compartments.
* To understand the effect of current commercial TECs on the performance of household refrigerators in terms of energy consumption, cooling and freezing capacity, and noise tests.
* To concentrate on the benefits of TECs to provide opportunities for the application of TE
* To predict air flow and temperature distribution inside the FFC by Computational Fluid Dynamics (CFD) analysis for optimisation purposes.
* To enhance overall performance of the HHR by additional improvement measures.

2. MATERIALS AND METHODOLOGY OF THE WORK

2.1. Methodology of the Work

It will be very serviceable to employ a convenient methodology for the current work to reach its objectives.

There were two phases of this work. In the first phase, the objectives of the work were set, and the following preparations were made: the reference refrigerators were selected and the target compartments, CC and FFC, for the implementation of the TECs were determined. The cooling loads for them were calculated to determine the suitable TECs. Another necessary test equipment was also collected. Then, two different HHRs (HHR-1 and HHR-2) were built along with two reference household refrigerators (serial refrigerators) and they were investigated experimentally. Energy consumption, cooling and freezing capacity and noise tests were conducted in compliance with the standards of EN62552:2013 [56], IEC62552:2015 [57], IC 15068 [58] and IEC60704-2-14 [59]. For further evaluations, necessary measurements were made to collect data for CFD analysis. In the second phase, HHR-2 was numerically analysed due to its better test results compared to HHR-1. The air flow field and temperature distributions in the FFC of HHR-1 were predicted by CFD analysis. The purpose of the CFD analysis was to determine the optimum location for the TEC. For this, five different configurations were analysed. Regarding the predictions, two new HHRs (HHR-3 and HHR-4) were built based on the configurations which have more uniform air flow and temperature distribution. In this phase, only two tests, energy consumption and cooling capacity, were performed complying with IEC62552:2015 [57]. This standard was opted for: (1) being a new standard; thus, there is no test data with it except former study of this work (2) easier execution of the tests: no packages are used (except warm packages used for cooling capacity tests) minimising inconsistency and uncertainty of the tests. Additional improvements were also implemented for further developments. The methodology of the work is briefly presented in Figure 2.1.

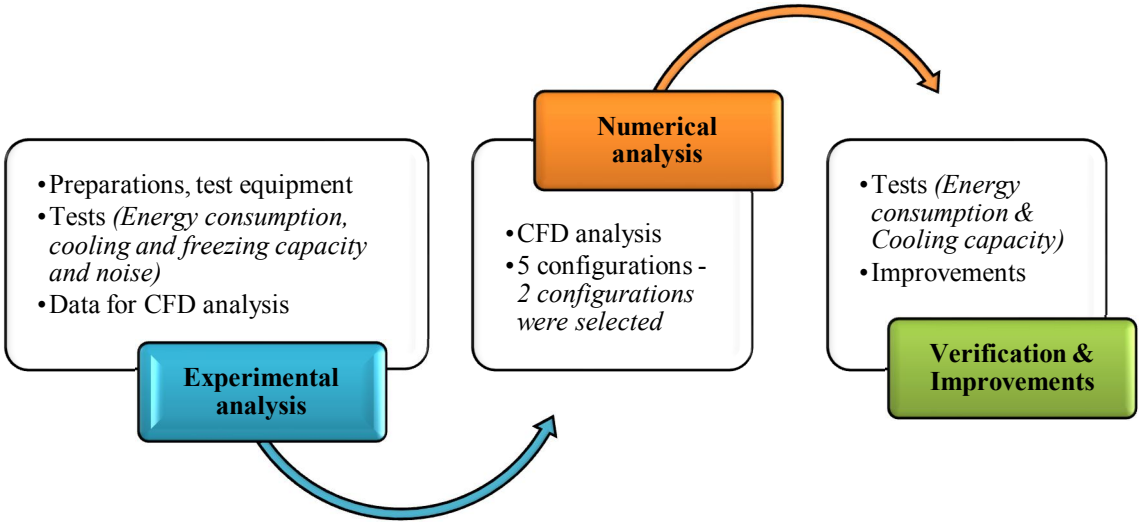


Figure 2.1. Methodology of the work

2.2. Experimental Analysis

2.2.1. Specifications of Serial Refrigerators

The serial refrigerator, KGN39 premium produced by Bosch, is a bottom freezer type with two doors and the following three compartments (see Figure 2.2): *fresh food compartment (FFC), chill compartment (CC) and freezer compartment (FC)*. Its specifications are shown in Table 2.1.



Figure 2.2. Serial refrigerator

Table 2.1. Specifications of the analysed serial refrigerators

Specifications	Units	Declaration values
Charge	g	46.0
Fresh food net volume	l	175.0
Chill net volume	l	81.0
Freezer net volume	l	87.0
Total net volume	l	342.0
Energy consumption, daily	kWh/d	0.492
EEC	-	A+++
Climate classes	-	SN-T
Noise level	dB(A)	38

FFC and CC are separated from each other by a division plate. This refrigerator contains a vapour compression cooling system presented in Figure 2.3. It has a single cooling circuit including a variable speed compressor, an air-cooled evaporator and two skin condensers in the side walls. The warm air is first cooled in the evaporator then, cooled air is pumped by a fan to the compartment by means of air channels. All compartments have their own thermostats to maintain inner temperatures at desired levels. Due to this operation mode, there is no simultaneous cooling of the compartments. On the other hand, the hybrid prototypes developed in the references Vian & Astrain (2009a) [23] and Astrain et al. (2012) [24], were based on the model of Bosch KGP39331. It has two compartments, fridge and freezer, and two compressors which enable independent cooling of the compartments. The two-compressor based system has a potential of being more efficient and having higher cooling and freezing capacity than the one-compressor system. However, more space is required for the compressors, system complexity is increased, and maintenance costs may go up. Finally, it may increase the noise level.

In order to collect reference testing data, two non-modified serial refrigerators named, Serial-1 and Serial-2, were examined beforehand. Their test results were compared with each other for validation purposes.

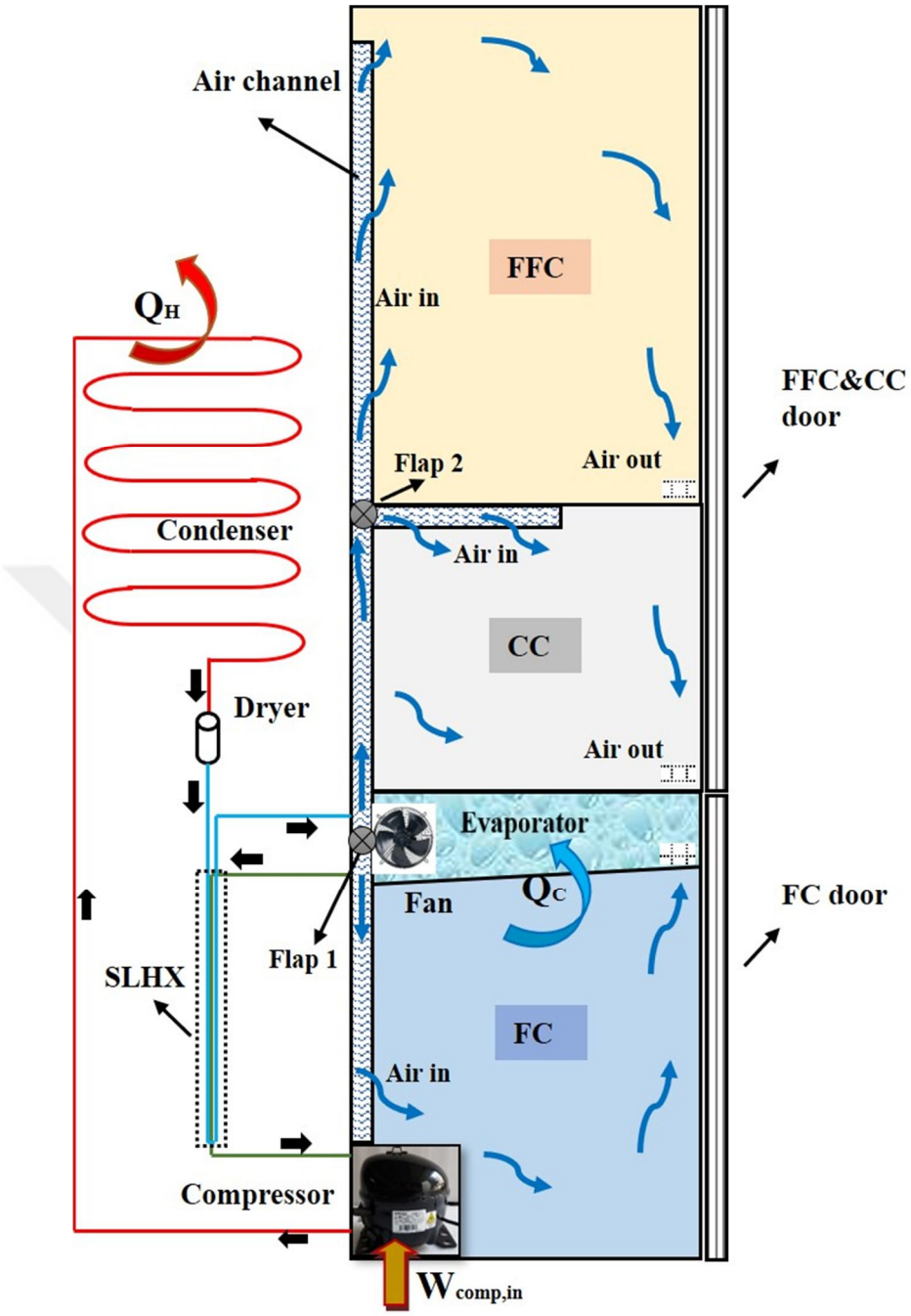


Figure 2.3. Schematic diagram of the vapour cooling system of the analysed serial refrigerator

The target temperatures of the compartments will be presented in the following sections. These targets can change depending on the standards, test specifications and the type of tests.

2.2.2. Thermoelectric Coolers

In this work, a commercial TEC manufactured by Marlow was chosen according to the heat load requirements of the compartments in where it was located. The heat load requirements' calculations are based on the measurements of overall transmission losses and the calculations stated in the following subsections of 2.2.2.1 and 2.2.2.2, respectively.

The properties of the TEC are shown in Table 2.2. The cooling technology is based on the system represented schematically in Figure 2.4. In the core of the system, there are P and N semi-conductors, connected with each other electrically in series and thermally in parallel, which are sandwiched between the ceramic materials. In order to increase heat transfer between the faces of the TEC and air, the heat dissipaters comprised of aluminum fins and fans are attached to the ceramics. Basically, on both faces of the TEC, air is withdrawn by the fans, moves through the fins and finally leaves the cooler. On the cold face, the cabinet air gets colder while ambient air gets warmer on the hot face.

Table 2.2. Thermoelectric coolers applied to prototypes

Model	CTA305 (TEC)
Capacity [W]	143
Heat transfer	Air to air
Voltage [V]	DC, 24V
Current [A]	7.5
External fan rating [dB(A)]	51

2.2.2.1. Overall Transmission Losses of the Cabinets

The overall transmission losses (thermal conductance) of the serial and hybrid prototype refrigerators, (kA)/L, which will be referred to as UA [W/°C] (Here U stands for k/L) were measured (Equation 2.1). Here, a controllable heat flow rate is supplied by a heating resistor of 30 W which is located inside the compartment. A constant ambient temperature of 16 ºC is provided in the test chamber during the whole measurement. The heaters increase the interior temperatures of the compartments and keep it around 40 ºC. After reaching stabilization, the test stops. The electrical power supplied by the heating

resistor is measured as well as the temperature difference between the inner temperature of the compartments and ambient temperature. Then, the UA value is calculated using Equation 2.1.

$$UA = (kA)/L = Q/\Delta T \quad [W/°C] \quad (2.1)$$

Experimental measurement is the most practical and effective method for calculating these transmission losses. This not only enabled the author to choose suitable thermoelectric coolers in terms of cooling load but also helped him understand heat loss through the walls and effect of insulation on the energy consumption.

The measured UA results of each model are shown below Table 2.3.

Table 2.3. UA results

Models	**FFC** **W/°C**	**CC** **W/°C**	**FC** **W/°C**
Serial-1	0.648	0.449	0.378
Serial-2	0.654	0.441	0.368
HHR-1	0.771	0.495	0.376
HHR-2	0.710	0.482	0.370

The results show that the transmission losses through the walls of serial units' compartments are in close alignment; the differences are under 3%.

Compared to Serial-1, the UA of HHR-1's FFC and CC are higher by 18.9% and 10.2%, respectively. Similarly, the UA of HHR-**2**'s FFC and CC are higher than those of Serial-2 by 8.5% and 9.2%, respectively. Those results are expected because there are heat leakages through the locations on the hybrid refrigerators' walls where TECs are mounted. The higher transmission losses of the prototypes surely lead to higher energy consumption.

2.2.2.2. Heat Load Calculations

For the purpose of selecting the most suitable TEC, the basic passive and active heat loads were calculated for FFC and CC in consideration of various scenarios because the

tests were performed at different room temperatures (ranging from 16 °C to 32 °C) at different loading conditions. TEC was determined taking into account of the maximum heat load conditions.

Passive heat loads (Q_P) are formed by the heat transfers through the compartments' wall and door gaskets (transmission losses). After measuring UA as explained in the earlier section, it was calculated by considering the differences between the ambient and compartment temperatures (Equation 2.2).

$$Q_p = UA\Delta T \qquad [W] \qquad (2.2)$$

Active heat loads (Q_A) arises from (1) Heat transfer through TEC, $Q_{A,TEC}$ (some of the heat produced by the Peltier moves into the compartment). It was provided by the manufacturer and its value was about 50% of passive heat load. (2) Inserting food into the compartments (because test packages (Tylose packs) were used in the tests rather than food, the properties of Tylose packs were used in the calculations*)*. Total active heat loads are calculated as presented in Equation 2.3.

$$Q_A = Q_{A,TEC} + mc_{pack}\,\Delta T \qquad [W] \qquad (2.3)$$

In where, c_{pack} is the specific heat of the package. For an unfrozen food, it is calculated by Equation 2.4 [60]:

$$c_{pack} = 4.19\text{-}2.30Xs\text{-}0.628Xs^3 \qquad [kJ/kg\,°C] \qquad (2.4)$$

In where, Xs is the mass fraction of the solids in the food (= 23.6% for the packs).

Total heat loads (Q_t) are the sum of all passive and active loads and simply shown in Equation 2.5.

$$Q_T = Q_P + Q_A \qquad [W] \qquad (2.5)$$

The maximum heat load conditions occur when warm foods are inserted into the compartments. This condition can be simulated by cooling capacity test performed at the ambient temperature of 25 °C mentioned in Section 3.1.3. As a result of calculations, the total heat loads of FFC and CC were roughly 45 W and 21 W, respectively.

2.2.3. Samples for HHR-1 and HHR-2

Within this work, the term “hybrid refrigerator” refers to the refrigerator in which both VC and TE cooling systems were implemented. They can refrigerate separate compartments at the same time thanks to their independent operation from each other. This brings extra flexibility in use. Here, when a TEC is applied for a compartment, the air channel openings in that compartment were sealed to eliminate the vapour compression cooling effect.

In the first phase of the work, as mentioned before, two prototype hybrid refrigerators, in Figure 2.4 and 2.5, were built after the modification of serial refrigerators (Serial-1 and Serial-2). HHR-1 was created by modifying Serial-1. TEC was used for the cooling of CC in this prototype. On the other hand, HHR-2 was created by modification of Serial-2. Here, TEC is used for cooling the FFC.

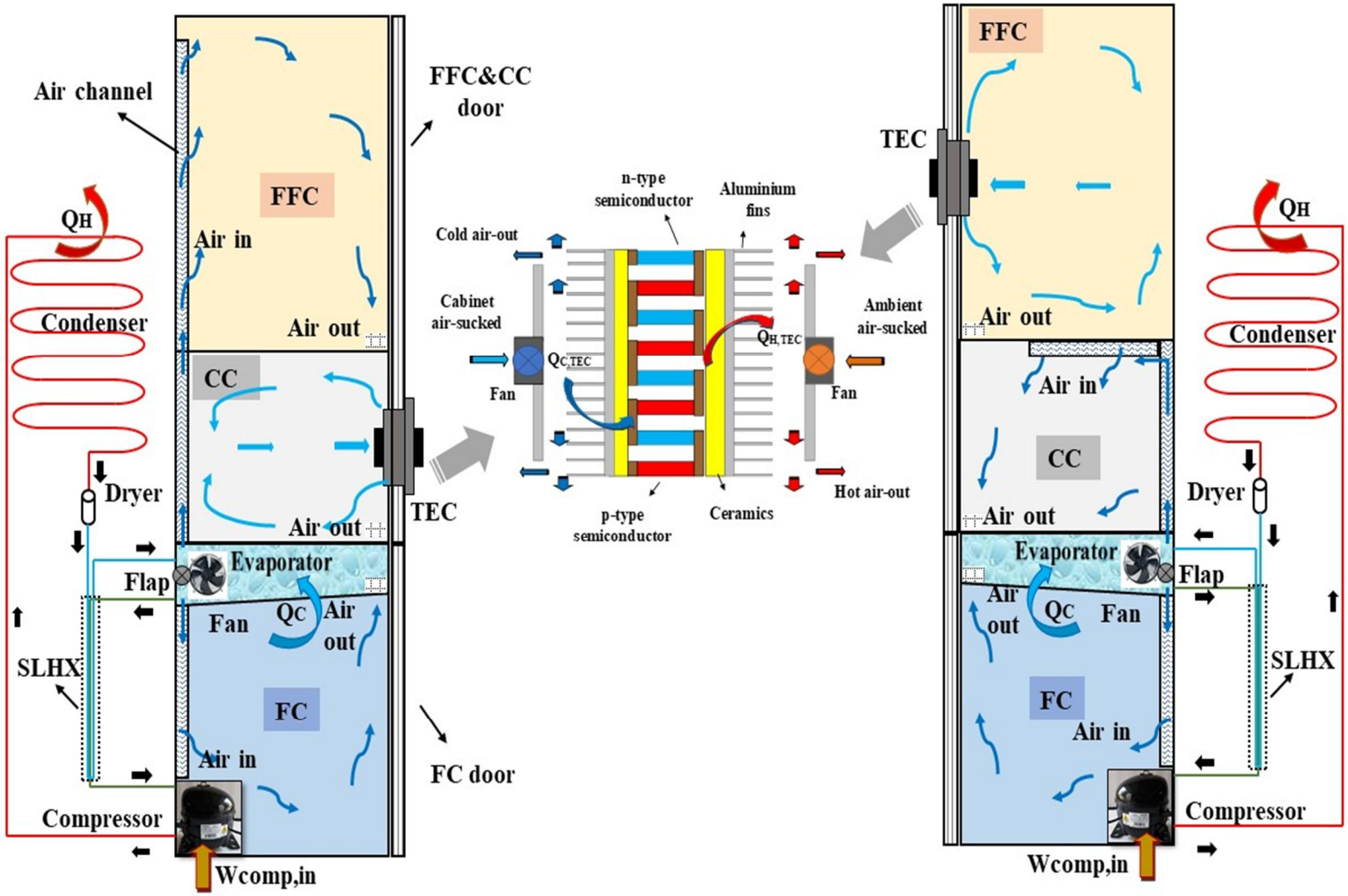


Figure 2.4. Schematic representation of the hybrid refrigerators (HHR-1 and HHR-2, left to right)



Figure 2.5. HHR-1 and HHR-2 (left to right)

* HHR-1: To build this refrigerator, Serial-1's body was used. The TEC was mounted into the door of Serial-1 (Figure 2.5) to chill CC.
* HHR-2: The TEC was applied to the Serial-2 to build this hybrid refrigerator. The purpose was to cool the FFC which makes this prototype different than the former one. The TEC was placed in the door of Serial-2 and the heat released into the environment (Figure 2.5).

In the HHRs, TECs were attached to the doors because,

* The intention was not to damage the main body of the original serial refrigerators in case of multiple tests. If the body is modified, hybrid refrigerator can no longer be converted to the serial refrigerator. But if the door is modified, it is enough to replace the modified doors with a door that hasn't been modified.
* In addition to the first clause, it was possible to use the refrigerators in the second phase of the work.
* There were skin condensers on the side walls and air channels in the rear of the serial refrigerators which prevented TECs being placed in these regions.

These two prototypes helped the author evaluate the performance and effect of TECs in many ways such as (a) application in different compartments which have different volumes and cooling capacity needs and (b) its behavior against different ambient temperatures.

In the work of Vian & Astrain (2009a) [23] following the computational analysis, it was decided to establish the hybrid refrigerator in which TEC's heat was released into the FFC rather than the ambient to increase TEC's efficiency. This was done because the lower the temperature difference between the TEC's faces, the higher the COP of TEC. Yet, in this current work, the configuration chosen by them was not preferred, because:

* To investigate TEC performance under real ambient temperatures (household refrigerators usually operate in the room temperatures between 16 °C to 32 °C).
* Despite being a more appropriate configuration from the energy point of view, its cooling capacity performance would most likely be poor because warm foodstuffs added into the FFC will cool down slower due to the heat released by the TEC.

2.2.4. Description of the Experimental Procedures

2.2.4.1. Standards, Specifications and Tests

Several tests were carried out complying with three different standards/test specifications. They are summarised in Table 2.4.

Table 2.4. Standards, specifications and the tests

Tests	EN 62552: 2013	IEC 62552: 2015	IC 15068	IEC 60704-2-14
Energy consumption	✓	✓		
Cooling capacity		✓	✓	
Freezing capacity	✓	✓	✓	
Noise				✓

2.2.4.2. Test Procedures

In the first phase of the work, all the tests, indicated in Table 2.4, of two serial refrigerators were initially performed to collect valid reference data. After completing Serial-1's tests, the HHR-1 was built on the Serial-1's body. The same tests were also performed with the Serial-2. Following the tests, the HHR-2 was built on the same body of Serial-2. Building the hybrid refrigerators on the same body of the serial units was very workable for a reliable comparison of the test results. This enabled us to measure only the effect of the cooling technologies (VC and TE) on the performance of the refrigerators; because, it lessened the variables such as insulation quality of the walls and heat transfer through the gaskets which directly affect the cooling performance of the refrigerators. The test results of HHRs were compared to results of serial ones to gain idea about the effect of the TECs on the refrigerators' performance. During the tests, the temperature (inner surface temperatures of the compartments, air inlet and outlet temperatures) and velocity data were also collected from the measurements to CFD analysis. According to the test results, HHR-2 which had the best performance was selected for further evaluations.

In the second phase, the air flow and temperature distributions inside the target compartment, FFC, of the HHR-2 were predicted by means of CFD tool. A simulation model was developed and validated via comparing its data with the reference measurements. Then, to find the most suitable place for the TEC to improve the uniformity of air flow and temperature fields in the FFC, various configurations were simulated in which the TEC was in different locations. Afterwards, the samples of new HHRs, HHR-3 and HHR-4, were built based on the configurations which had more uniform air flow and temperature distributions. Only the energy consumption and cooling capacity tests were conducted in compliance with IEC62552:2015 [57] to verify numerical results.

2.2.4.3. Test Chamber

The tests were carried out in the climate chambers located at the R&D Cooling Center of BSH Bosch Household Appliances in Çerkezköy, Tekirdağ, Turkey. The humidity and temperature inside the chambers can be set as required. The humidity was kept between

35% and 50% during the tests. The temperature inside the chamber ranges between 10 ºC and 43 ºC.

It was possible to collect test data of temperature, pressure, mass flow and power at each test station inside the chambers. Test data were collected by means of data acquisition systems and then transmitted to the test software to analyse the results.

2.2.4.4. Measurement Instruments

During the tests, the power was measured with a basic accuracy of 0.025% by Zimmer LMG500 power meter. Daily energy consumptions were measured with an uncertainty of less than 2%. Relative humidity had an overall uncertainty of measurement of not greater than 5%. Calculated values of test duration, made up of the difference between time recorded at each sampling period, had an expanded uncertainty of 0.1%. T type calibrated thermocouples were used as temperature instruments with a measurement uncertainty of ±0.2 °C. The test data were collected by the datalogger, Agilent 34970A.

For cooling and freezing capacity tests, the packages which consist of oxyethylmethylcellulose, sodium chloride and water, described in the given standards [56, 57], were used. The tolerances for the dimensions of packages were ±2mm for small packages, ±3mm for larger ones. The tolerances of masses of the packages were ± 2%.

Regarding the noise level measurements, the tests were performed in a semi-anechoic chamber-class 1 (ISO 3741) [59] with the measurement sensitivity of 43.1 mV Pa^{-1} and the microphone uncertainty of 0.2 dB.

2.3. Numerical Analysis

The main goal of numerical analysis in the context of this work was to gain an idea about the air temperature and flow field inside the FFC in order to place the TEC at the most optimum position. Since the focus in the work was to improve the HHR, a simpler and less time-consuming way of simulation was considered to be satisfactory.

Reference test data were collected from HHR-2's FFC when the room temperature and temperature inside the FFC were around 25 °C and 4 °C, respectively. ANSYS R18.1 FLUENT was used in the numeric analyses. Five different configurations were analysed

to find the optimum location for the TEC. For CFD analysis, the following assumptions were made:

- The fluid flow is incompressible because of a low Mach (Ma) number, *Ma < 0.3*.
- The flow is assumed as steady state.
- The analysed compartment, FFC, is unloaded.
- Air leakages through the gaskets are neglected.
- Boussinesq model is considered.
- Radiation effects are neglected.
- Flow inside the FFC is turbulent
- Forced convection occurs inside the FFC.

2.3.1. Mathematical Model / Governing Equations

The model was developed considering the above assumptions; it consists of three key governing equations.

2.3.1.1. Conservation of Mass (Continuity Equation)

$$\nabla \cdot \vec{V} = 0 \tag{2.6}$$

where $\vec{V}$ is fluid velocity and ∇ **is the gradient** operator.

2.3.1.2. Conservation of Momentum (Navier-Stokes Equations)

For incompressible flow of Newtonian fluids, the conservation of linear momentum is governed by the Navier-Stokes equation.

$$\rho \frac{D\vec{V}}{Dt} = \rho\vec{g} - \nabla\rho + \mu\nabla^2\vec{V} \tag{2.7}$$

Here, ρ is the fluid density, $\vec{g}$ is gravitational acceleration, p is the static pressure, μ is the dynamic viscosity coefficient. In Equation 2.7, D/Dt represents the material (or substantial) derivative, and ∇^2 is the Laplacian operator.

2.3.1.3. Conservation of Energy

The first law of Thermodynamics written for the incompressible air in the FFC is given in Equation 2.8.

$$\rho c_p \frac{DT}{Dt} = k\nabla^2 T + \Phi \tag{2.8}$$

where, c_p is the heat capacity under constant pressure, T is the fluid temperature, k is the thermal conductivity of the fluid and Φ is the dissipation function.

2.3.2. Geometric Model

Because the real geometry of the FFC has many complexities, the geometric model was simplified to provide the most optimum mesh size. In this way, the solution time was significantly decreased. The geometry was drawn by means of the Solidworks software and then transferred to the ANSYS DesignModeler to prepare for meshing. The simplified geometries of the FFC and the TEC are demonstrated in Figure 2.6, the dimensions are given in Table 2.5.

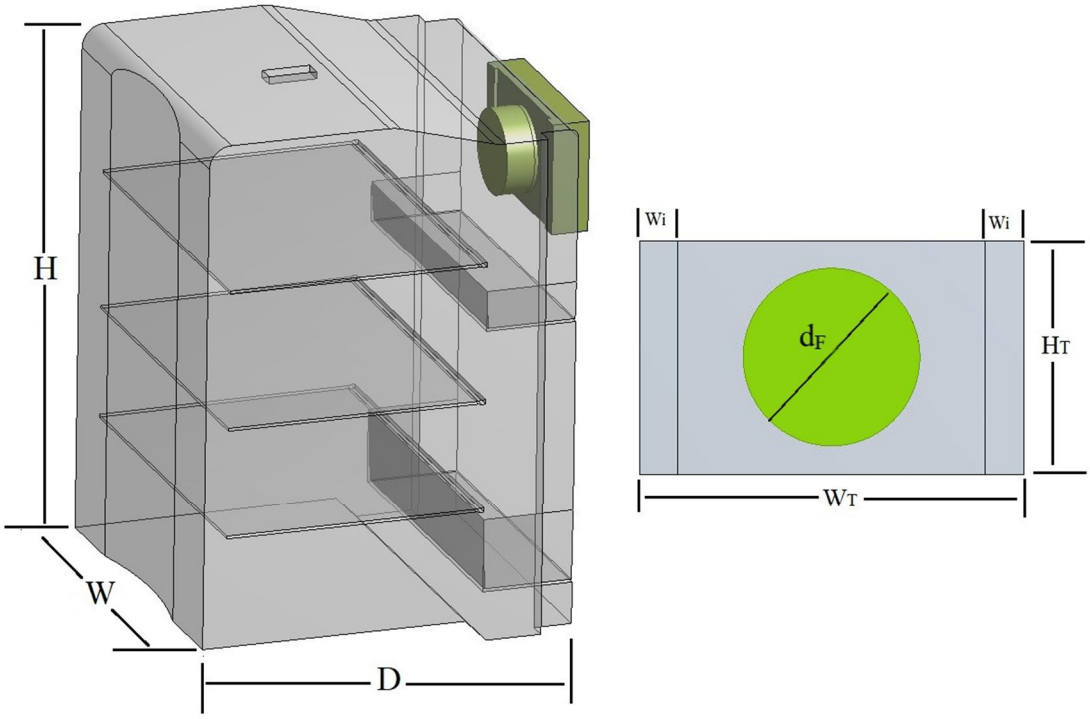


Figure 2.6. Geometric model of the FFC and TEC

Table 2.5. FFC and TEC dimensions

Dimensions		Values
FFC	H	695 mm
	W	495 mm
	D	515 mm
TEC cold face	H_T	150 mm
	W_T	250 mm
	Wi	30 mm
	d_F	120 mm

2.3.2.1. Meshing the Geometry

The geometry of FFC was meshed using ANSYS AUTODYN. The details of the mesh for HHR-2's FFC geometry (including TEC) are given in Table 2.6. An unstructured mesh composed of tetrahedron elements was constructed. The resulting surface mesh is displayed in Figure 2.7.

Table 2.6. Details for the meshing

Parameters		Values
Sizing	Size function	Curvature
	Relevance centre	Fine
	Transition	Slow
	Span angle centre	Fine
	Curvature normal angle	18^0
	Min size [m]	0.00009
	Max face size [m]	0.0047585
	Max tet size [m]	0.5
Quality	Smoothing	Medium
Statistics	Nodes	673,347
	Elements	3,501,386

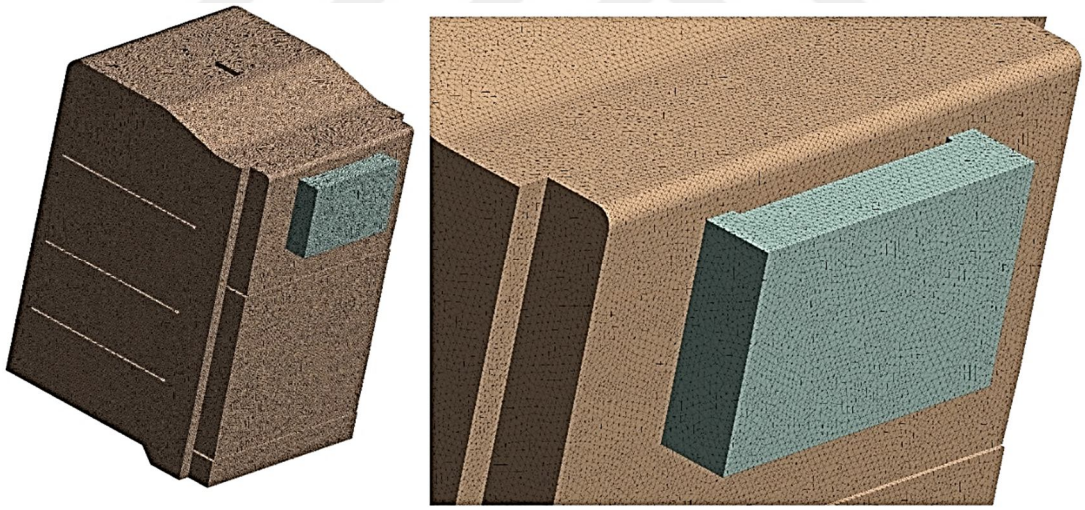


Figure 2.7. An unstructured mesh composed of tetrahedron elements

2.3.2.2. Convergence of Solution and Mesh Independent Study

In order to perform a mesh independence study, variations of inner wall surface temperatures at 14 different locations (see Figure 2.8) of the FFC with respect to the mesh size were monitored.

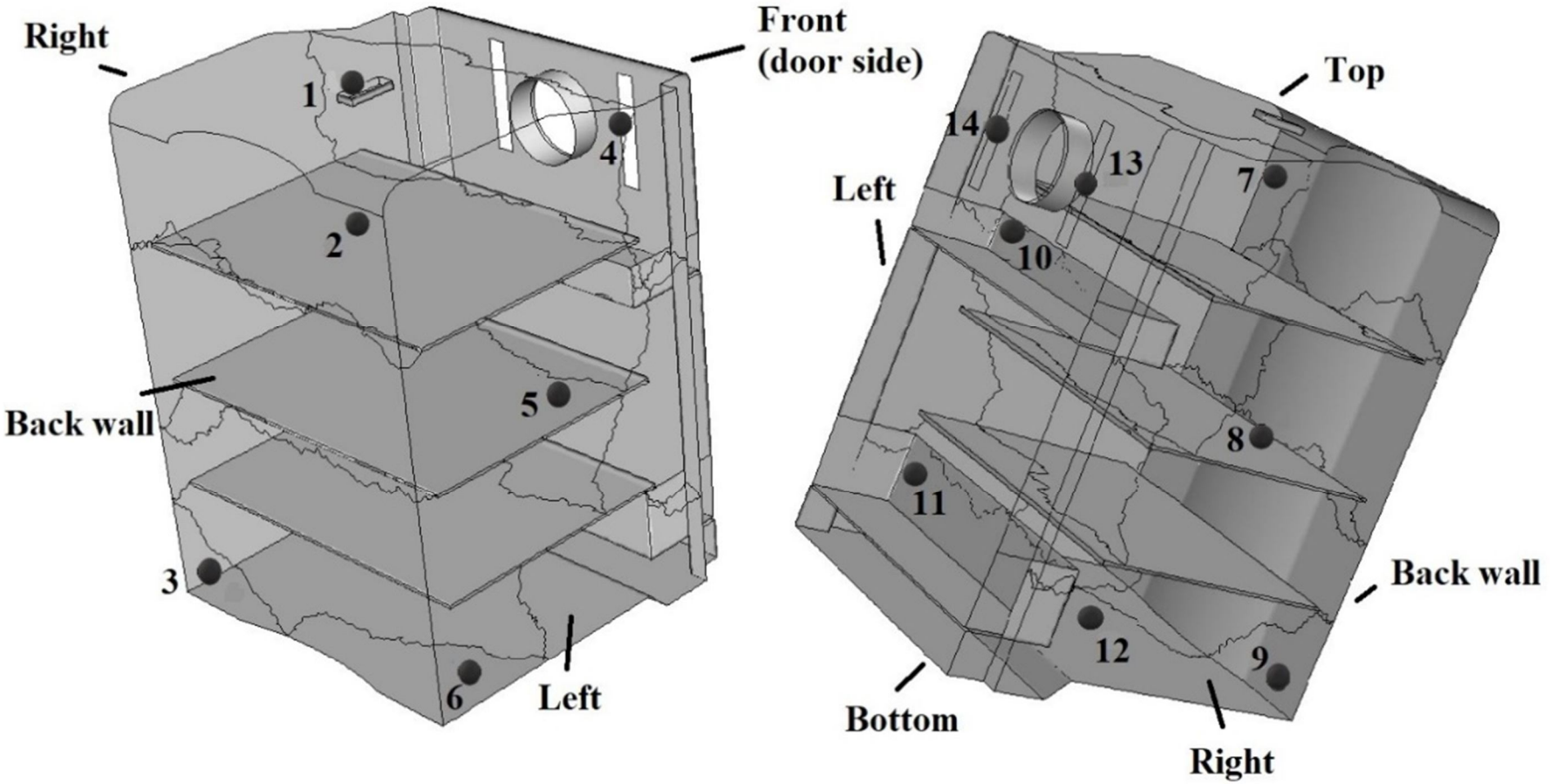


Figure 2.8. The location of surface temperatures

For this purpose, initial solutions were obtained for a mesh size of 500,000 cells, then the mesh size was gradually increased to 5,000,000 cells. Here, only the minimum cell size and the maximum face size were changed, while other mesh parameters were kept constant. For each mesh size, the residual for continuity, momentum and energy equations were monitored as well as inner wall surface temperatures. The variations of residuals and wall temperatures for the mesh size of 3,500,000 elements with respect to the iterations are demonstrated in Figure 2.9 and Figure 2.10, respectively. As seen in Figure 2.10, after the iteration number of 1000, the temperatures reach the stability. Thus, the maximum number of iterations was selected as 7,500 in each solution to consume less time. According to Figure 2.11, the temperature predictions did not exactly become mesh independent. However, considering the scale of the vertical axis of Figure 2.11 and noting the fact that the main purpose of the CFD predictions here is to rank different configurations in terms of velocity and temperature distributions, the average temperature variation after the mesh size of 3,500,000 is assumed to be small enough. Therefore, mesh size of 3,500,000 is used in the rest of the study; analysis by Belman-Flores et al. [38] determined that a similar mesh size could be applied to an FFC.

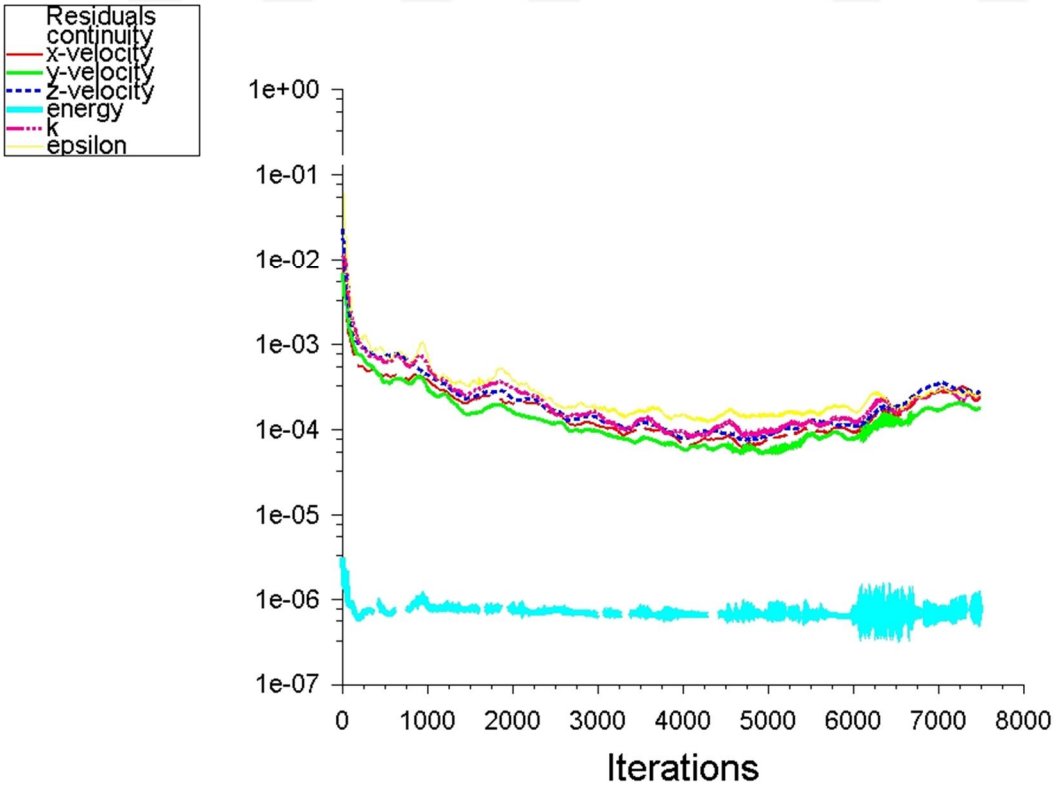


Figure 2.9. Changes of residuals with iteration

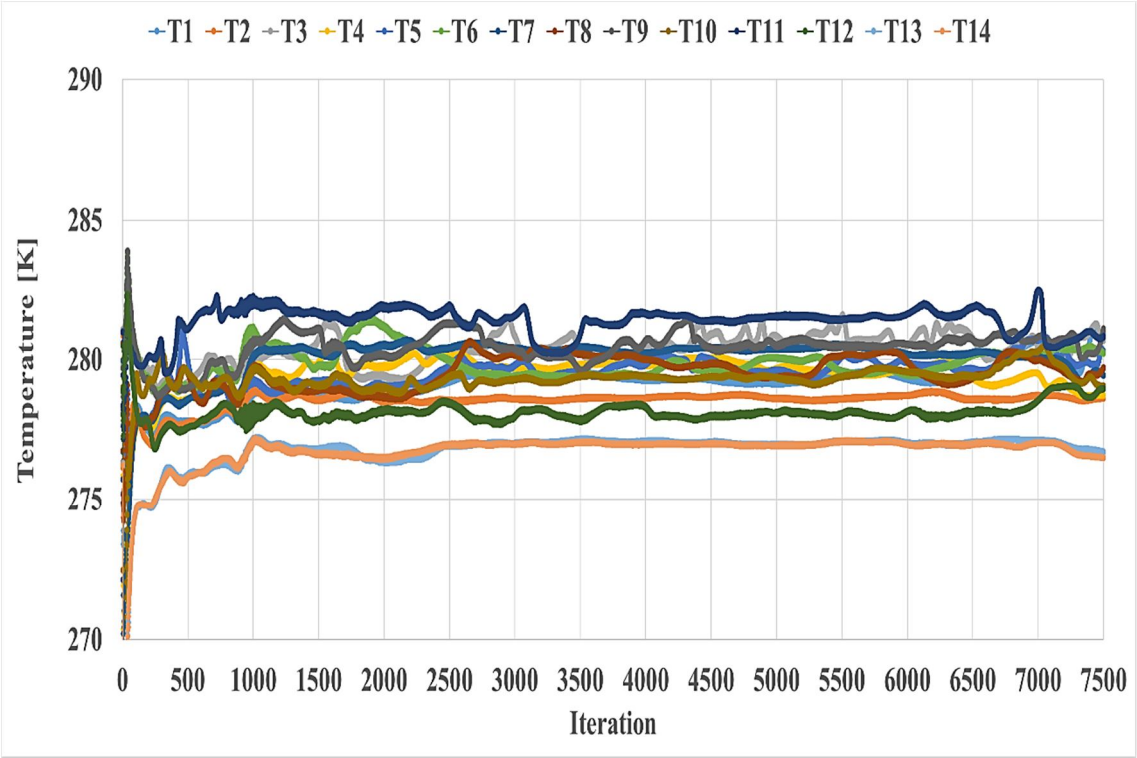


Figure 2.10. Changes of the inner wall surface temperatures with iteration for the mesh size of 3.5 million elements

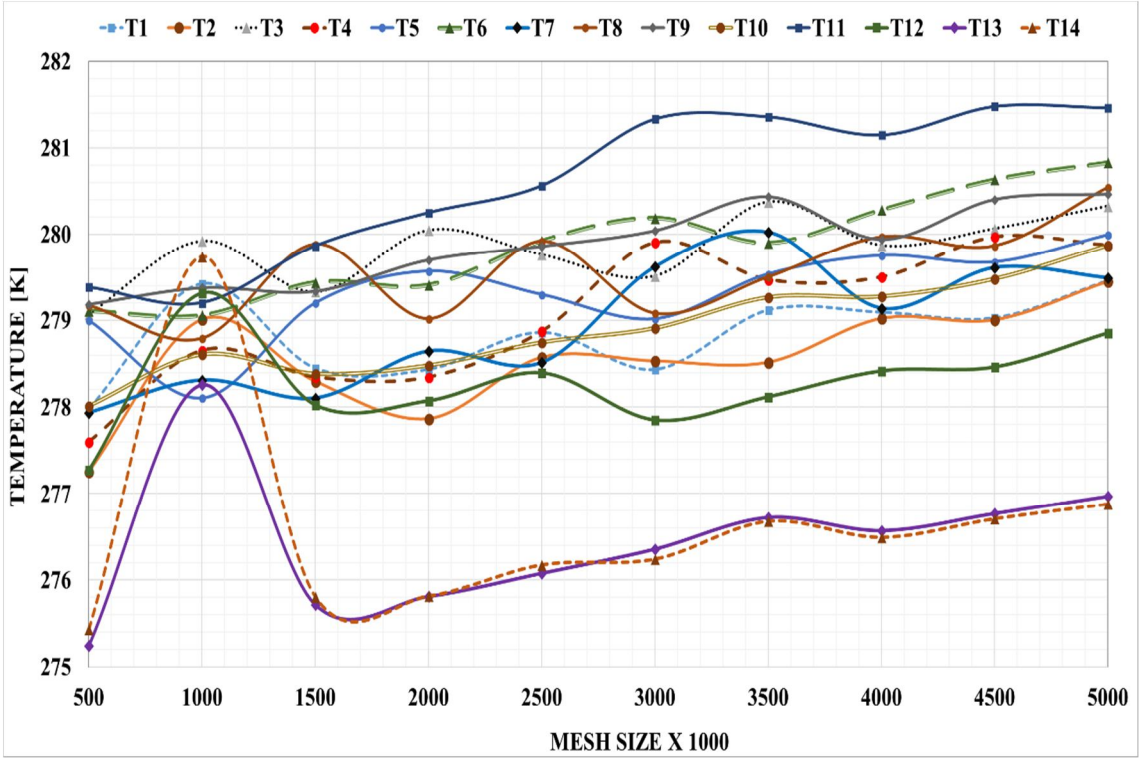


Figure 2.11. Determination of mesh size

A viscous model of realizable k-epsilon model with a standard wall function was applied. As options, full buoyancy effects and curvature correction were considered. Regarding the solution methods, SIMPLE algorithm, with second order upwind discretization for pressure, momentum, and turbulent kinetic energy and first order upwind discretization for turbulent dissipation rate were used.

2.3.4. Properties of Materials

The properties of the materials are summarised in Table 2.7.

Table 2.7. Material properties

Materials		**ρ**	**C_p**	**k**
		[kg/m^3]	***[J/kgK]***	***[W/mK]***
Fluid	*Air*	1.225	1,006.43	0.0242
Solid	*Aluminium*	2719	871	202.4
	Expandable polystyrene	26.25	1,200	0.0345
	Polyurethane	34.5	1,500	0.022
	Vacuum insulated panel	225	1,050	0.005

2.3.5. Boundary Conditions

The boundary conditions of FFC are shown in Table 2.8. The real average ambient temperature was 25.05 °C (298.20 K). Yet, around the hot face of TEC and on the side walls of FFC (due to the skin condensers) the ambient temperature was relatively higher.

Table 2.8. Boundary conditions for the FFC

Boundary		**Thermal conditions**	**Ta**	**h**	**v**
			[K]	***[W/m^{2}K]***	***[m/s]***
Air inlet	*Inlet 1*	-	274.35	-	2.10
	Inlet 2	-	274.55	-	2.45
Air outlet Walls	*Top wall*	Convective	298.20	1.80	No slip
	Side walls	Convective	305.65	2.91	No slip
	Front wall	Convective	298.20	1.64	No slip
	Back wall	Convective	298.20	2.15	No slip
	Bottom wall	Convective	275.65	2.17	No slip
TEC	*TEC wall*	Convective	300.05	16.37	No slip

3. RESULTS AND DISCUSSION

3.1. Results of the First Phase of the Work

In this phase, only experimental investigation was conducted. As highlighted previously, two serial refrigerators, as reference, and two HHRs were tested.

3.1.1. Energy Consumption

EN 62552:2013 and new IEC 62552:2015 standards were considered as reference norms for energy consumption tests because both norms have different measurement methods. Moreover, it was useful to know the energy consumption performance of the HHRs under different test conditions. While EN 62552:2013 is still in use in the Europe and Turkey, the forthcoming IEC 62552:2015 will be valid after 2020. A new regulation to define energy efficiency classes for household refrigerators were also published in accordance with the new standard. The adaptation efforts of current household refrigerators to the new regulation have been in progress since 2015. Yet, due to ambiguities in the standard, a new version will be published at the end of 2018 as IEC62552:2018.

The target compartment temperatures and measurement methods are presented in Table 3.1. The critical differences between them are the ambient and the FFC target temperatures as well as the loading conditions of the freezer. The test packages are not used in IEC 62552:2015.

Table 3.1. Compartment target temperatures and loading conditions for energy consumption tests

Standard	Ambient	FFC		CC		FC	
	(°C)	*Target temp.(°C)*	*Measured by*	*Target temp.(°C)*	*Measured by*	*Target temp.(°C)*	*Measured by*
EN 62552:2013	25.0 ±0.5	≤ +5.0	Cu cylinder	-3.0 ≤Tcc≤ +3.0	Packages	≤ -18.0	Packages
IEC 62552:2015	16.0 ±0.5 and 32.0 ±0.5	≤ +4.0	Cu cylinder	≤ +2.0	Cu cylinder	≤ -18.0	Cu cylinder

3.1.1.1. Energy Consumption Test Results According to EN62552:2013

The energy consumption test results are given in Figure 3.1. Here, Serial-2 has lower energy consumption than Serial-1 although they have the same configurations. Serial-2's consumption is 2.7% less than that of Serial-1. However, those differences in the results are reasonable due to changes in the efficiency of the components, particularly the compressor COP of the cooling systems and insulation quality, etc.

It is also clear from the Figure 3.1 that hybrid refrigerators have higher energy consumption levels than the serial ones. HHR-1's energy consumption result at ambient temperatures of 25°C is four times higher than those of Serial-1. The same differences can be seen between the energy consumption of Serial-2 and HHR-2.

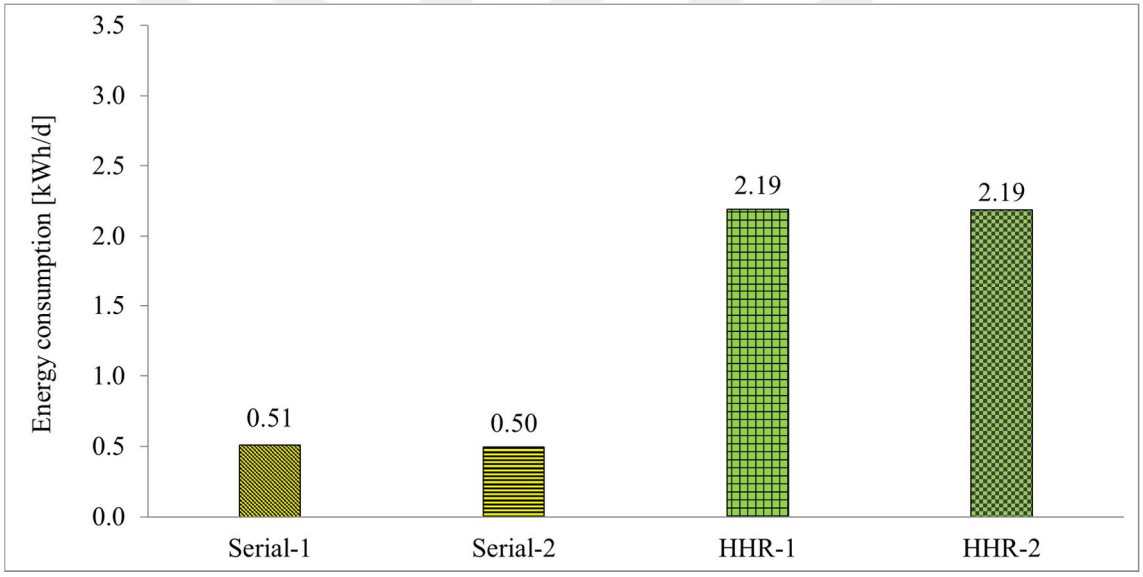


Figure 3.1. Energy consumption test results according to EN62552:2013

3.1.1.2. Energy Consumption Test Results According to IEC62552:2015

The results are like those conducted according to EN 62552:2013. Figure 3.2 reveals that the energy consumption level of Serial-2 is lower than Serial-1; it has 5.4% and 7.5% lower energy consumption levels at 16 °C and 32 °C ambient temperatures, respectively.

The HHRs have higher energy consumption compared to the serial ones. HHR-1's energy consumption at an ambient temperature of 32 °C is four times higher than Serial-1. On the other hand, at an ambient temperature of 16 °C, its consumption is three times

higher than Serial-1's consumption. This is because, at low ambient temperatures, thermoelectric coolers operate more efficiently. Similar differences can be seen between the energy consumption of Serial-2 and HHR-2.

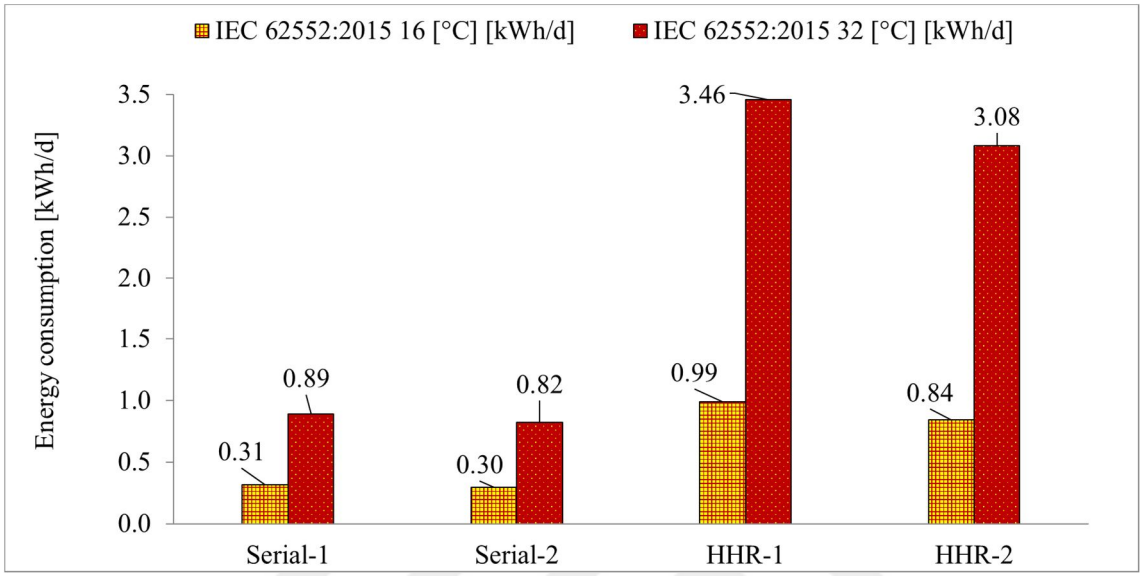


Figure 3.2. Energy consumption test results according to IEC62552:2015

3.1.1.3. Root Causes for the Higher Energy Consumptions of HHRs

The root causes for the higher energy consumption of hybrid refrigerators are discussed considering the critical components of energy consumption: COP, power consumption, run time of the components and insulation.

Lower COP values of TECs:

The TEC system COPs used in HHR-1 and HHR-2 are up to 0.37 W/W and 0.30 W/W respectively. Those values are considerably lower compared to the vapour compression system COPs used in the serial units which are close to 1.8 W/W.

Higher power consumption of TECs:

In Figure 3.3, the power consumed for each compartment at the ambient temperature of 25°C [56] are presented. HHR-2's FFC has the highest power consumption, just above 170 W; followed by HHR-1's CC, 136 W. Those compartments were cooled by TECs, which need high power to provide target temperatures.

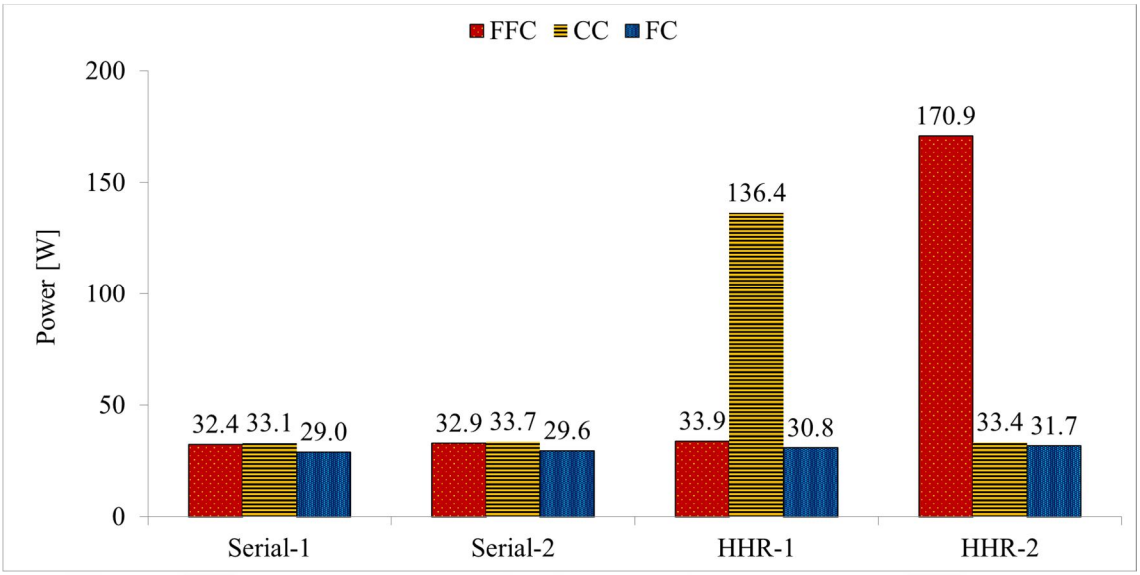
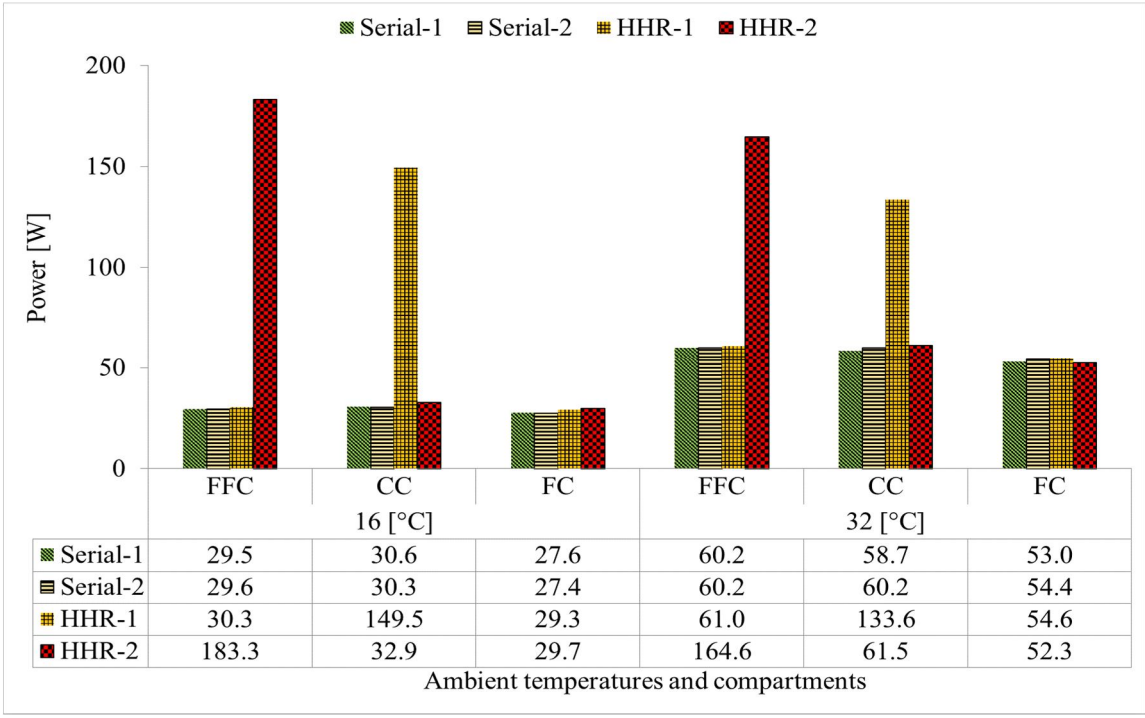


Figure 3.3. Electrical power consumption (EN62552:2013)

The power consumptions at the ambient temperatures of 16 °C and 32 °C [57] have similar behaviors with the ones at 25 °C as seen in Figure 3.4. At 16 °C, HHR-2's FFC has higher power consumption of 183.3 W compared to rest of the compartments; followed by HHR-1's CC, 149.5 W. The power consumptions for HHR-2's FFC and HHR-1's CC at 32 °C are 164.6 W and 133.6 W, respectively.



	16 [°C]			32 [°C]		
	FFC	CC	FC	FFC	CC	FC
Serial-1	29.5	30.6	27.6	60.2	58.7	53.0
Serial-2	29.6	30.3	27.4	60.2	60.2	54.4
HHR-1	30.3	149.5	29.3	61.0	133.6	54.6
HHR-2	183.3	32.9	29.7	164.6	61.5	52.3

Ambient temperatures and compartments

Figure 3.4. Electrical power consumption (EN62552:2015)

Run time ratio (RTR):

Another crucial factor determining energy consumption level is run time of the compressor and TEC. It is the ratio of their operation time to the total analysing time. The longer these components run, the more energy that is consumed in the on/off operation. Because low COP of the TECs, the RTR values of them have much more effect on the high energy consumption compared to compressor run. In light of this, it is more useful to focus on the TEC's run.

As noted before, HHR-1's CC and HHR-2's FFC were cooled by the TECs and it is clear from Figure 3.5 that they need more time to provide required temperatures in the target compartments compared to the compressors of serial refrigerators (The RTR of compressors for FCs are high but they will be out of discussion due to their minor effect on energy consumption). Furthermore, although HHR-1 has lower power consumption than HHR-2 as presented in Figures 3.3 and 3.4, its daily energy consumption is still higher than that of HHR-2. This is due to the higher RTR of the TEC for HHR-1's CC. Its RTR is 25.6% and 3.5 times higher than the RTR of HHR-2's CC. Even though HHR-2's FFC has the second highest RTR with a value of 20%, which is 61% higher compared to HHR-1's FFC, HHR-2 still has the lower energy consumption. One of the reasons for that the differences in the target compartment temperatures: for CC, it is below 3 °C (the temperature of the warmest package (EN-62552:2013, 2013)) and for FFC, it is 5 °C, see Table 3.1. The other reason is that the drawers inside the CC causes poor air circulation in the cabinet.

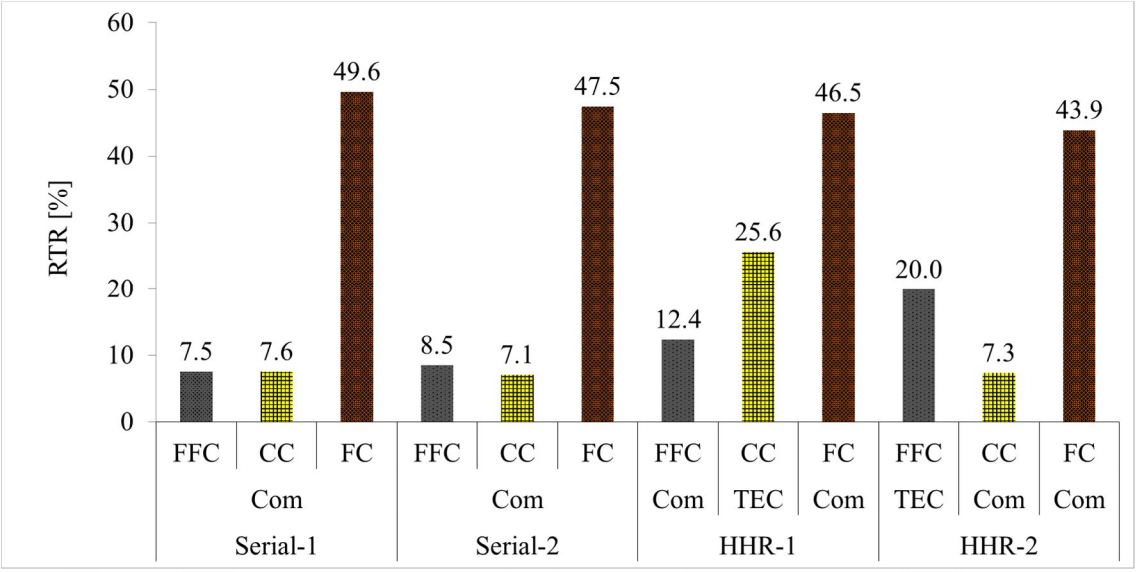


Figure 3.5. Run time ratio of the refrigerators (EN62552:2013)

Effect of insulation, UA:

Insulation of the compartments is the most critical parameter which affects energy consumption. Since there is a heat flow from the ambient to the inside of the compartments through the TECs, the insulation quality of the hybrid refrigerators is worse than the serial ones. According to the UA test results highlighted in Section 2.2.2.1, mounting the TEC on the serial refrigerators increases heat transfer by almost 10%. The TECs applied to the hybrid units have an on/off cycle, operated according to the compartment temperature sensor. Even when the TEC is off, there is a great deal of heat flow through them.

3.1.2. Temperature control with TECs

It is extremely important to control the interior temperature of the compartment where highly perishable foodstuffs are held. Examples of these foods are meat, poultry, fish, milk, eggs and cooked meals. The chill compartment is designed specifically for preserving perishable foods.

In Figure 3.6, fluctuations in the average chill temperatures of Serial-1 and HHR-1 are displayed (as mentioned in the previous sections, the CC of HHR-1 is cooled by TEC). The temperature data was collected from the energy consumption tests performed at an

ambient temperature of 32±0.5 °C. The temperature control settings were set to values to provide an average chill compartment temperature of just below 2.0 °C. The CC temperatures were measured from the locations emphasized in IEC 62552: 2015 (*no packages were used*). The effect of TEC hysteresis on the HHR-1's CC temperature behavior was studied.

According to Figure 3.6, it is possible to control interior temperature of the chill compartment with high accuracy of ±0.1 °C if the TEC hysteresis is set to 0.1 °C. With the increasing hysteresis, the temperature stability inside the chill compartment worsens. Although it is more serviceable to decrease hysteresis for a sensitive temperature control, decreasing hysteresis leads to high energy consumptions. In the tests, the TECs' hysteresis was adjusted to 0.5 °C which was the optimum value in terms of temperature control and energy consumption. By increasing hysteresis from 0.1 °C to 0.5 °C, energy savings of approximately 5% were achieved. It should also be noted that at all hysteresis values of TEC, the cut-in and cut-off temperatures were kept at certain ranges which provides steadier temperature conditions inside the compartment.

Besides, the control on the Serial-1's CC temperatures is at acceptable level with an accuracy of ±0.4 °C. However, it is difficult to control more sensitively due to the on/off run of the compressor and dependent on the operation between the compartments.

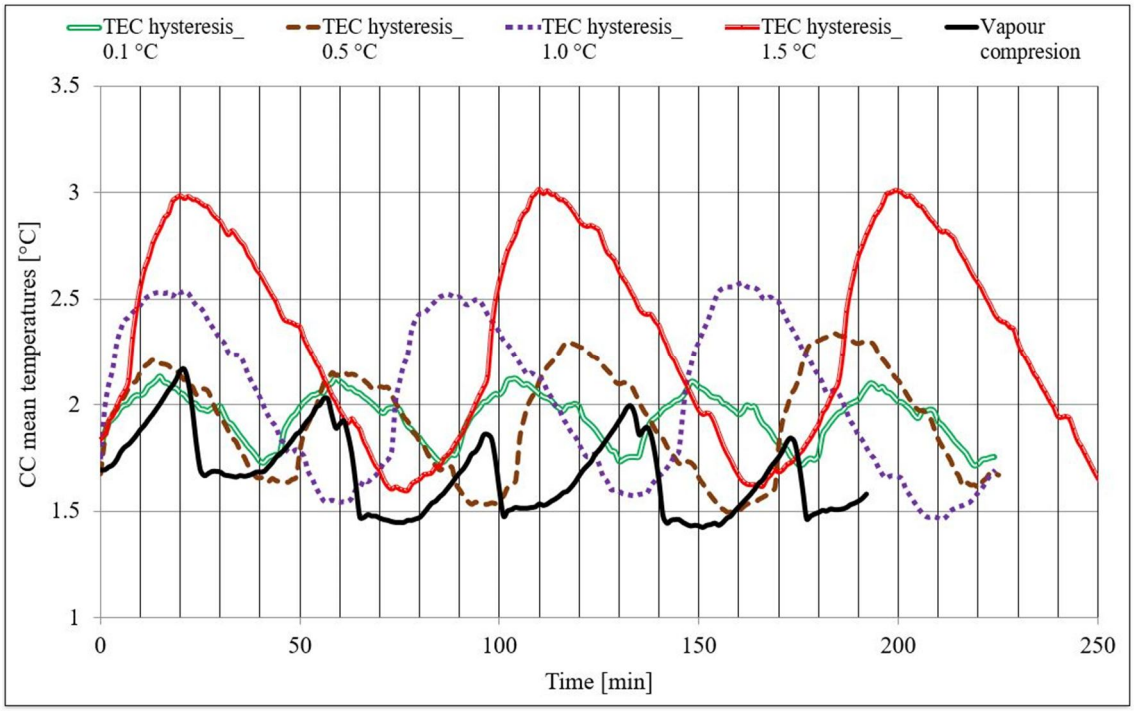


Figure 3.6. Fluctuations in the average CC temperatures of Serial-1 and HHR-2

3.1.3. Cooling capacity

The cooling capacity test is the most critical test to measure cooling capability of the compartments which have a target temperature of above 0 °C. The fundamental purpose is to determine the time for a specified amount of load to be cooled from 25 °C to 10 °C (IEC-62552:2015, 2015).

The test was performed in regard to the IEC 62552:2015 (IEC-62552:2015, 2015) norm and IC 15068 (ICRT, 2015) test specification. The EN 62552:2013 (EN-62552:2013, 2013) standard does not include this test. Table 3.2 demonstrates the compartment target temperatures and loading conditions for cooling capacity tests.

In IEC 62552-2015 (IEC-62552:2015, 2015), the tested refrigerator is prepared for the cooling capacity test as it is done for the energy consumption test. The refrigerator is empty at first, that is, there are no cold packages inside the compartments (the temperature measurements are again made by Cu cylinders). It shall provide the target compartment temperatures and be in stable conditions before insertion of warm test packages. Before loading, the warm packages are kept at 25 °C ±0.5 °C and then they are

inserted into the FFC. If a super cooling function exists for the FFC, this should be activated immediately after loading.

Table 3.2. Compartment target temperatures and loading conditions for cooling capacity tests

Standard	Ambient	FFC		CC		FC	
	(°C)	*Target temp.(°C)*	*Measured by*	*Target temp.(°C)*	*Measured by*	*Target temp.(°C)*	*Measured by*
IEC 62552:2015	25.0±0.5	≤ +4.0	Cu cylinder	-3.0 ≤Tcc≤ +3.0	Cu cylinder	≤ -18.0	Cu cylinder
IC 15068	25.0±0.5	≤ +4.0	Packages	≤ +2.0	Packages	≤ -18.0	Packages

The same procedure also applies for the IC 15068 (ICRT, 2015); however, all compartments should initially include cold packages. Before inserting the warm packages, the average temperature of the cold packages must be at or below the target compartment temperature. Additionally, the warm packages are loaded not only into the FFC but also CC. Finally, the test is performed twice with and without activation of super cooling function.

The authors preferred to use the naming used in the norm of IEC 62552:2015 for "cooling capacity". Normally, after the cooling time is determined, the capacity and the amount of load to be cooled, is calculated in kg. However, in this investigation, the results were analyzed in time required for cooling in minutes.

The amounts of warm and cold packages loaded into the compartments are shown in Table 3.3. As this table indicates, IC 15068's loading amounts and conditions are more challenging compared to those of IEC 62552:2015's.

Table 3.3. The amount of test packages for cooling capacity test

Standards	**IEC 62552:2015**			**IC 15068**		
Packages	*FFC*	*CC*	*FC*	*FFC*	*CC*	*FC*
Warm [kg]	7.5	-	-	17.5	8.0	-
Cold [kg]	-	-	-	10.5	5.0	21.0

Before the analyses, it must be noted that there are some critical factors which directly affect the cooling time test results. Those are: (1) the temperature of the warm packages and compartments at the moment of insertion, (2) loading time, (3) air humidity of the test chamber and (4) measurement errors. In each test, the temperature variation between the warm packages was kept at ±0.1 °C. The differences in average temperature of the warm packages in the tests are maintained at ±0.5 °C. Additionally, the allocated time for the loading was less than 90 s. All those factors affect the accuracy of the results.

3.1.3.1. Cooling Capacity Test Results According to IEC 62552:2015

The results are presented in Figure 3.7. The average cooling time of the serial refrigerators is approximately 250 min and the difference between them is roughly 10%. The difference is at an acceptable level; thus, the results of the serials can be considered as a reference. The HHR-1 refrigerator has very similar results to the serial ones. This is expected because HHR-1's FFC is cooled by a compressor-based system as in the serial ones. HHR-2 has the lowest cooling down time with a value of 146.6 min (2.44 h). This is a considerable achievement because the warm packages were cooled down in approximately 40% less time compared to the others.

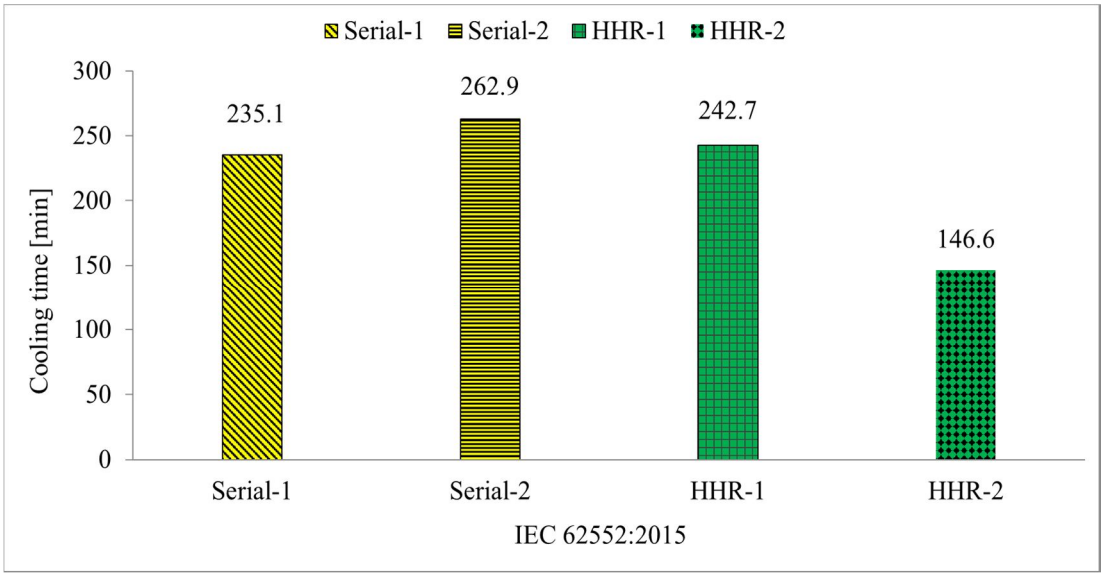


Figure 3.7. Cooling capacity test results according to IEC 62552:2015

3.1.3.2. Cooling Capacity Test Results According to IC 15068

Figure 3.8 summarises the test results in compliance with IC 15068 for cases of super cooling activation (Yes) and without super activation (No). Serial refrigerators' cooling time measurements have a difference of 9.0% and 9.8%, with and without super activations, respectively. Those reference results are also relevant. Additionally, in the case of super activation, the cooling time is around 6% lower than the no super activation case. One could expect much lower cooling time values when the super cooling function is activated. Yet, there is a certain restriction about the average internal temperatures of the FFC which shall be kept between 0 and 8 °C. The temperature is not allowed to drop below 0 °C. This restriction must be carefully considered in the design of the super cooling process otherwise icing may occur in some regions of the FFC.

HHR-1's performance is poor, cooling time levels are 10% to 17% higher than those of serial ones and 70% higher than the HHR-2's. HHR-1 is in fact supposed to have less cooling time, since the CC is cooled by the TEC. The primary reason for this is the location of TEC, positioned to the front side of the CC. The CC has drawers that prevent cooled air to flow over the warm packages. As mentioned earlier, the CC of other refrigerators are cooled by a compressor-based system and air is blown into it from the top and directly flows over the packages.

HHR-2 has the best cooling performance; its cooling time is at least 32% lower than those of the serial ones. This is also a notable success. The leading factors are (1) as explained in the previous paragraph, direct air flow over the warm packages in the CC and (2) using separate cooling technology, TEC, for FFC. Since there is no drawer in the FFC, the cooled air passes through the warm packages accelerating the cooling process. It must be highlighted that 17.5 kg of warm packages were cooled in HHR-2's FFC by TEC in less than 250 min.

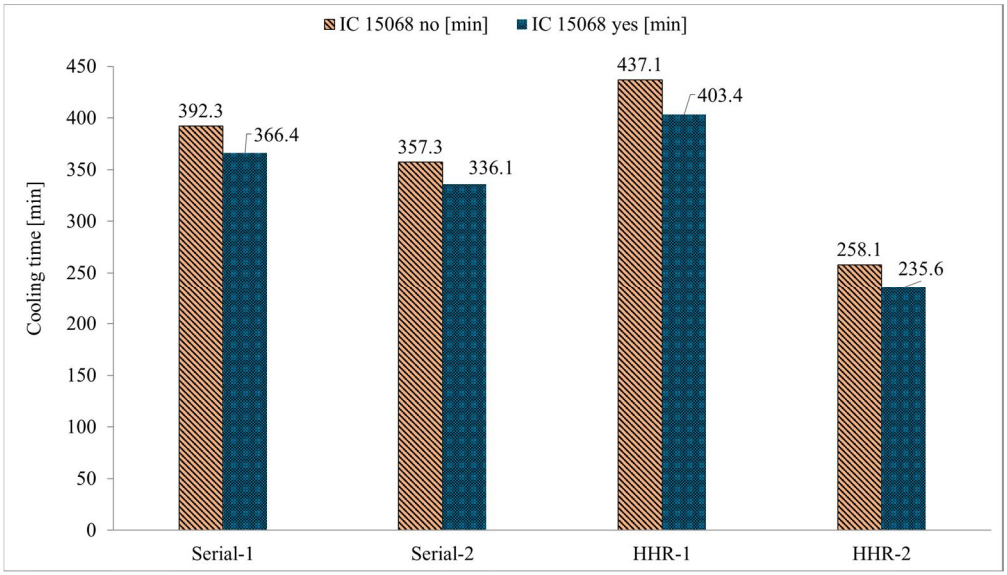


Figure 3.8. Cooling capacity test results according to IC 15068

3.1.3.3. Rapid Cooling with TEC

The promising results of HHR-2 ought to be investigated in detail because it is possible to design a compartment that has a rapid cooling feature.

Rapid cooling, also called blast chilling, is strongly advised for protecting the nutritional values and freshness of the foodstuffs. This feature is therefore increasingly paid attention by the domestic refrigerator users. There is no common definition of rapid cooling in the norms but a small number of food agencies which describe it and define its criteria (Dixon & Hall-Campell, 2016; FSA, 2016). In the report of (Dixon & Hall-Campell, 2016), it is pointed out that hot food must be cooled from 57.2 °C to 21.1 °C within 2 h and then from 21.1 °C to 5 °C within 4 h (in total 6 h). The cooling speeds of the first and second steps are *0.30* °C/min and *0.07* °C/min, respectively. In other words, the average cooling speed from 52.2 °C to 5 °C is approximately *0.15* °C/min. According to The Food Standards Agency of the UK (FSA, 2016), hot food must be cooled in 2h from 63 °C to 8 °C (cooling speed is nearly *0.46* °C/min). In the "Defence Catering Manual" (MoD-UK, 2014), it is indicated that cooked food must be chilled from 63 °C to 5 °C within 90 min with a speed of around *0.65* °C/min. In those requirements, there is no indication about the capacity of the foodstuff to be cooled. If a refrigerator manufacturer claims a compartment with a rapid cooling feature, it shall declare a

maximum loading capacity to be inserted into it. For a rapid cooling, the cooling speed must certainly be above *0.07* °C/min.

Because only the FFC is loaded, the results according to IEC 62552:2015 are to be discussed in terms of rapid cooling. This enables us to figure out the effect of TEC on the cooling performance.

Figure 3.9 demonstrates the temperature curves which show the temperature change of the warm packages with time for each refrigerator. The loaded amount is 7.5 kg. HHR-2 has a cooling speed of 0.10 °C/min while the others have speeds of 0.06 °C/min. With respect to Dixon & Hall-Campell (2016) HHR-2's FFC can be considered as having rapid cooling feature.

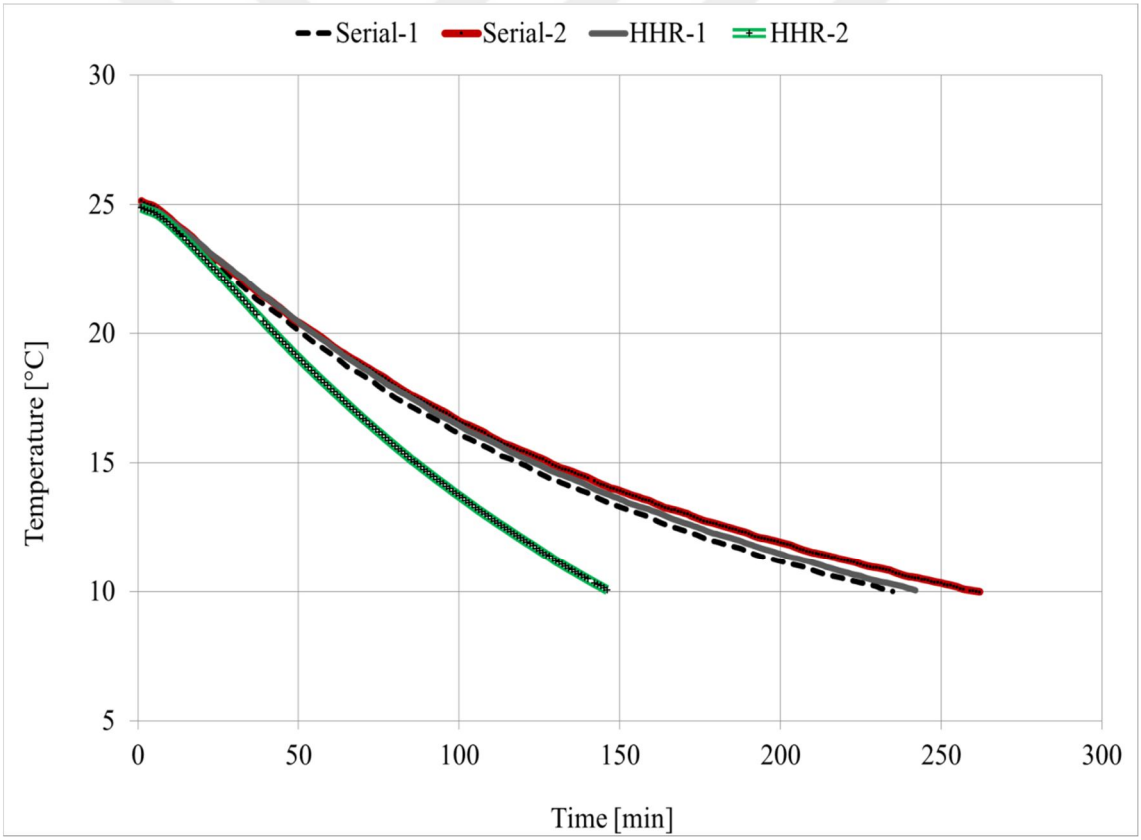


Figure 3.9. Cooling down time of the refrigerators according to IEC 62552:2015

Figure 3.10 presents the results of the tests conducted with Serial-1 and HHR-2 with different number of warm packages (Because Serial-1 had better results than Serial-2, it is compared with HHR-2). If the amount is decreased to 2.5 kg, the HHR-2's cooling

time drops significantly to 104 min with a speed of 0.14 °C/min. Its reaction to a smaller number of packages was more positive than the Serial-1's.

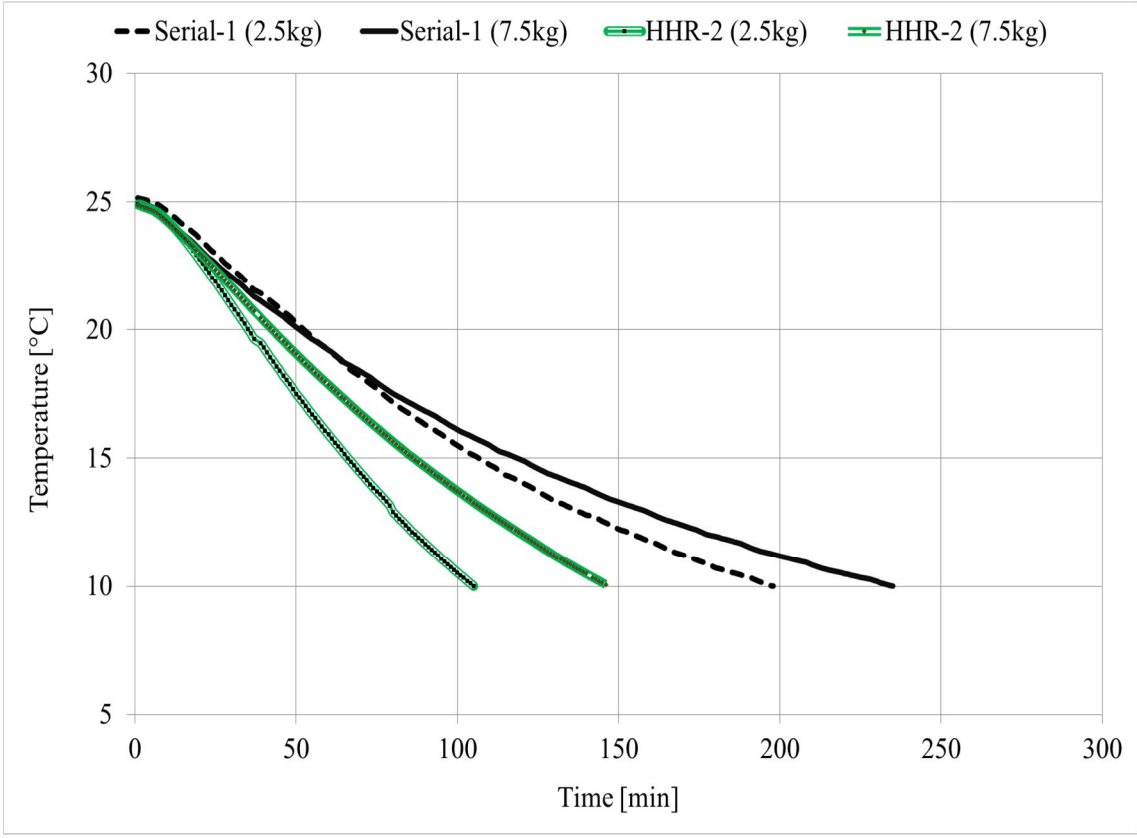


Figure 3.10. Cooling down time of Serial-1 and HHR-2 with different amount of warm packages

The results are very promising. If a special design is made for this feature inside the FFC (or in any unfrozen compartment) it is possible to cool a significant amount of foodstuff in minutes.

3.1.4. Freezing capacity

The objective of the test is to measure the freezing capacity of the food freezers and freezer compartments (EN-62552:2013, 2013; IEC-62552:2015, 2015).

In terms of method, this test is like a cooling capacity test. The freezer compartment is first loaded with cold packages, but an empty space is reserved for warm packages. The warm packages are kept at 25 °C ±0.5 °C. Once stable conditions are met, if there is no

super freezing function, the warm packages are inserted into the freezer. Afterwards, the warm packages are frozen, with their temperature dropping from 25 °C to -18 °C and the time is recorded. If there is a super freezing function, after stable conditions are met, this function is activated for a given time (super freezing time) indicated in the user manual of the refrigerator. As soon as this time is completed, the warm packages are introduced into the freezer compartment.

The target compartment temperatures and the loading conditions are presented in the Table 3.4.

Table 3.4. Compartment target temperatures and loading conditions for freezing capacity tests

Standard	Ambient	FFC		CC		FC	
	(°C)	Target temp.(°C)	Measured by	Target temp.(°C)	Measured by	Target temp.(°C)	Measured by
EN 62552:2013	25.0 ±0.5	≤ +5.0	Packages	-2.0 ≤Tcc≤ +3.0	Packages	≤ -18.0	Packages
IEC 62552:2015	25.0 ±0.5	≤ +4.0	Packages	Not measured	Packages	≤ -18.0	Packages
IC 15068	25.0 ±0.5	≤ +4.0	Packages	≤ +4.0	Packages	≤ -18.0	Packages

The freezing capacity test results are given in Table 3.5. It is clearly seen that there are no significant differences in the freezing time of reference and hybrid refrigerators.

Table 3.5. Overview of freezing capacity test results

Standards	EN 62552:2013	IEC 62552:2015	IC 15068	
FC super activation	*Yes*	*yes*	*no*	*yes*
Freezing time	[h]	[h]	[h]	[h]
Serial-1	19.7	-	24.4	12.8
Serial-2	19.2	11.3	27.8	14.2
HHR-1	20.4	10.7	28.2	13.4
HHR-2	17.6	11.3	26.6	13.8

One could expect a decrease in the freezing time of the hybrid refrigerators since one of their compartments has its own separate cooling system. Namely, compressor-based

system is supposed to spend more time to freeze the warm packages throughout the freezing process. To figure out this, we should refer to the compressor run time of the refrigerators. Table 3.6 presents the ratio of the compressor run time for each compartment during the freezing capacity tests of serial and hybrid refrigerators. The FC has already had the highest share in the run time of serial ones over the other compartments with an average value of 85.6%; consequently, there is less room for improvement. Even though run time for HHR-2's FC increased to 95.1%, this had almost no effect on the freezing performance.

Table 3.6. Compressor run time ratio for each compartment during freezing time

Standard	**EN 62552:2013**			
	Serial-1	**Serial-2**	**HHR-1**	**HHR-2**
FFC, CRTR [%]	7.9	9.5	11.0	0.0
CC, CRTR [%]	5.4	6.0	0.0	4.9
FC, CRTR [%]	86.7	84.5	89.0	95.1

There are other key factors that affect the freezing time such as the instant temperatures of the warm and cold packages in the moment of loading, loading duration and deviations in temperature measurements. In each test, the temperature variations between the warm packages were kept at ±0.1 °C. The differences in average temperature of warm packages in the tests are maintained at ±0.3 °C. Additionally, it was made sure that the allocated time for the loading was less than 1min. Accordingly, even though there is a probable enhancement due to increase in the FC compressor run time, these factors may eliminate this.

3.1.5. Noise test

The fans are located on the finned heat sinks of the TEC in order to increase heat transfer. The fans significantly improve heat removal through the heat sinks and therefore enhance the overall performance and COP of the TEC (Astrain, Aranguren, Martinez, Rodriguez, & Perez, 2016). Although in the TEC core system there are no moving components, using fan could lead to high noise level, thus eliminating one of its key advantages.

In the datasheet of the TEC, the external fan rate is given as 51 dB(A). It was therefore assumed that hybrid refrigerators could have higher noise levels compared to the serial ones which have a noise declaration of 38 dB(A).

Table 3.7. Average noise test results

Models	**Noise level dB(A)**
Serial-1	35.6
Serial-2	35.6
HHR-1	62.2
HHR-2	65.1

In Table 3.7, the average noise results are given. Hybrid refrigerators' noise levels are almost twice those of serial ones. In Figure 3.11 and Figure 3.12, the noise test diagram of Serial-1 and HHR-1 are shown, respectively to further demonstrate the effect of the TEC fan on the noise level. In Serial-1, the noisiest component is the compressor and it is followed by the evaporator fan. As presented in Figure 3.11, the maximum noise level of Serial-1 did not exceed 40 dB(A). However, HHR-1's maximum noise level is up to 65 dB(A) which is triggered by the external TEC fan, see Figure 3.12.

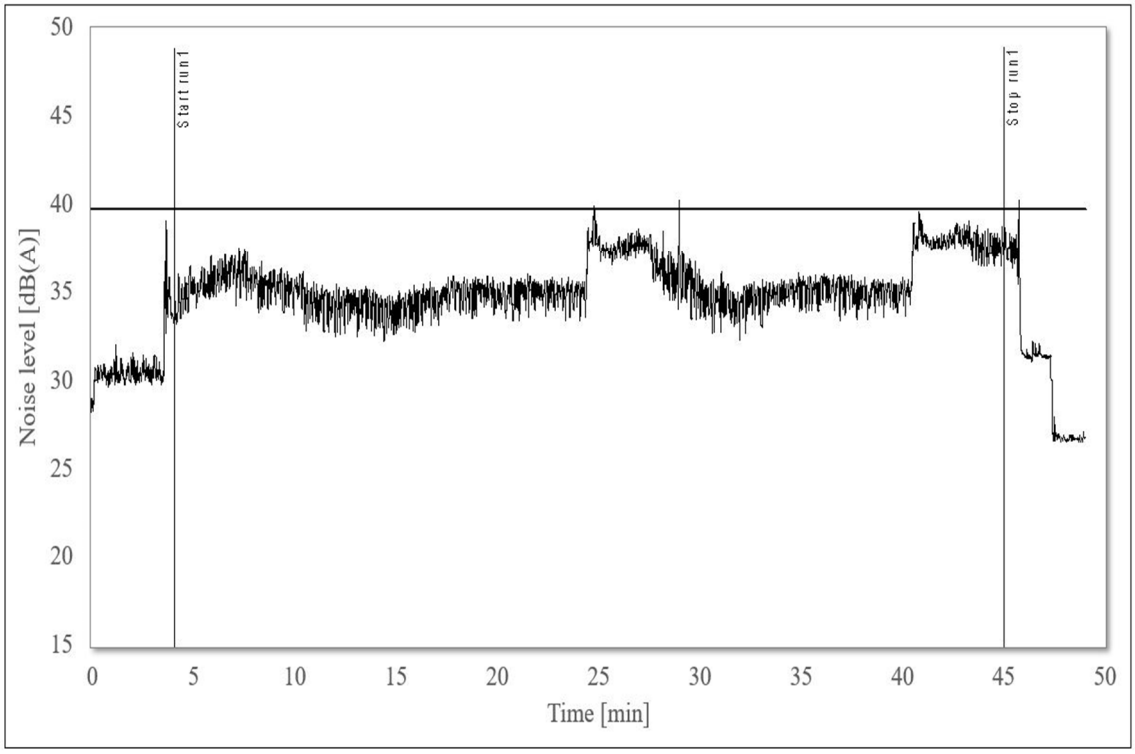


Figure 3.11. Noise level of Serial-1

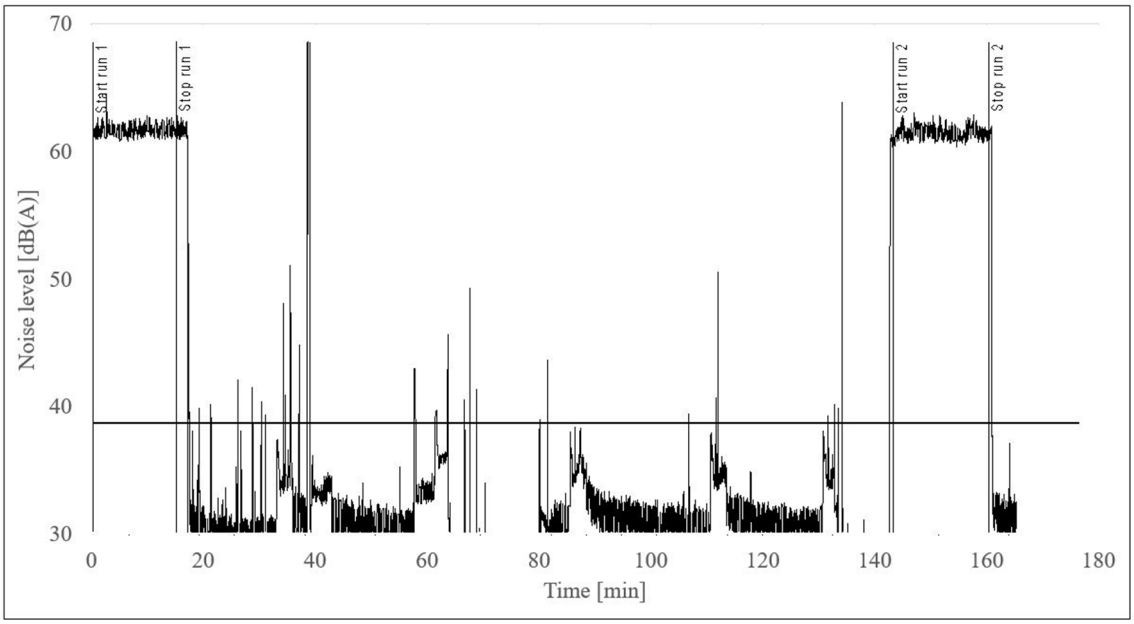


Figure 3.12. Noise level of HHR-1

3.2. Results of the Second Phase of the Work

3.2.1. CFD Results

After developing numerical model and validating it with the measurements for the FFC of HHR-2 as reference, the CFD analysis for the proposed five configurations were conducted with the same solution method and mesh size of around 3,500,000. In each configuration, the TEC was placed at a different location.

3.2.1.1. Reference CFD Analysis

As mentioned in Section 2.3.2.2, the inner wall surface temperatures of the HHR-2's FFC, including TEC's cold face, were measured from 14 different points as shown in Figure 2.8. To validate the CFD predictions, numerical and experimental results were compared for these temperatures.

Table 3.8 shows the comparison of temperature predictions at the 14 locations with the corresponding measurements. Overall, there is a good agreement between the CFD predictions and experimental results.

Table 3.8. Surface temperatures

Temperatures	Test [K]	CFD [K]	Δavg [K]
T1	278.15	279.12	0.96
T2	277.84	278.52	0.68
T3	279.26	280.38	1.12
T4	277.91	279.47	1.56
T5	278.23	279.54	1.31
T6	279.09	279.90	0.82
T7	277.54	280.02	2.48
T8	279.14	279.50	0.36
T9	279.12	280.44	1.32
T10	278.39	279.27	0.88
T11	279.75	281.36	1.61
T12	277.20	278.12	0.92
T13	277.92	276.72	-1.20
T14	278.15	276.68	-1.47

The temperature distribution inside the FFC is demonstrated in Figure 3.13. The lower parts of the compartment are relatively warmer which is likely the consequence of TEC location. The slightly lower temperature region observed at the bottom is due to the location of the lowermost shelf (compartment floor) just above the chill compartment. The regions with the highest temperature values are the door side, between door shelves, and middle bottom of back wall. These results are reasonable because (1) the FFC door is thinner and has less insulation material (polyurethane) compared to the other walls (2) there is an air channel on the middle of back wall that decreases the insulation thickness by 20 mm.

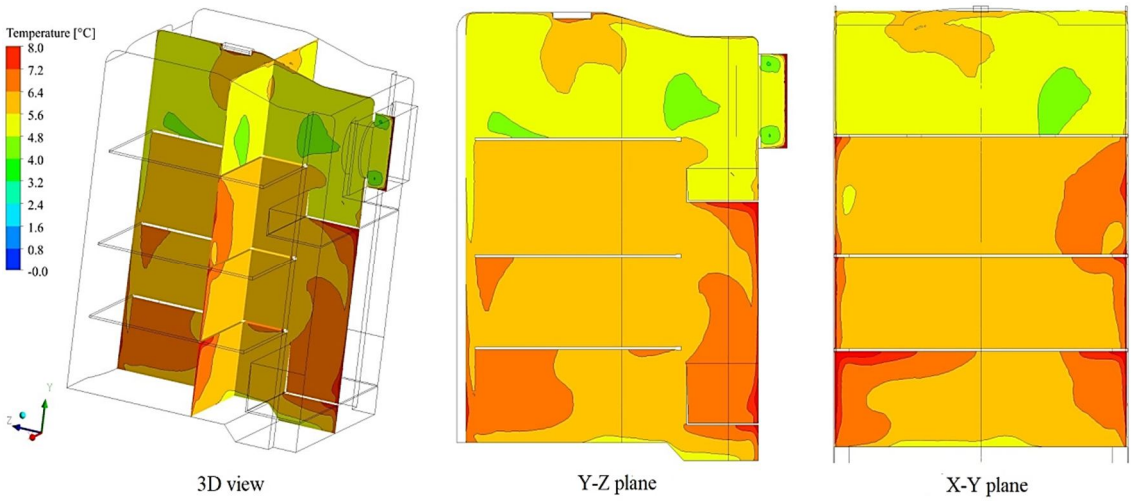


Figure 3.13. Temperature distribution inside the FFC

The streamlines inside the compartment are presented in Figure 3.14. Strong and intense streamlines can be observed in the upper regions, especially in the top part of the compartment which accounts for colder temperature values in this part.

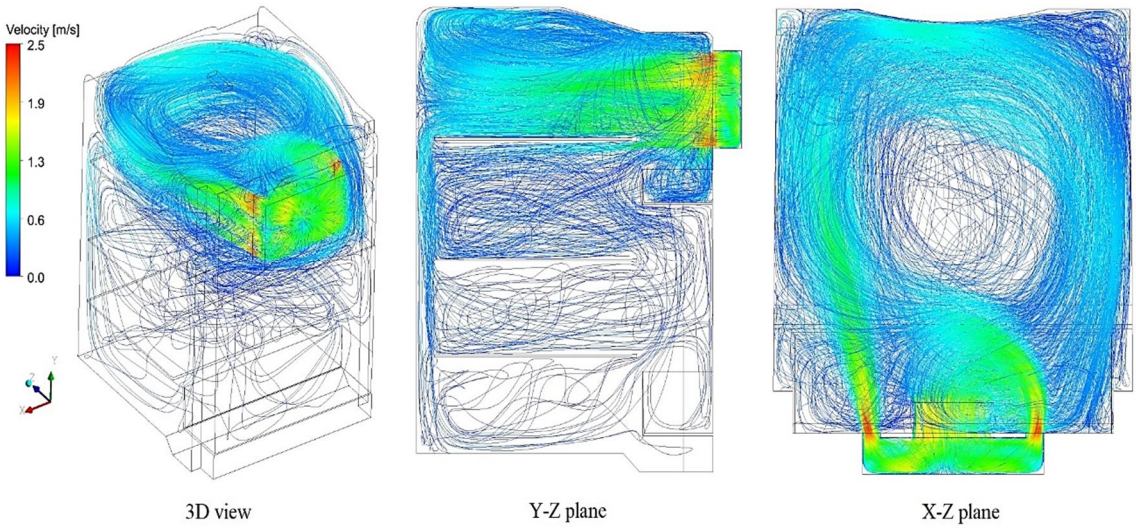


Figure 3.14. Air streamlines inside the FFC

To decrease temperature stratification and therefore, provide more uniform temperature and air flow distribution, the TEC can be located to a more suitable position. For this reason, five different configurations for FFC were proposed.

3.2.1.2. CFD Analysis of Proposed geometries

Configuration-1:

In this configuration, the TEC is placed on the roof of the compartment.

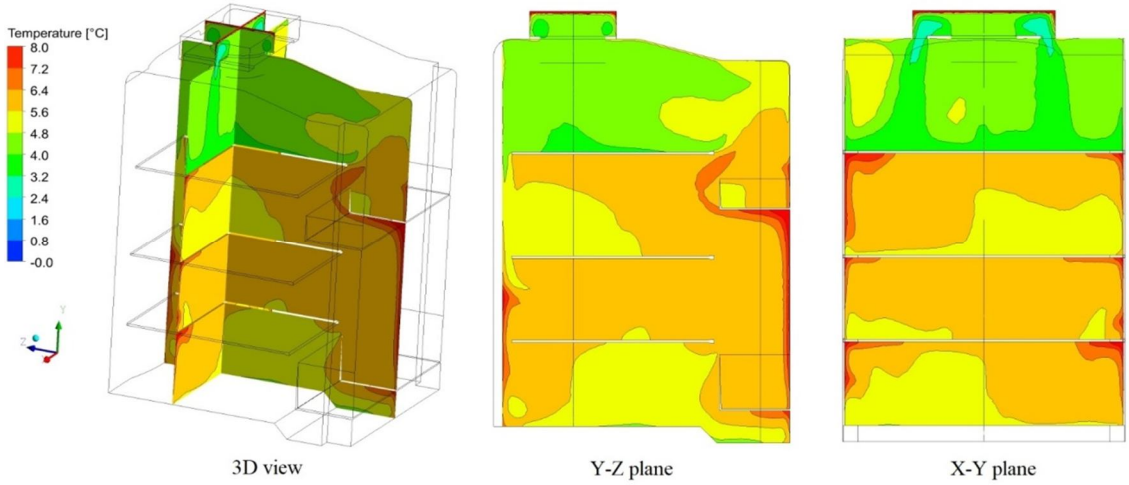


Figure 3.15. Temperature distribution for Configuration-1

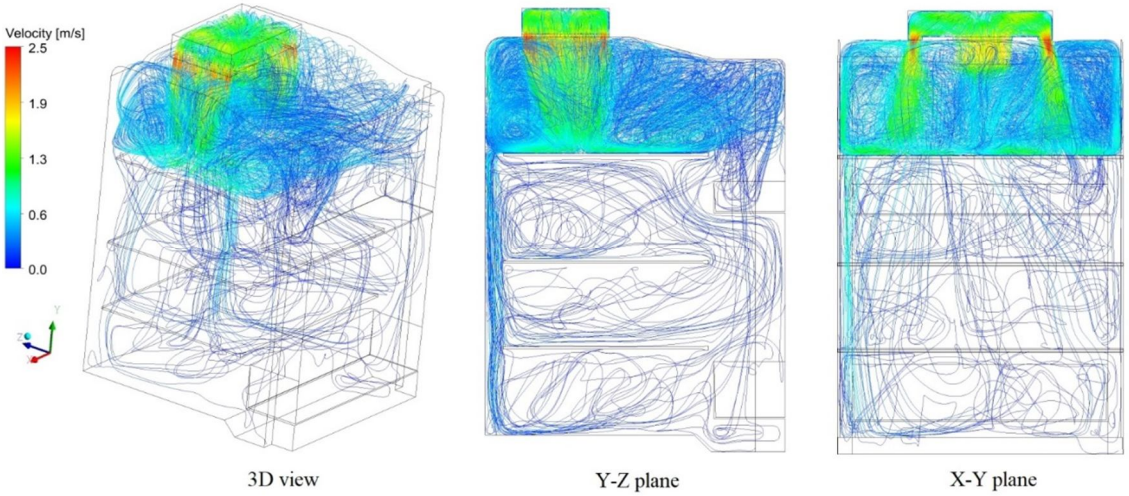


Figure 3.16. Air streamlines for Configuration-1

Configuration-2:

The TEC is located at the top of the back wall of the compartment.

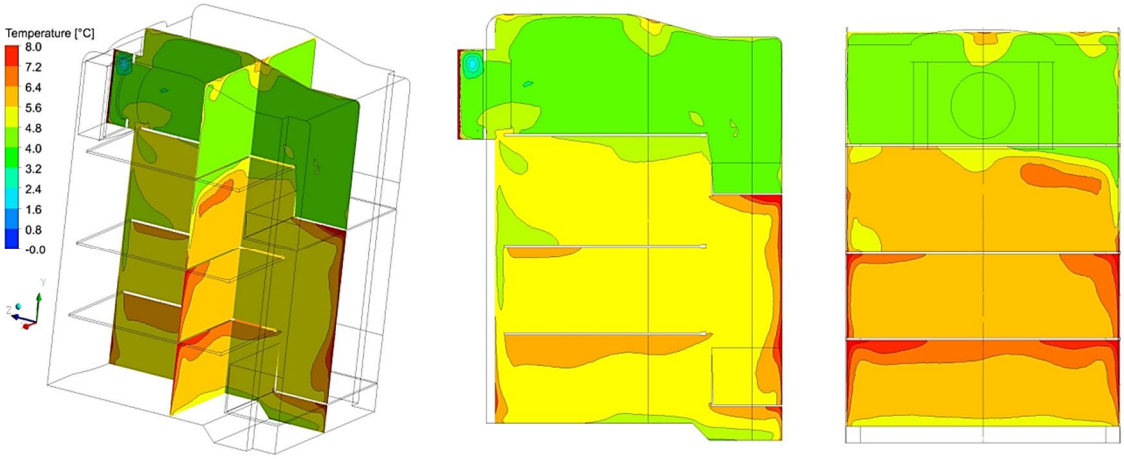


Figure 3.17. Temperature distribution for Configuration-2

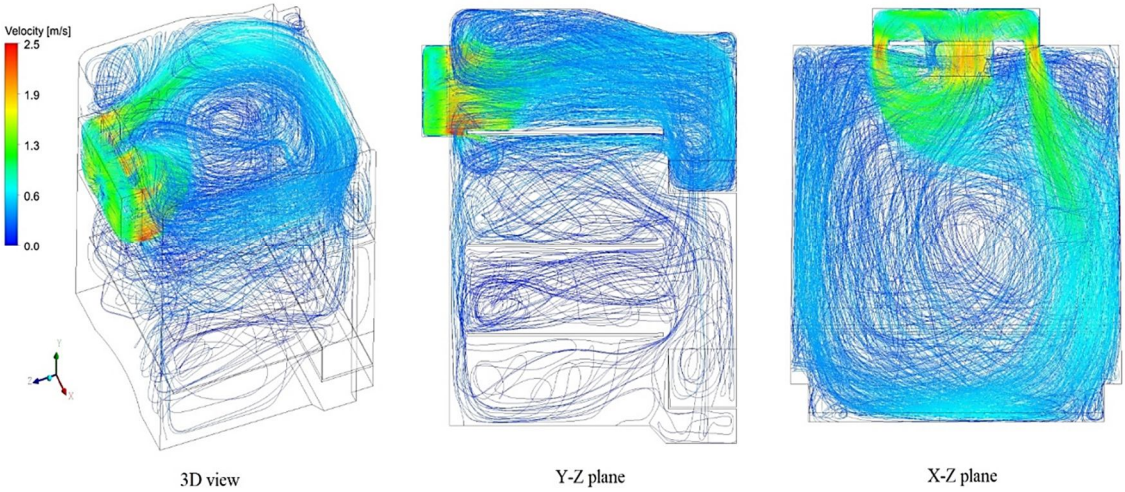


Figure 3.18. Air streamlines for Configuration 2

Configuration-3:

The TEC is placed in the middle of the rear wall of the compartment, between the top and middle shelves.

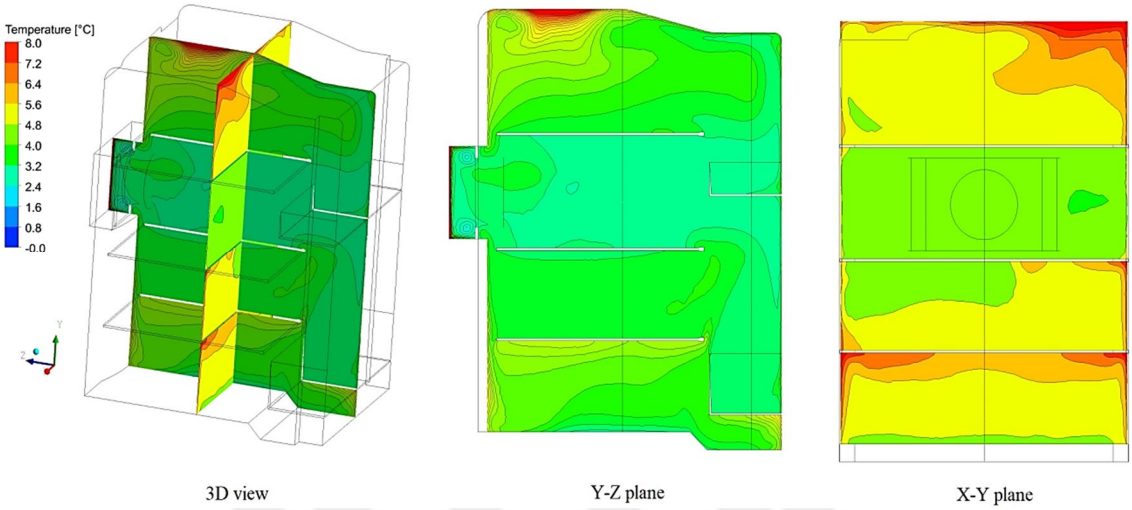


Figure 3.19. Temperature distribution for Configuration-3

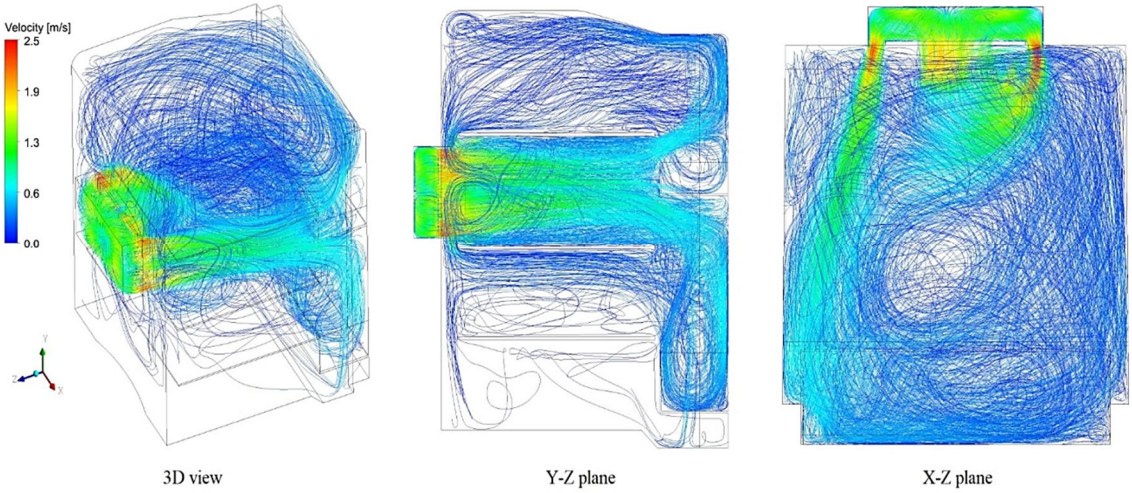


Figure 3.20. Air streamlines for Configuration-3

Configuration-4:

The TEC is horizontally positioned in the middle of the rear wall of the compartment and vertical to the top shelf. The cooled air is pumped to top and middle region of the compartment.

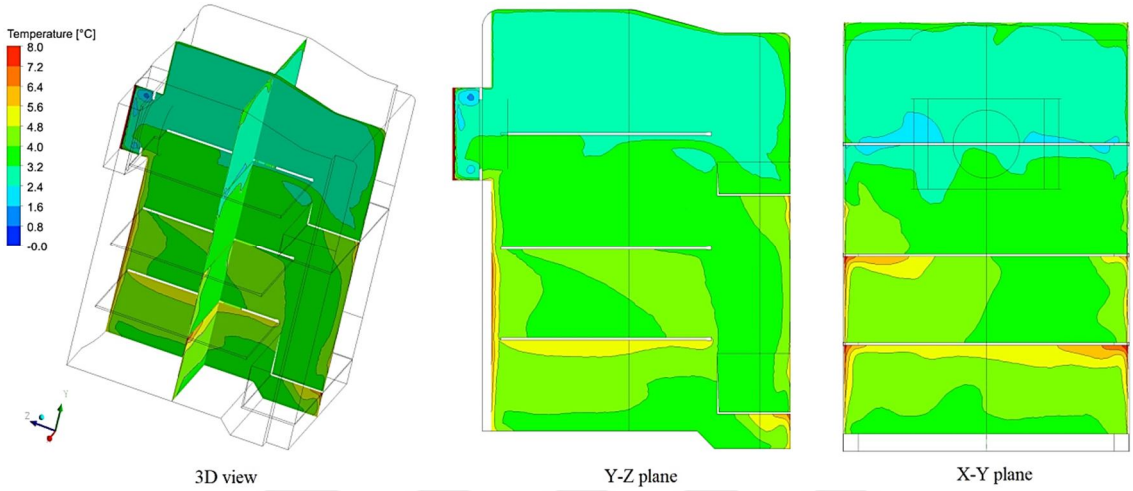


Figure 3.21. Temperature distribution for Configuration-4

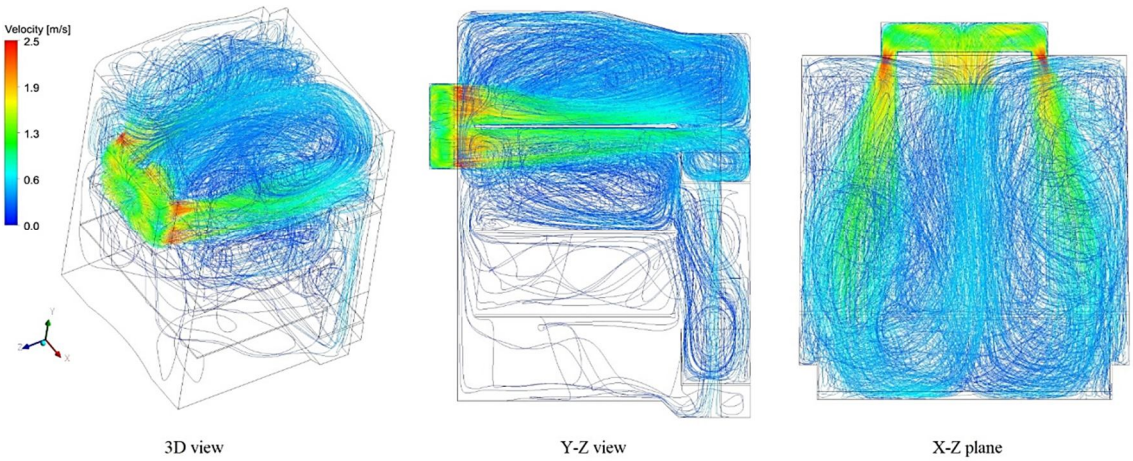


Figure 3.22. Air streamlines for Configuration-4

Configuration-5:

This configuration is like Configuration-4. However, the TEC has a vertical position rather than a horizontal positioning. The goal of this positioning is to pump more cold air to the bottom region of the compartment.

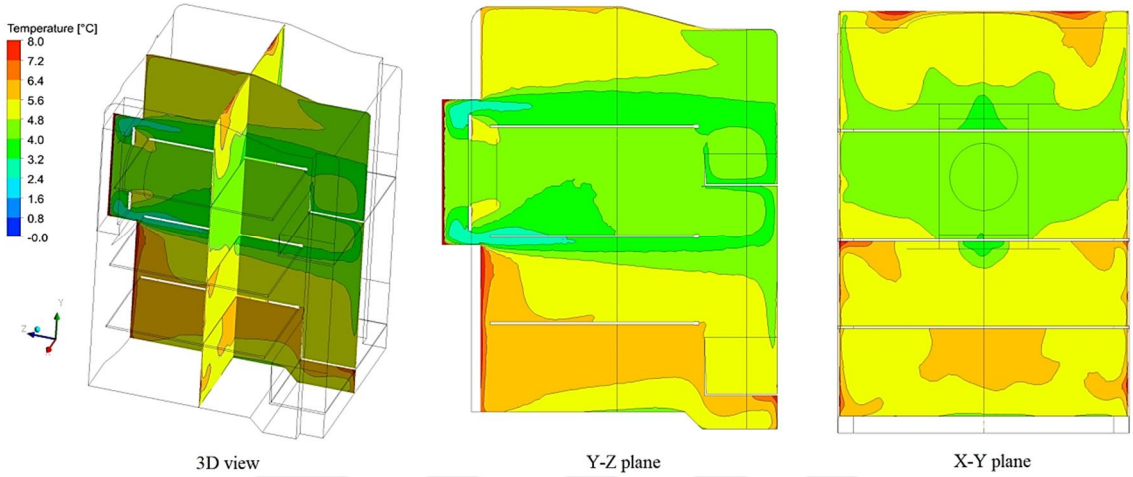


Figure 3.23. Temperature distribution for Configuration-5

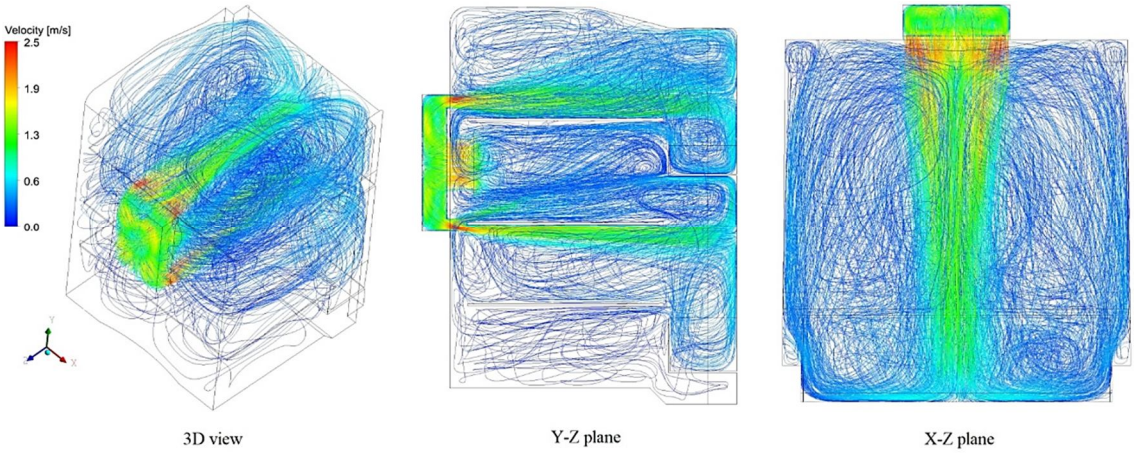


Figure 3.24. Air streamlines for Configuration-5

In Configuration-1 and Configuration-2, it is obvious from the Figures 3.15 to 3.18 that the cooled air is trapped in the upper region of the compartment (between the top shelf and ceiling). For this reason, the upper region has the lowest temperature levels. The warmest zones are observed below the top shelf, particularly on the inner wall of the door between the door shelves and on the side walls just under the middle and bottom shelves.

Configuration-3 has a slightly better air flow and temperature field compared to Configuration-1 and Configuration-2 as seen in Figures 3.19 and 3.20. The cooled air is first pumped to the zone between the upper and middle shelves. The coldest region is the middle region, between the upper and bottom shelves. Although warmer areas on the door side are eliminated in this configuration, elevated temperature levels are observed in the upper region, especially in the zones just under the ceiling.

The location of the TEC in Configuration-4 and Configuration-5 leads to more uniform temperature distribution compared to other configurations (Figure 3.21 and Figure 3.23). The temperature stratifications are almost eliminated. There are few small zones with relatively higher temperature values. Due to this, the previously mentioned HHR-3 and HHR-4 were constructed based on Configurations-4 and Configuration-5, respectively.

3.2.1.3. Samples for HHR-3 and HHR-4

The schematic representation of these HHRs are almost the same with the HHR-2, except the location of the TECs. The CCs and FCs of the HHRs have VC cooling system. The test samples built for HHR-3 and HHR-4 are presented in Figure 3.25 and Figure 3.26, respectively.



Figure 3.25. HHR-3



Figure 3.26. HHR-4

3.2.2. Experimental Results for the Second Phase of the Work

This section contains the energy consumption and cooling capacity test results obtained for HHR-3 and HHR-4 in accordance with IEC62552:2015 (IEC-62552:2015, 2015).

3.2.2.1. Energy consumption

An optimisation study was first conducted by analysing the effect of pre-specified improvement measures. Afterward, with the final configurations, the energy consumption and cooling capacity tests were performed.

3.2.2.1.1. Optimisation study

Measures for the enhancement of energy efficiency were applied to the HHR-3 and the final configurations, which give the highest energy efficiency, were fixed. The same configurations were also used for HHR-4. The temperature hysteresis inside the FFC (TEC hysteresis), the main and idling voltages (IVs) supplied to the TEC and compressor speeds were optimised in the given order. During the measurements for optimisation, only the steady state period of energy consumption test was analysed (defrost and recovery period was neglected) to save testing time. Yet, the energy consumption tests were fully performed with the final fixed configurations.

HHR-2's TEC temperature hysteresis was adjusted regarding the cut-in and cut-off temperatures of the reference (series) KGN39P. In the FFC of this commercial refrigerator, the differences in the minimum and maximum air temperatures ranged from 1 °C to 4 °C. To keep the same temperature ranges and consume less energy, HHR-2's TEC temperature hysteresis was fixed at 0.5 °C.

In this study, it became possible to set the temperature hysteresis of HHR-3 and HHR-4 to 1.1 °C and 2 °C, respectively. The hysteresis was increased thanks to more uniform air flow and temperature fields observed inside their FFC.

Following the hysteresis adjustment, the voltage of TECs' main supply for HHR-3 was optimised at an ambient temperature of 16 °C. The nominal voltage of the TEC main supply was 24 V; voltages of 20 V and 16 V were also supplied to the TEC. The results are presented in Figure 3.27. For all the cases, the compartment temperature was 4.4 °C.

The energy consumption was 0.916 kWh/d at nominal voltage; the lowest value was achieved at a voltage of 20 V with a value of 0.840 kWh/d. The energy consumption dropped by 8.3% due to the 12.2% increase in the TEC's COP. The TEC's COP was 0.41 W/W and 0.46 W/W for the voltages of 24 V and 20 V, respectively. At 16 V, the highest energy consumption was seen due to the sharp increase in the TEC run time. Therefore, the main supply was set to 20 V.

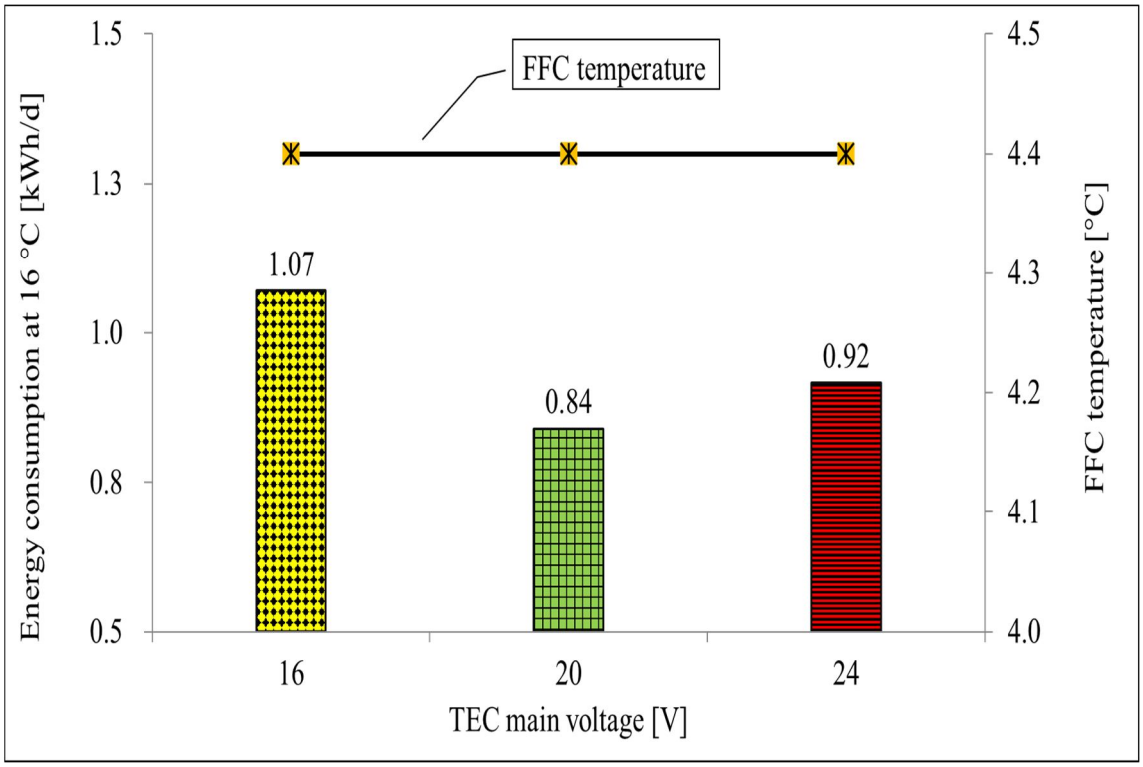


Figure 3.27. TEC main voltage optimisation for HHR-3 at 16 °C

Placing the TEC in the wall of the cabinet leads to an increase in heat transfer from ambient to the inside of the cabinet and therefore reduces the insulation quality. If the TEC has on/off operation cycle, the heat flow through the TEC is high during the off period. As it was proved in Section 2.2.2.1 that the overall transmission losses rose by around 10% when the TEC was off. To avoid heat flow through the TEC, one of the solutions is to supply IVs to the TEC through the off period. That is to say, the TEC will have a continuous operation. The purpose is to keep the temperatures of the TEC's faces as close as the temperature of the environments they are in. The idea was originally proposed by Martinez et al. (Martinez, Astrain, Rodriguez, & Perez, Reduction in the

electric power consumption of a thermoelectric refrigerator by experimetal optimization of the temperature controller, 2013).

In Figure 3.28, the effect of supplied IVs to the HHR-**3**'s TEC on the energy consumption at 16 °C are presented. Here, the lowest energy consumption occurred with an IV of 3.5 V, 0.648 kWh/d which is 23% lower compared to the no IV case (0.840 kWh/d). Further increases in the IV have a negative effect on the energy consumption.

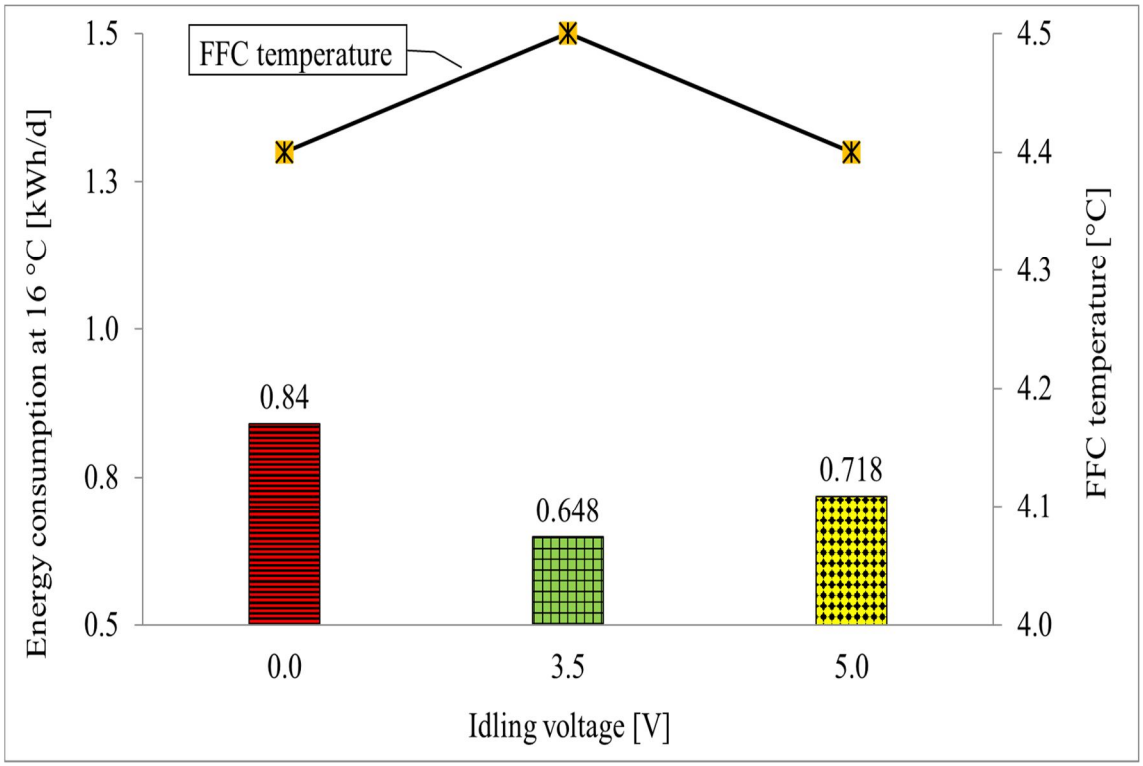


Figure 3.28. Applying idling voltages to HHR-3 at 16 °C

The detailed information about the effect of IVs on the TEC COP and energy consumption is given in Table 3.9. In this table, the analysis is based on a steady state period of approximately 6 h. The temperature of the TEC cold and hot faces and the upper region of the FFC are demonstrated. The temperature sensor location placed in the upper region of the FFC is shown in Figure 3.28. In Table 3.9 and Figure 3.29, the aim is to present temperature changes during the period of analysis to well understand the root causes of COP differences.

Table 3.9. Effect of IVs on the COP of HHR-3's TEC

Idling voltage	T_a	$T_{TEC-cold,min}$	$T_{TEC-cold,max}$	$T_{TEC-hot,min}$	$T_{TEC-hot,max}$	$T_{FFC-upper,min}$	$T_{FFC-upper,avg}$	$T_{FFC-upper,max}$	$COP_{TEC,on}$	$COP_{TEC,idling}$
[V]	*[°C]*	*[°C]*	*[°C]*	*[°C]*	*[°C]*	*[°C]*	*[°C]*	*[°C]*	*[W/W]*	*[W/W]*
0.0	15.9	-2.2	11.6	11.4	21.7	2.7	4.6	6.6	0.46	-
3.5	15.9	-2.2	7.1	15.3	21.7	2.5	4.3	6.2	0.52	2.71
5.0	15.9	-2.5	6.8	17.6	21.7	2.4	4.2	6.1	0.52	2.05

It is clear from Table 3.9 that with the IVs of 3.5 V and 5.0 V when the TEC is on (on period), the TEC's COP is higher than the without IV case by 13%. While supplying the IV to the TEC (off period), the TEC's COP is highest at 3.5 V with a COP value of 2.71 W/W. This explains the reasons for less energy consumption with supplying 3.5 V. To determine the optimum IV and therefore the highest efficiency during the off period, one should carefully analyse the temperature changes of the TEC's cold and hot faces. When supplying IV to the TEC, the aim should be to keep the TEC's hot face's minimum temperature as close to the ambient temperature (T_a) as possible while keeping the TEC's cold face maximum temperature at the same level of compartment temperature surrounding the TEC's cold face (in our case, the upper region of the FFC). Additionally, the minimum temperature of the TEC's hot face ought not to exceed the ambient temperature.

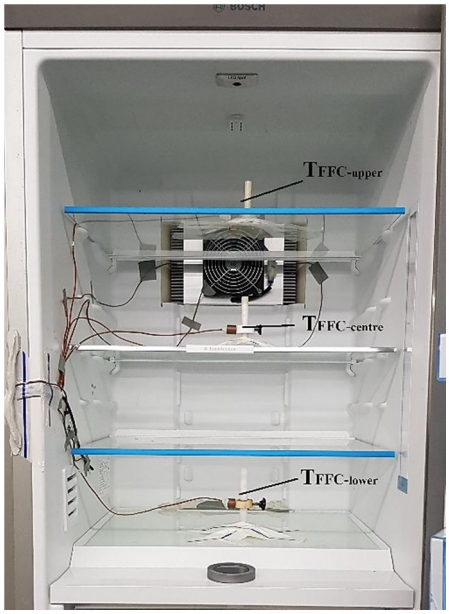


Figure 3.29. Location of temperature sensors in the HHR-3's FFC

The same measurements for determining the optimum IV were also conducted at an ambient temperature of 32 °C. The ideal value of IV to be supplied to the TEC was 8.0 V as demonstrated in Figure 3.30. The energy consumption was decreased by nearly 27% compared to the consumption of no IV case.

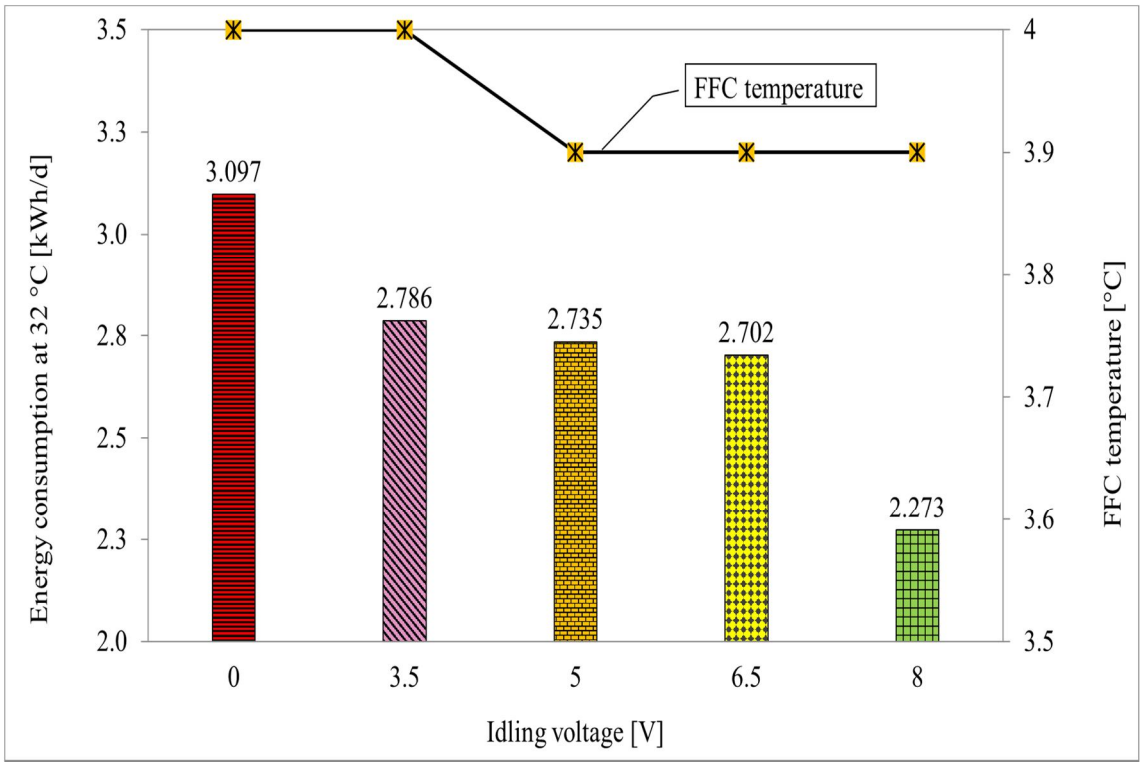


Figure 3.30. Applying idling voltages to HHR-3 at 32 °C

Although the primary focus of this study is to investigate the effect of the TEC on the household refrigerator's performance, an additional improvement step for optimising compressor speed at 32 °C was also conducted. The variable speed compressors used for household refrigerators generally have speed ranges between 1000 rpm and 4500 rpm. At lower room temperatures, the compressor should run at relatively lower speeds due to reduced cooling load. Because HHR's compressors operate at optimum speeds at 16 °C room temperature, the optimisation was conducted at only 32 °C in this study. The compressor speed was decreased by 400 rpm which reduced energy consumption by 2% from 2.273 kWh/d to 2.224 kWh/d.

3.2.2.1.2. Energy consumption test results

After improvement studies of HHR-3 and HHR-4 were completed, the energy consumption tests were performed with their final configurations. The test results are presented in Table 3.10. At 16 °C room temperature, all HHRs have the same compressor speeds, whereas HHR-3 and HHR-4 have 400 rpm less speed compared to HHR-2 at 32 °C (the exact compressor speeds are not mentioned due to privacy policy of BSH Bosch). According to the results, it can be remarked that HHR-3 and HHR-4 have almost identical energy consumption levels. The differences in their energy consumptions at 16 °C and 32 °C are 0.7% and 1.5 %, respectively. Energy efficiency improvements of up to 10% and 27% were achieved at the ambient temperatures of 16 °C and 32 °C, respectively, compared to HHR-2.

Table 3.10. Energy consumption test results

Refrigerator	Main supply	IV		Energy consumption	
	[V]	at 16 °C *[V]*	at 32 °C *[V]*	at 16 °C *[kWh/d]*	at 32 °C *[kWh/d]*
HHR-2	24.0	-	-	0.844	3.083
HHR-3	20.0	3.5	8.0	0.754	2.285
HHR-4	20.0	3.5	8.0	0.759	2.254

3.2.2.2. Cooling capacity

In the standard (IEC-62552:2015, 2015), this test is named as the “cooling capacity test”. The reason for that is after measuring the time for the cooling of warm packages from 25 °C to 10 °C, load mass is calculated in kg. For the purpose of this study, only the cooling time was measured during the tests because this will provide an adequate idea about the performances of the HHRs.

Before performing the test, the HHRs were prepared and the temperature sensors were located as previously for the energy consumption test as shown in Figure 3.28. As soon as the steady state period was achieved, and the target compartment temperatures indicated in Table 3.2 were met, the warm packages, which were being kept at an ambient temperature of 25 °C, were inserted into the FFC according the load plan

prepared in compliance with the standard. The temperature of the warm packages was 25 °C ±0.5 °C in the moment of insertion.

The test results for the cooling down time of the HHRs are summarised in Table 3.11. At a nominal voltage of 24 V, all HHRs have pretty much the same cooling time results. Although more uniform temperature and air flow distribution was achieved by relocating the TEC, this did not affect the cooling time of HHR-3 and HHR-4. One should also note that, however, the CFD predictions were obtained without the loads, which would definitely affect the air flow inside the FFC.

Table 3.11. Test results for cooling time

Refrigerator	**Main supply**	**IV (16 °C/ 32 °C)**	**Cooling time**
	[V]	***[V]***	***[min]***
HHR-2	24.0	-	146.6
HHR-3	20.0	3.5/8.0	164.5
	24.0		146.5
	26.5		142.1
HHR-4	20.0	3.5/8.0	165.4
	24.0		144.7
	26.5		139.5

To decrease the cooling time of HHR-3 and HHR-4, the tests were carried out with the main supply of 24 V and 26.5 V. As seen from Table 3.11, the lowest cooling time was observed at 24 V and a slight decrease was achieved with 26.5 V. At 20 V, despite increase in the energy efficiency, the cooling time of the HHRs increases.

4. CONCLUSIONS & FUTURE WORK

Effects of current commercial TECs on the performance of household refrigerators were investigated. Analysis focused on the benefits of TECs to provide opportunities for the application of the thermoelectric cooling technologies.

To do this, in the first phase of this work, experimental investigation was conducted. Two serial refrigerators were tested initially to collect reference data. Afterwards, HHRs were built by attaching TECs to the walls of the serial refrigerators' body. The same test procedures were followed and then the HHRs' test results were compared to the results of the serial ones.

In this phase, it was demonstrated that energy consumption of HHR-1 and HHR-2 was higher than those of serial ones. The fundamental reason for this was the low COP of the TECs. Lower COP levels lead to higher power consumption. Another key reason was higher heat transmission losses through the TECs.

In terms of cooling time, it was proved in this study that the TECs can cool 17.5 kg of loads from an ambient temperature of 25 °C to the desired temperature as mentioned in Section 3.1.3. A special interest was given to HHR-2 because it achieved the best performance; its cooling time was 30-40% less than those of the others. It was also validated that HHR-2's FFC can be declared as having rapid cooling feature owing to the achieving a cooling speed of 0.10 °C/min and 0.14 °C/min for the 7.5 kg and 2.5 kg of warm packages, respectively.

A freezing capacity test does not directly measure the influence of the TEC on the refrigerator performance. It is clearly seen that there are no significant differences in the freezing time of reference and hybrid refrigerators. Although one could expect a decrease in the freezing time of the hybrid refrigerators since one of their compartments has its own separate cooling system, such a situation did not come up in our tests.

HHRs' noise levels were much higher than those of serial ones. Higher noise levels of HHRs were triggered by the external TEC fan. In the core of a TEC, there is no moving parts or a fluid which may cause disturbing noise (in VC system, refrigerants contribute to flow sound). Nevertheless, a TEC should be evaluated as an entire system considering all its components including heat sinks and fans. The solutions for less noise levels can

be: (1) mounting a fan on the hot side of TEC with less noise rate, (2) increasing heat sink surface area of TEC hot side and removing fan and (3) implementing a different heat sink to the hot side of TEC without using a fan (for example, heat pipes). Yet, the proposed last two solutions will not be as efficient as the one with fan. The improvement of noise levels of HHRs was out of the scope of this work. It was simply aimed to present effect of TEC's fan on the noise level of refrigerators.

In the second phase of the work, because of its lower energy consumption and promising cooling down time rates, HHR-2 was determined for further improvements.

In the beginning, in order to find the best location for the TEC to deliver more uniform air flow and temperature distribution inside the FFC, CFD analysis was performed. HHR-2's FFC was initially analysed as reference to develop the most suitable numerical model for this cooling application. After that, proposed five configurations, in which the TECs were in various positions, were numerically analysed.

Two new HHRs, HHR-3 and HHR-4, based on the configurations having more uniform air flow and temperature distributions were established. Experimental analysis was conducted to validate CFD predictions. Unlike the experimental investigation carried out in the first phase, only energy consumption and cooling capacity tests according to IEC62552:2015 were performed to save testing time.

Before starting the tests, optimisation and improvement studies were first implemented in the following order:

* The temperature hysteresis for TEC was determined.
* The optimum values for the main and idling voltages (IVs) were determined supplied to the TEC.
* Compressor speeds were optimized.

Thereafter, the tests were performed with the final configurations of the HHRs. The test results showed that in the FFC of HHR-3 and HHR-4, more uniform temperature distribution was achieved as predicted by CFD.

Moreover, HHR-3 and HHR-4 had almost identical energy consumption levels. For both of them, energy efficiency improvements of up to 10% and 27% were achieved at the ambient temperatures of 16 °C and 32 °C, respectively, compared to HHR-2. Despite

these significant improvements, their energy consumptions were 147.4% and 164.6% higher than those of serial ones at the ambient temperatures of 16 °C and 32 °C, respectively. Regarding cooling time, no further improvement was achieved.

It is obvious that although thermoelectric cooling technology is less efficient compared to the compressor-based system, it brings significant benefits in terms of lower cooling times and higher temperature uniformity inside the target compartment as proved. Therefore, this technology can be applied to household refrigerators to meet the particular needs of the end-users in the short term.

The stricter requirements for the measurements of energy consumption have been highlighted in the forthcoming standard to prevent test recognition tools; in other words, the usage of circumvention devices and the appliances with higher efficiency are demanded with the new regulation. This will certainly make it harder to use TECs in household refrigerators in the near and mid future. However, there are a great deal of research activities to increase TECs' efficiency and therefore they may be implemented to household refrigerators in the long run. The outputs of this study are going to be a baseline for further works and let researchers to find more optimum ways to apply the TECs to the refrigerators.

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PUBLICATIONS

Experimental analysis of hybrid household refrigerators including thermoelectric and vapour compression cooling systems (*status: accepted but not published yet, International Journal of Refrigeration*)