

**DOKUZ EYLÜL UNIVERSITY**  
**GRADUATE SCHOOL OF NATURAL AND APPLIED SCIENCES**

**VIBRO-ACOUSTIC ANALYSIS AND  
IMPROVEMENT OF REFRIGERATORS WITH  
DIFFERENT TYPES**

by

**Bircan DEMİRTEKİN**

**August, 2016**

**İZMİR**

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IMPROVEMENT OF REFRIGERATORS WITH  
DIFFERENT TYPES**

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

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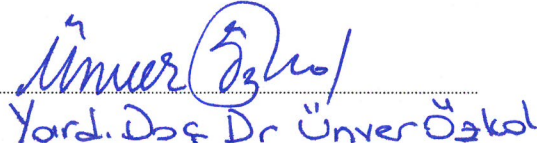
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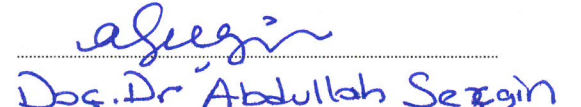
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## ACKNOWLEDGMENTS

I would like special thanks my thesis advisor Prof Dr. A.S. SARIGÜL for all her time, effort and patience.

I would also like to thank my mother, my brother Ercan DEMİRTEKİN and my friend Asil MADENLİ.

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# **VIBRO-ACOUSTIC ANALYSIS AND IMPROVEMENT OF REFRIGERATORS WITH DIFFERENT TYPES**

## **ABSTRACT**

Vibration and noise are important environmental issues nowadays. Noise may affect human health by its level, type or time. This effect is in the form of threatening the human health or at least reducing her/his life standard.

Vibro-acoustics is one of the basic criteria of our present day's technological level of products. Vibro-acoustic studies are important not only in academia but also in industry. There is a competition between research and development of different companies for more silent products. Therefore the Research & Development (R&D) departments following innovative ideas focus on vibro-acoustic implementations. This trend is also valid for white good sector.

This study deals with refrigerators, one of the white goods. The aim is to achieve silent refrigerators with different solution proposals. Substantial acoustic measurements were performed; design modifications on the basis of these tests were developed and applied. Alternative solutions were accomplished for different types of refrigerators, noise levels and sources. The noise reduction for each case was satisfied and vibro-acoustic improvement of the refrigerators was achieved successfully.

**Keywords:** Vibro-acoustic, noise, refrigerator

# FARKLI TİP BUZDOLAPLARININ VİBRO-AKUSTİK ANALİZİ VE İYİLEŞTİRİLMESİ

## ÖZ

Günümüzde titreşim ve gürültü önemli çevresel konulardır. Gürültü insan sağlığını, gerek düzeyi, gerek tipi ve gerekse zamanı açısından etkileyebilir. Bu etki insan sağlığı için tehlikeli olabileceği gibi yaşam standartlarını da düşürebilir.

Vibro-akustik günümüz ürünlerinin teknolojik seviyesini gösteren en temel ölçütlerden biridir. Vibro-akustik çalışmalar sadece akademi açısından değil endüstri için de önemlidir. Farklı firmaların araştırma ve geliştirme konusundaki çalışmalarını arasında daha sessiz ürünlerin tasarımı için rekabet görülür. Bu nedenle yenilikçi fikirleri takip eden Araştırma-Geliştirme (Ar-Ge) bölümleri vibro-akustik uygulamalara yoğunlaşır. Bu eğilim beyaz eşya sektörü için de geçerlidir.

Bu çalışmada beyaz eşya ürünlerinden olan buzdolapları ele alınmıştır. Çalışmanın amacı farklı çözüm önerileri ile sessiz çalışan buzdolaplarının geliştirilmesidir. Çok sayıda akustik ölçümler gerçekleştirilmiş, bu testler temelinde tasarım değişiklikleri geliştirilmiş ve uygulanmıştır. Farklı tip buzdolapları, gürültü düzeyleri ve kaynakları için alternatif çözümler üretilmiştir. Her durum için gürültü azaltımı sağlanmış ve buzdolaplarındaki vibro-akustik iyileştirme başarı ile sağlanmıştır.

**Anahtar kelimeler:** Vibro-akustik, gürültü, buzdolabı

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## CHAPTER ONE

### INTRODUCTION

Sound propagates in waves causing changes in the pressure of acoustic media. It is a form of energy and it constitutes an unavoidable part in our lives. On the other hand, noise being undesired, irritating form of the sound has a disturbing effect on the environmental comfort.

Sound is an important quality aspect. In today's world, sound of products has become a significant problem in white goods sector. When the current situation of white goods technology is reviewed, it can be seen that substantial achievements have been made as far as the energy efficiency, water consumption, and ergonomic structures are concerned. Studies to produce new appliances operating more silently and adaptable to new technologies as being environmental friendly are carried on. White goods industry invests a large amount of money to remove or at least to reduce the vibro-acoustic problems.

Vibro-acoustic studies advance year by year and include benchmarking. Before developing a new product, almost every manufacturer should be able to list the critical success factors and make a marketing research. Companies always compare their products' sound levels with their competitor products' declared levels. The sound of a white good is always an important marketing concept. Table 1.1 presents a marketing research for refrigerators in terms of the declared sound power levels.

Table 1.1 Marketing research for refrigerators' sound power levels

Marketing Research	Refrigerator (dB(A) / Lt )				
	Single	Double	No Frost	Combi	Mini
A	42 / 350	44 / 424	44 / 480	43 / 480	41 / 134
B	38 / 330	40 / 520	42 / 480	42 / 470	38 / 144
C	38 / 496	40 / 520	42 / 480	42 / 560	37 / 142
D	42 / 350	43 / 450	43 / 480	43 / 480	No info

The vibro-acoustic sources of refrigerators may be listed as compressor, compressor's chassis, fan, refrigerant and refrigerator panels. Refrigerant causes flow noise. Cracking noise during the defrost process is another type of noise. In some cases, compressor chassis and refrigerator panels may vibrate in resonance.

This thesis follows the crucial, popular trend in technology and attempts to improve refrigerators in terms of vibro-acoustic quality. Refrigerators handled have considerable noise levels, different sources and types of noise. They do not have resonance problem. Therefore, accomplishments in alternative ways were made to attenuate their noise.

In this thesis two types of refrigerators have been examined. The first refrigerator has high capacity (820 lt) and many components. This refrigerator has been investigated in terms of overall noise as well as compressor and fan noise. Any noise study on this type of refrigerators had not been published in the open literature yet. Therefore this study would make a contribution. On the other hand, patent applications have been made for the solutions proposed in this study.

The second refrigerator is standard, household type. Analyses have been made with regard to defrost noise and flow noise. Defrost noise had not been ever studied in the literature. Therefore this study constitutes a contribution too. Although flow noise has a high impact in the literature, the solution to reduce this noise in this thesis is new and includes originality.

With regard to the contents of the thesis, the first chapter is introduction. The second chapter includes an explanation of refrigeration system. The third one presents sound definitions and measurements. The fourth, is related with overall, compressor and fan noise of the refrigerators. In this chapter, the high-capacity refrigerator is examined. The fifth, includes defrost noise and the sixth, contains flow noise. Standard household refrigerator is considered in both of these chapters. The last chapter is devoted to conclusion of whole thesis, although chapters four, five and

six include their own conclusions. On the other hand, these chapters have their own literature surveys. Therefore, they have integrity in themselves.



## CHAPTER TWO

### REFRIGERATION SYSTEM

Refrigeration is the process of removing heat from a lower-temperature domain and discarding it to a higher-temperature. Heat naturally flows from hot to cold divisions.

Refrigeration is commonly used for keeping food, vegetable and beverage fresh. It has many applications in everyday's life including chilling, freezing and air-conditioning. Precise temperature control is necessary for safe keeping of foods as well as the quality of the refrigeration components.

#### 2.1 Refrigeration Cycle

A refrigeration cycle is composed of four processes, as shown in Figure 2.1;

- Compression
- Condensation
- Throttling
- Evaporation

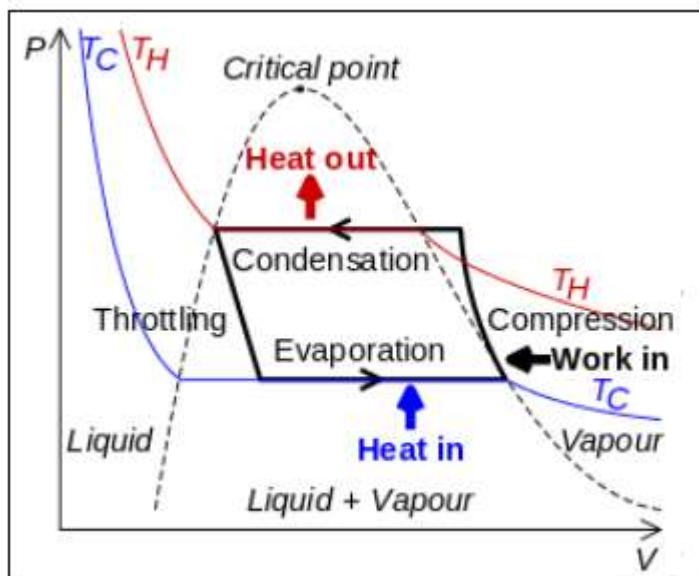


Figure 2.1 A fictitious pressure-volume diagram for a typical refrigeration cycle (Çengel & Boles, 1996)

## 2.2 Refrigeration System Components

A typical refrigeration system is shown in Figure 2.2. System has five basic components:

- Compressor
- Condenser
- Expansion valve
- Evaporator
- Refrigerant fluid

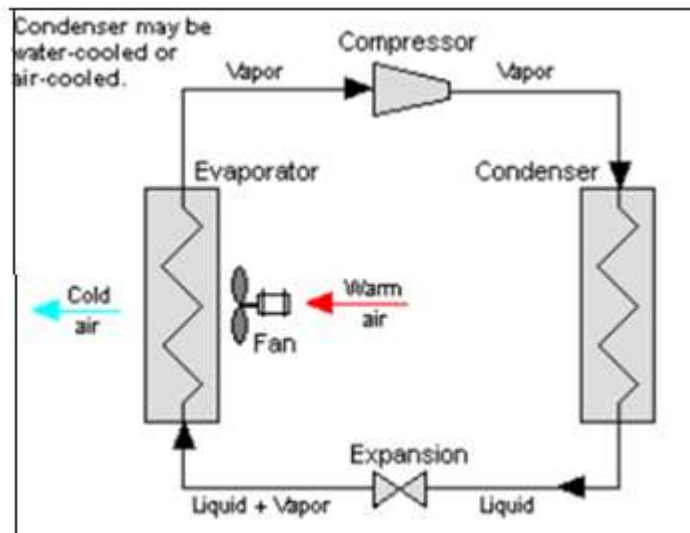


Figure 2.2 A schematic view of the refrigeration system (Çengel & Boles, 1996)

### 2.2.1 Compressor

Compressor draws the low-temperature, low-pressure vapor from the evaporator via the suction line. Then transforms this vapor into a high-temperature, high-pressure vapor. After, pushes this vapor into the condenser coils on the outside of the refrigerator.



### ***2.2.2 Condenser***

The high-pressure refrigerant vapor from the compressor is then cooled to the point where it becomes a liquid within the condenser.

### ***2.2.3 Expansion Valve***

The high-pressure liquid from the condenser reaches to the expansion valve. The valve reduces the pressure of the refrigerant isenthalpic as it passes through an orifice, located inside the valve. Therefore, temperature of the refrigerant drops below that of the surrounding air. In this throttling process, a two-phase (liquid + vapor) flow is produced and directed to the evaporator.

### ***2.2.4 Evaporator***

The cold mixture is directed to evaporator coils through capillary tubes to be vaporized. The refrigerant directed from expansion valve to evaporator is in low temperature and low pressure. Evaporator absorbs heat of air inside the fridge and freezer. This heat is transferred to the refrigerant. This procedure may be supported by a fan unit.

### ***2.2.5 Refrigerant Fluid***

The refrigerant fluid is used commonly as a "high-temperature" refrigerant for the refrigeration system. In this study, R600a was used in the refrigerators handled.

## CHAPTER THREE

### SOUND DEFINITIONS AND MEASUREMENTS

Sound is the physical result of pressure changes due to vibrations of the particles forming the medium. Sound is an indispensable part of our daily life and a criterion to evaluate the quality of tools which we use in daily life. Noise is realized as the disturbing sound that decreases life standards. Both sound and noise are concepts of our everyday life.

#### 3.1 Definitions, Terms and Units

##### 3.1.1 Sound Pressure ( $p$ )

Sound pressure is the difference between the pressure produced by a sound wave and the barometric pressure at the same point. It is measured in Pascal  $Pa$  (Beranek, & Ver, 2006).

##### 3.1.2 Sound Pressure Level ( $L_p$ )

Sound pressure level is a logarithmic measure of effective pressure of a sound relative to a reference value.

$$L_p = 10 \log \left( \frac{p_{rms}}{p_{ref}} \right)^2 = 20 \log \left( \frac{p_{rms}}{p_{ref}} \right) \text{ dB} \quad (3.1)$$

where,

$p_{ref} = 2 \times 10^{-5} \text{ Pa}$  (Reference sound pressure)

$p_{rms}$ : Root mean square value of the pressure, it is defined as

$$p_{rms} = \left[ \frac{1}{T} \int_0^T p(t)^2 dt \right]^{1/2} \quad (3.2)$$

Here,  $t$  is time,  $T$  is period.

### 3.1.3 Sound Intensity ( $I$ )

Sound intensity is the acoustic energy passing through a unit normal area in the medium in unit time. The SI unit for sound intensity is  $Watt/m^2$ . It is also defined as the time average of the multiplication of sound pressure and particle velocity  $U$ .

$$I = \frac{1}{T} \int_0^T p(t)U(t)dt \Rightarrow I = \overline{pU}. \quad (3.3)$$

Sound intensity of a source varies with the position of the receiver. This means that, sound intensity has directivity property.

### 3.1.4 Sound Intensity Level ( $L_I$ )

Sound intensity level is a logarithmic quantity and can be expressed as,

$$L_I = 10 \log \left( \frac{I}{I_{ref}} \right) \text{ dB}. \quad (3.4)$$

where

$$I_{ref} = 1 \times 10^{-12} \text{ Watt}/m^2 \text{ (Reference sound intensity)}$$

### 3.1.5 Sound Power ( $W$ )

Sound power is the energy of sound radiated from a sound source in unit time. The SI unit of sound power is  $Watt$ . Its theoretical formulation is given as the integration of sound intensity along the surface area.

Sound power is a characteristic quantity of sources that does not change with distance and does not have directivity.

### 3.1.6 Sound Power Level ( $L_w$ )

Sound power level is a logarithmic measure of sound power relative to a reference value.

$$L_w = 10 \log \left( \frac{W}{W_{ref}} \right) dB \quad (3.5)$$

where,

$W_{ref} = 1 \times 10^{-12}$  Watt (Reference sound power)

### 3.1.7 Equivalent Sound Level ( $L_{eq}$ )

$L_{eq}$  is a sound indicator used for the evaluation of noise with variable amplitudes. It is assessed using the time average of the measured levels.

$$L_{eq} = 10 \log \left( \frac{1}{T} \int_0^T \left( \frac{p}{p_{ref}} \right)^2 dt \right) dB \quad (3.6)$$

## 3.2 Sound Measurements

Reliable sound measurements of appliances should be performed in anechoic, semi-anechoic or reverberation chambers with respect to the necessary standards. The measurements in semi-anechoic and reverberation rooms are shown in Figures 3.1 and 3.2, respectively. These measurements are made to obtain the sound pressure and/or sound power levels of the refrigerators. The acoustic characteristics of acoustic rooms in which the measurements performed are presented in Table 3.1.



Figure 3.1 Sound measurement of a refrigerator in semi-anechoic room



Figure 3.2 Sound measurement of a refrigerator in reverberation room

Table 3.1 Characteristics of acoustic rooms used in this thesis

Characteristics	Semi-anechoic room	Reverberation room
Cut-off frequency (Hz)	100	100
Back-ground noise level (dB)	20	23

### 3.3 Sound Power Measurements

In practice, sound power levels of the machinery are determined using sound pressure measurements. One method used for refrigerator measurements in semi-anechoic chambers includes utilizing rectangular parallelepiped surfaces (ISO3745). As shown in Figure 3.3, a virtual rectangular prism is formed around the refrigerator by taking its size as reference. Refrigerator is placed before the panel to be used as reflecting panel inside the room by leaving 1 cm apart. Microphones are placed at six

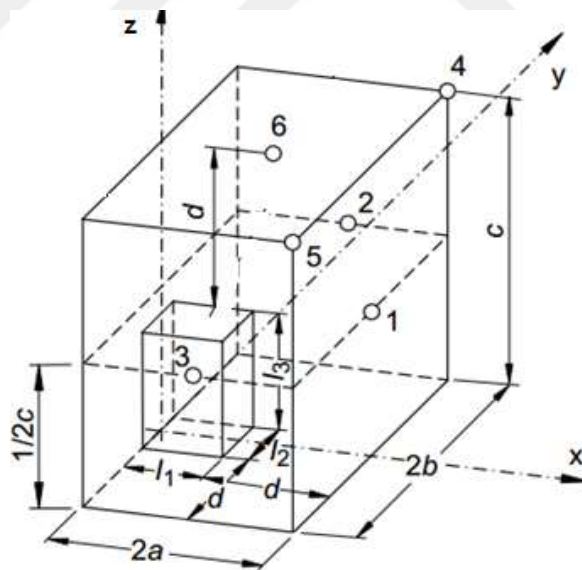


Figure 3.3 Microphone positions on rectangular parallelepiped surfaces

points of the prism and sound pressure measurements are performed. These measurements give the logarithmic values of the pressures as “sound pressure levels”. However, A-weighting scale is used in measurements related with white

goods since this scale reflects human aural capability in the best way. Logarithmic sound pressure value expressed in A-weighting scale is called as “sound level”.

Sound power level is calculated by using the average sound pressure level  $L_p$  and measurement surface area  $L_S$ .

$$L_W = L_P + L_S = 20 \log\left(\frac{p}{p_0}\right) + 10 \log\left(\frac{S}{S_0}\right) \text{ dB}(A) \quad (3.7)$$

The derivation of sound power level formulation is given in Table 3.2.

Preparation of refrigerators before measurement and measurement procedures have been realized in accordance with the International Organization for Standardization (ISO) 3741 (2010), ISO 3742 (1988), ISO 3744 (2010) and ISO 3745 (2012). In general, these standards are valid for standard refrigerators. In advance with technology, refrigerators have been highly developed. However, standards are not in conformable with this development yet. In this thesis, both big-size and standard no-frost refrigerators were handled using these standards. The determination of the defrost noise of the refrigerators has also no established standard.

Before the measurement, refrigerator should be operated at least 16 hours in order to reach sufficient regime. The temperature at the deep freezer is fixed at  $-22^\circ\text{C}$ , and for the cooling section at  $5^\circ\text{C}$ . Room temperature and humidity are kept at  $23 \pm 3^\circ\text{C}$ ,  $50\% \pm 20\%$  rH, respectively.

Table 3.2 The derivation of sound power level formulation

<p>Beranek, &amp; Ver (2006)</p>	<p>Average sound intensity on measurement surface <math>I</math> and sound power <math>W</math> are;</p> $I = \frac{p_{rms}^2}{\rho c}, \quad W = \sum_i I_i S_i$ <p>where,</p> <p><math>p_{rms}</math> = root mean square sound pressure  <math>\rho c</math> = characteristic resistance of air  <math>I_i</math> = sound intensity averaged over <math>i</math>th segment of area  <math>S_i</math> = <math>i</math>th segment of area</p> <p>Sound power and pressure relation is;</p> $W = \frac{1}{\rho c} \sum_i p_i^2 S_i$ <p>Sound power level is;</p> $L_W = 10 \log \sum_i S_i 10^{L_{pi}/10} - 10 \log K$ <p>where,</p> <p><math>L_{pi}</math> = sound pressure level over <math>i</math>th area segment  <math>10^{L_{pi}/10} = (p_i/p_{ref})^2</math>  <math>K = \rho c/400</math>; at normal temperatures and atmospheric pressures, this term is negligible. Therefore;</p> $L_W = 10 \log \sum_i S_i 10^{L_{pi}/10}$ $L_W = L_P + 10 \log\left(\frac{S}{S_0}\right)$ <p>where,</p> <p><math>L_P</math> = sound pressure level averaged on a mean square basis over measurement surface  <math>S</math> = area of measurement surface  <math>S_0 = 1</math> unit</p>
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## **CHAPTER FOUR**

### **OVERALL, COMPRESSOR AND FAN NOISE IN REFRIGERATORS**

#### **4.1 Introduction**

This chapter is focused on the overall noise and also compressor, fan noise in refrigerators. In this part of the study, the acoustic tests of a refrigerator have been performed, noise sources have been localized and recovery solutions have been applied. The refrigerator under test is a new design, high-capacity, six-door appliance.

In literature, generally vibration and noise studies on refrigerators have been focused on the principle noise sources, compressors and fans. Nevertheless, there are some other studies examining refrigerators as a system and they are being summarized below.

#### **4.2 Literature Survey**

Since compressor is a general noise source, the studies on the noise and vibration of compressors have been started before the researches on noise of refrigerators. Diehl (1972) has examined major noise sources of different types of compressors. These are turbulence and also interaction of rotating and stationary vanes in centrifugal compressors; discharge and inlet of axial compressors; and inertia forces in reciprocating compressors. Ryu et al. (1994) have examined noise sources of hermetic compressors used in refrigeration systems and designed a suction silencer to reduce base noise. Silveira (2004) has presented the contributions of noise sources in reciprocating compressors on the frequency spectra by a sensitivity study; and developed a new compressor design that reduces noise and vibration. Lee et al. (2005) have examined refrigerator noises through a series of experiments; and classified the noises on the basis of the operation steps of the loudest source compressor. Suh et al. (2006) have created different linear compressor designs by using analysis programs; and compared results of characteristics like noise and cooling capacity with trials. Park et al. (2008) have examined the low frequency

noise of the reciprocating compressor in a refrigerator; and compared the simulation results of the model that they have developed to reduce the noise, with the experiments. Seçgin et al. (2009) have performed an experimental modal analysis of a refrigerator compressor; determined modal parameters such as damping and natural frequency; and compared the results with the numerical analysis findings.

With regard to fan noise, Kim et al. (2006) have analyzed noise sources in refrigerators as cooling fan, compressor fan, and condenser fan. They have found out that the main source of noise in fans was the fluctuation in forces created by air flow passing through fan blades and beams; and developed a new axial fan with low level of noise. Lee et al. (2010) have analyzed the noise at the blade passing frequency of the household refrigerators' centrifugal fan in their experimental and numerical studies; and developed a design for reducing this noise. Heo et al. (2011) have reduced the noise at the centrifugal fan's blade passing frequency in refrigerators by modifying the blade profiles. Seo et al. (2012) have used a perforated panel system to reduce the noise of cooling fan of a household refrigerator without reducing the cooling performance capacity.

Some studies examine the overall noise of the refrigerators and determine the major noise sources. Lee et al. (2005) have analyzed the noise of a typical refrigerator with a set of trials; determined the frequency characteristics of sounds; found that the noise was mainly created by the compressor and fan used for air circulation; and then classified this noise according to the operating phases of the compressor. Lee et al. (2006) have recorded the noise of eight refrigerators in an anechoic-chamber and a real-life apartment; determined their common features and predicted perceptual noise levels; and then developed models of the refrigerator's noise. Jeon et al. (2007) have investigated noise radiation and sound quality characteristics of various types of refrigerators in a real-life apartment; and compared the findings with those obtained in an anechoic chamber. Şahin (2009) has analyzed no-frost refrigerators with evaporator fan as only noise source; supported experimental studies with numerical models; and determined critical regions by forming vibration and noise maps. Seçgin & Sarigül (2011) have made a study by

using boundary element solution of the Helmholtz integral equation and applied this solution to a refrigerator. Surface velocities measured on the refrigerator have been fed to in-house computer code; and surface acoustic pressures have been computed as well as the exterior pressures. In this way, sound source localization and determination of the refrigerator have been performed. Lee et al. (2013) have made numerical and experimental studies to investigate the impulsive noise due to opening and closing of the door of a large refrigerator; and suggested a solution to reduce this noise at the design stage. Akaydın (2013) has compared the vibration levels of refrigerators for their normal working conditions; analyzed the frequency components of vibrations; determined vibration mode-shapes and noise levels of the lower parts; compared sound pressure levels at the front and back parts; worked on the sound quality; and searched for a method to reduce vibration and noise levels.

The refrigerator on which measurements have been taken and noise recovery solutions have been applied is a new appliance. It is a high-capacity, superior refrigerator with a big size and many components. Therefore there is a substantial noise problem to be solved. This type of refrigerators is not studied in the literature yet. The study was initiated with the determination of sound power of the refrigerator. Then, sound intensity tests were carried out in order to localise the noise sources. After, to eliminate the noise, some recovery applications were performed on the same appliance.

The high-capacity refrigerator handled in this chapter has a total volume of 820 Lt with dimensions 2120 mm x 1210 mm x 710 mm. The refrigerator system is no frost and refrigerant type is R600a. There are 2 compressors, 2 condenser fans and 3 evaporator fans on the refrigerator. Compressor compartment and four of the fans are mounted at the top of the appliance. In these refrigerators, variable speed compressors are used for energy saving. All acoustic tests of this refrigerator have been performed in the semi-anechoic acoustic chamber defined in Table 3.1. The refrigerator and test room is shown in Figure 3.1.

### 4.3 Sound Power Measurements of the Refrigerator

In order to obtain sound power levels, sound pressure level measurements are made and the computations given in Chapter 3 are applied. These computations may be performed by some instruments automatically. In this study, sound power levels have been computed directly by using Pulse program of Brüel & Kjaer instruments.

Since the compressors are of variable speed, their activation and deactivation periods vary. During sound measurements, the cooling period of the refrigerator has been determined. One period is shown in Figure 4.1. Then, A-weighted highest sound power level ( $L_{Wmax}$ ) has been recorded for the activation periods of two compressors together. Also, A-weighted equivalent sound power level ( $L_{Weq}$ ) has been measured for the total cooling period. The equipment used in the measurements is given in Table A.1. The measurements have been performed many times to satisfy repeatability.

To be conformable with the standards, atmospheric conditions of the room have been also recorded throughout the measurements and presented in sound power tables together with the period of the measurements. Peak sound power levels and equivalent sound power levels are presented in Tables 4.1 and 4.2 respectively. It is seen that small variations in room temperature and humidity do not cause to a meaningful change in the sound power of the refrigerator. Although this kind of variations may be noticeable for small-size refrigerators, they did not influence the sound power of the big-size refrigerator. Peak sound power values in Table 4.1 have been measured during two compressors and two condenser fans were working simultaneously. This means that the noise which comes out as disturbing to the user and which needs to be controlled, occurs during four components are active. Peak level is almost 4 dB(A) higher than the equivalent level and should be attenuated. Tolerance of these measurements are  $\pm 0.1$  dB(A) for peak levels and  $\pm 0.6$  dB(A) for equivalent levels. As expected, tolerance of equivalent levels are higher than that the peak levels.

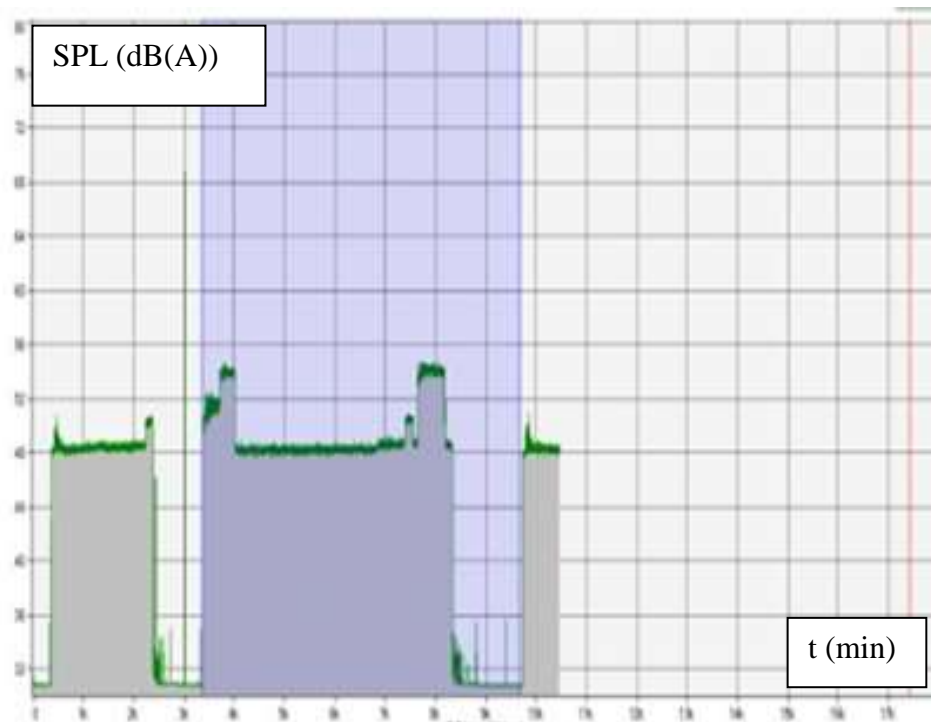


Figure 4.1 One period of a cooling cycle

Table 4.1 A-weighted peak sound power levels ( $L_{Wmax}$ ) of the refrigerator ( $\pm 0.1$  dB(A))

	dB(A)	duration (two compressors active)	temperature °C	humidity % rH
test1	54	6.45min	25-25.8	38-38.8
test2	54	8.20min	26-26.3	37-38.2
test3	54	8.14min	26-26.1	35-36
test4	54	8.20min	27-28.3	37-38.2
test5	54.1	7.33min	26-26	38-38
test6	54.1	8.28min	26-29.3	38-38.7
test7	54.1	8.35min	25.8-26.3	37-38.7
test8	54.1	8.31min	26-27.3	38.5-38.7
test9	54.2	8.27min	26-26.3	38-38.8
test10	54.1	8.16min	27.1-27.8	38-38.5

Table 4.2 A-weighted equivalent sound power levels ( $L_{Weq}$ ) of the refrigerator ( $\pm 0.6$  dB(A))

	dB(A)	one cooling period	temperature °C	humidity % rH
test1	50.3	1.29 hour	26-28	35-38.4
test2	49.9	1.9 hour	27-29	36-38.7
test3	49.8	1.48hour	26-29.1	37-39.1
test4	49.6	1.34 hour	28-29.2	36-37.7
test5	49.1	1.44 hour	26-29.2	35-38.2
test6	49.6	1.36 hour	28-29.2	36-37.7
test7	50	1.59 hour	25.1-28.2	36-36.7
test8	49.8	1.44 hour	27.8-28.5	35.8-36.9

#### 4.4 Determination of Noise Sources in the Refrigerator

A refrigerator can be seen as a small system and in this system there are different noise sources as compressor, condenser fan and flow of cooling fluid. In a system including many noise sources, the total noise level increases with the number of sources. For this reason, choosing and assembling these components are important issues; and the fact that this process must be done by protecting the energy efficiency of the refrigerator puts forward the importance of the subject.

In order to determine the main noise sources of the refrigerator and to control them, noise mapping study has been performed. For this reason, sound intensity measurements have been made by using the sound analyzer and microphone probe shown in Figure 4.2. Sound pressure level measurements have been performed on five surfaces (front, top, back, right, and left) of the top region of the refrigerator where the motor group is located. For this purpose, the surfaces have been divided into square elements and measurements have been carried out by placing the probe across the middle points of the elements, in turn. The probe has been positioned at a distance of 10 cm from the source.

As a result of measurements, a map has been formed showing the distribution of sound on the surface area considered (Figure 4.3). On the noise maps, regions with light colors correspond to places where the sound sources are present. Among the five maps the clearest one is that for the top surface shown in Figure 4.4. Examining

this figure, the condenser fan may be localized as the main noise source of the refrigerator. Besides, the noise distribution around the cooling compressor may be detected. Figure 4.5 for the back surface shows the significant amount of noise around the air suction hole. As the back surface is connected to the top surface, a high level of noise is radiated around the hole by the influence of the condenser fan. 1-octave frequency spectra of these measurements show that noise is effective mainly at 250 Hz and 400 Hz center frequencies.



Figure 4.2 Sound analyzer and intensity probe including microphone

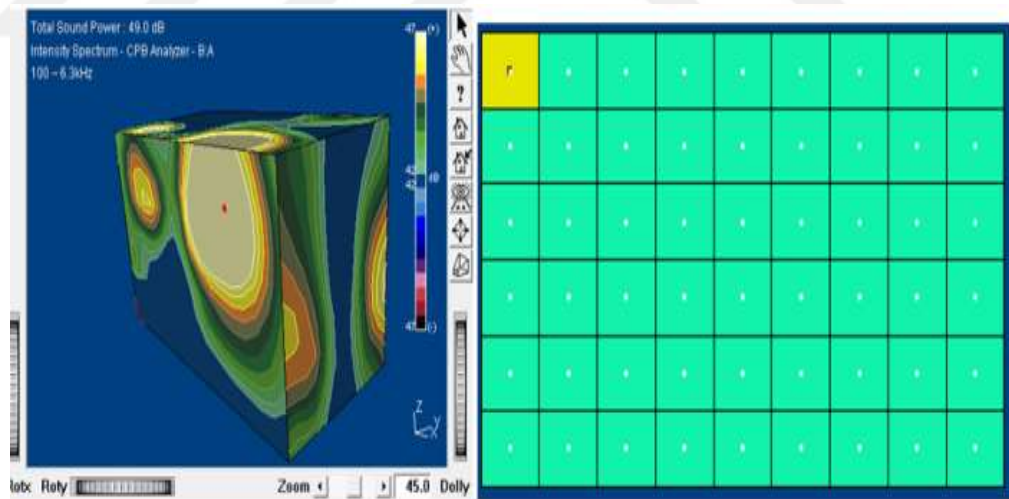


Figure 4.3 Noise maps, surface elements and measurement locations of surfaces

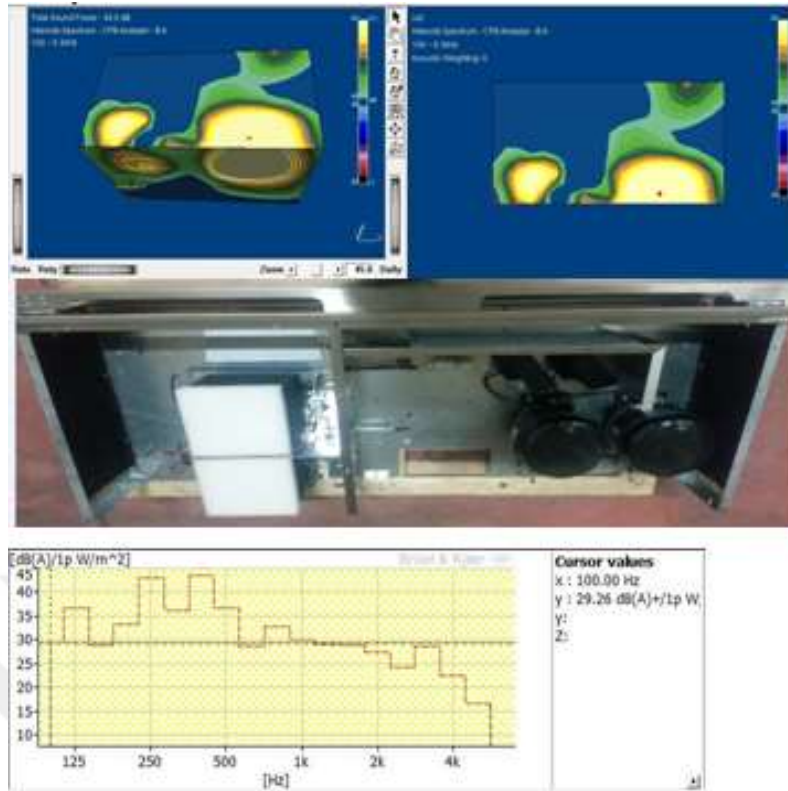


Figure 4.4 Noise map and frequency analysis of the top surface

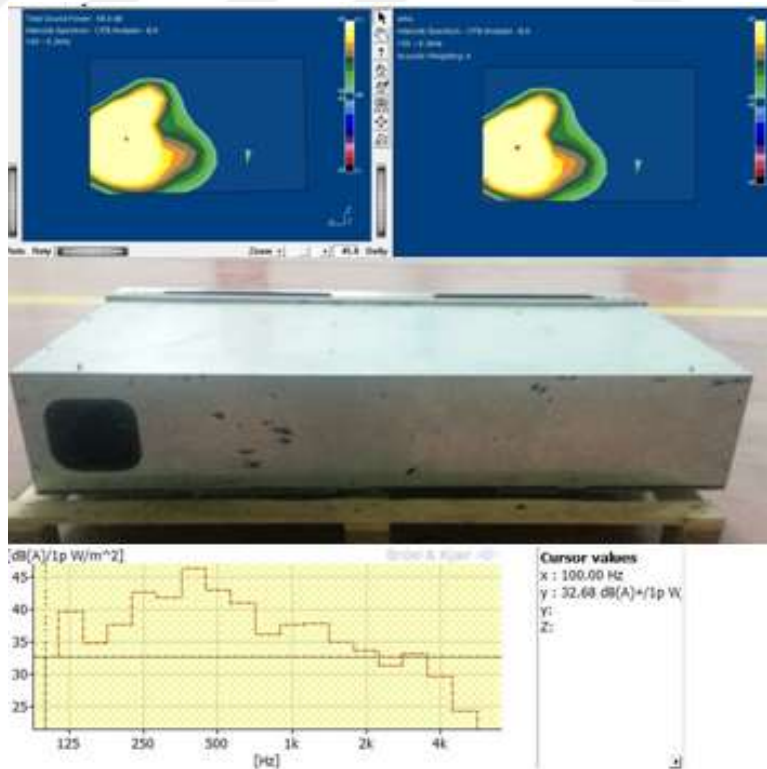


Figure 4.5 Noise map and frequency analysis of the back surface



#### 4.5 Sound Power Measurements of Compressors

As compressors are one of the basic sound sources of refrigerators, the freezer and fridge compressors of the refrigerator have been operated separately and their sound power levels have been measured in 1/3-octave band frequencies. During the measurements, the compressors have been operated with maximum efficiency.

Table 4.3 Frequency analysis of A-weighted sound power levels of compressors

Hz	Freezer compressor [dB(A)]	Fridge compressor [dB(A)]
100	15.3	15.6
125	25.2	24.7
160	20.6	22.2
200	25.2	18.2
250	26.5	24.3
315	33.6	32.5
400	30.9	46.5
500	29.5	42.2
630	23.1	33.2
800	21.7	26.4
1000	19.1	22.5
1250	19.6	21,6
1600	21.4	21.7
2000	22.7	21.4
2500	23.9	25
3150	30	23.8
4000	25.5	22.6
5000	23.9	22.4
6300	24.2	22.5
8000	23.5	21
10000	19.6	19.3
total	39.5	48.4

Fridge compressor's working speed is 1600 rpm and freezer compressor's speed is 1400 rpm during the activation period. Frequency analysis results of the freezer and fridge compressors are presented in Table 4.3. These measurements were taken for a short period of time that the air around the compressors' not to be affected. It is seen that the freezer compressor is not an important noise source. The noise of the fridge compressor is effective between 400 Hz-500 Hz frequencies and this may contribute to the total noise created by the refrigerator.

#### **4.6 Control of Noise in the Refrigerator**

It was tried to reduce the noise of the refrigerator using two different approaches. One was using insulation material and the other was changing the condenser fan.

##### **– Stage 1**

It has been planned to use the insulation material on the top compartment. For this procedure two different materials have been applied on the yellow regions in Figure 4.4. These regions are under the effect of high noise intensity.

In the first step, phenolic felt which is used commonly in white goods industry has been placed. Since this application was made on the basis of the noise maps, phenolic felt has been placed only at necessary positions. Therefore the waste of the material has been prevented.

In the second step, a specific insulation material was developed as a solution for this problem. This material shown in Figure 4.6 was prepared using phenolic felt together with sawdust that is another sound insulation material. Thereby, a sandwich structure was formed. While felt is a material that can absorb sound more at high frequencies, the wooden materials show good absorption capacity at low frequencies as well (Erol & Öztürk, 2009; Altunok & Ayan, 2012). 15 mm thickness of felt increased to 18 mm after the sawdust addition. This structure satisfied the necessary requirements for the insulation (against air-borne noise and impact noise) and sufficiently rigid (Figure 4.6 and 4.7).

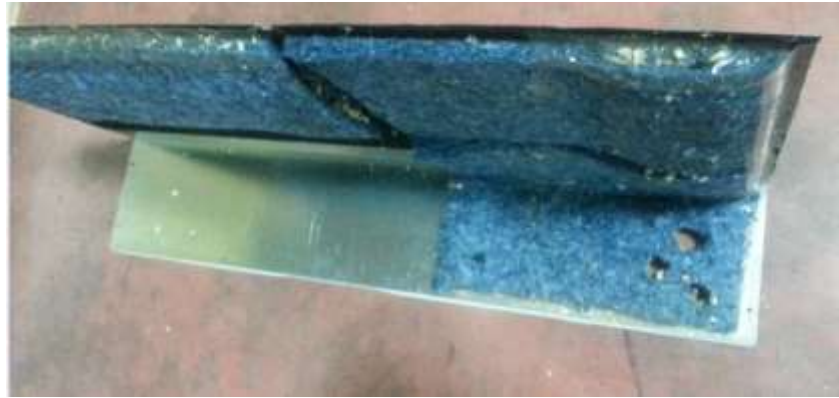


Figure 4.6 Sandwich structure with phenolic felt and sawdust

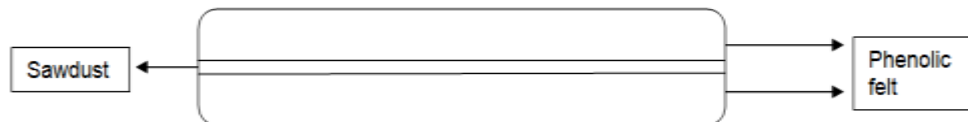


Figure 4.7 New sound insulation material (Using felting + sawdust mixing)

The results obtained from the noise reduction studies made for reducing noise are presented and compared with no insulation case in Tables 4.4 and 4.5. In Figure 4.8, the results obtained are presented in the form of frequency spectra for a general view. Also, comparisons on the basis of the total sound power level show the priority of felt + sawdust insulation (Table 4.6). Besides, according to frequency analysis, this material is effective in regions of low frequencies between 100-125 Hz and high frequencies between 1000-8000 Hz.

Table 4.4 Explanation of  $L_{W\ meas1}$ ,  $L_{W\ meas2}$  and  $L_{W\ meas3}$

$L_{W\ meas1}$ [dB(A)]	No insulation
$L_{W\ meas2}$ [dB(A)]	Insulation with felt
$L_{W\ meas3}$ [dB(A)]	Insulation with felt + sawdust

Table 4.5 Frequency analysis of A-weighted sound power levels of  $L_{W\ meas1}$ ,  $L_{W\ meas2}$  and  $L_{W\ meas3}$  ( $\pm 0.1$  dB(A))

Hz	$L_{W\ meas1}$ [dB(A)]	$L_{W\ meas2}$ [dB(A)]	$L_{W\ meas3}$ [dB(A)]
100	37.7	36.1	35.6
125	43.4	41.8	39.1
160	39.1	39	39.9
200	43.1	41.6	42
250	47.9	47.5	46.3
315	43.3	43.6	43.5
400	47.3	46.6	45.5
500	45.4	44.7	44
630	40.8	39.6	39.6
800	37.4	34.5	34.4
1000	36.3	31.7	30.8
1250	34.3	28.5	27.9
1600	31.8	27	26.2
2000	30.4	26	24.6
2500	30.4	25.4	23.8
3150	34.5	32	23.3
4000	29.5	26.5	22.8
5000	25.5	23.9	22.4
6300	26.6	25	22.3
8000	25.1	23.5	20.9
10000	19.5	19.4	19.7
Total	54.1	53.5	52.8

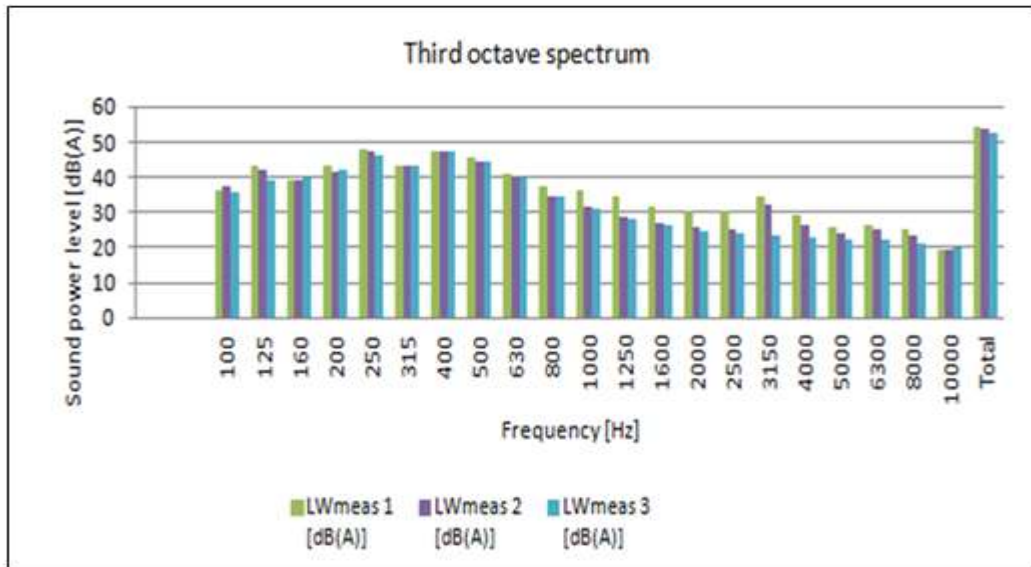


Figure 4.8 Comparisons of sound spectra for the refrigerator

Table 4.6 Sound reduction after the insulation ( $\pm 0.1$  dB(A))

Test	Sound reduction [dB(A)]
$L_{W\ meas2}$	0.6
$L_{W\ meas3}$	1.3

– Stage 2

As another factor for improvement, condenser fan has been analyzed as it was found to be an important source of noise. This fan had five blades (Figure 4.9.a). In order to compare sound power of this fan with the other three types separate measurements have been taken with fan with four blades (Figure 4.9.b), plastic horizontal cylindrical fan and horizontal cylindrical fan. As can be seen from the results presented in Table 4.7, horizontal cylindrical fan placed in front of the condenser shows the best acoustic performance. Horizontal cylindrical fan and its assembly are shown in Figure 4.10.

Table 4.7 A-weighted sound power levels of different fans ( $\pm 0.1$  dB(A))

Type of Fan	$L_w$ [dB(A)]
Fan with five blades (on the refrigerator)	51.4
Plastic horizontal cylindrical fan	54.2
Horizontal cylindrical fan	36.9
Fan with four blades	39.7



(a) (b)  
Figure 4.9 Fans with (a) five and (b) four blades



Figure 4.10 Horizontal cylindrical fan

As a result of these tests, it has been found out that condenser fan with five blades needed to be changed with horizontal cylindrical fan as it was determined to be the main source influencing acoustic performance of the refrigerator negatively. But for this change to be made, it is not enough to have a good acoustic performance. In

order to protect the energy level of the refrigerator, its cooling performance also needs to be good. For this analysis, flow rate measurements and of the cylindrical fan have been made and compared with each other. In flow rate measurements, data has been collected from six different locations of the fan propellers and this process has been repeated five times. As can be seen from the results given in Table 4.8, the flow rate provided by the cylindrical fan is much higher than the flow rate of the fan on the refrigerator. As a result, cylindrical fan has been found to be much better in both acoustics and energy criteria. Therefore condenser fans used in high-capacity refrigerators have been replaced with cylindrical fans. The use of cylindrical fan in refrigerators is a new application according to our current knowledge.

Table 4.8 Comparison of the fans' flow rates

Location of flow meter	Flow rate (lt/min)									
	Fan with five blades (on the refrigerator)					Horizontal cylindrical fan				
	Repetition number					Repetition number				
	1	2	3	4	5	1	2	3	4	5
1	487	449	463	479	479	490	458	463	474	505
2	75	48	63	67	81	104	89	69	82	73
3	322	301	372	348	339	344	352	362	349	352
4	70	79	72	67	76	74	72	77	73	75
5	72	74	65	67	73	68	70	71	70	74
6	160	149	168	137	159	147	151	153	150	175

#### 4.7 Conclusion

Overall sound power level is the main acoustic characteristic of machines but it is not enough for noise reduction studies. For the noise to be taken under control, sources of sound in the machine need to be localized. Noise mapping test based on sound power measurement is an effective method used for this purpose. In this

chapter, the results of sound power level measurements of high-capacity refrigerators were presented and noise maps were given. As one of the findings from the study in this chapter, it was seen that using sawdust inside commercial felt created an effective sound insulation material in refrigerators. Another finding was that horizontal fans used in high-capacity refrigerators were the main sources of noise and that they needed to be changed with horizontal cylindrical fans which were more appropriate both in acoustical and energy aspects.





## **CHAPTER FIVE**

### **DEFROST NOISE IN REFRIGERATORS**

#### **5.1 Introduction**

Defrosting or defrost system is the process of removing ice on evaporator surface. This process controls the refrigeration system by preventing formation of frost on evaporator surface.

The defrost system is used in all of the no-frost refrigeration systems. Ice or frost accumulates on the evaporator surface when the evaporator surface is lower than 0°C and surrounded by humid air. This accumulation leads to performance loss of the evaporator and therefore, the loss of the precious cooling effect and reduction of performance coefficient of the refrigerator. Therefore the defrost system gets benefit to control the appliance.

The so-called “defrost noise”, “cracking noise” or “crack noise” occurs at the defrost process of a refrigerator. Cracking noise generated during the automatic defrost cycle is one of the most important quality issues for no-frost refrigerator developers. However, there is no international standard obligating manufacturers to suppress or limit the level of cracking noise. On the other hand, no previous study has been published up to now regarding on this type of noise in the literature.

In this study, the cracking noise level of a standard no-frost refrigerator shown in Figure 3.2 has been measured. The acoustic tests have been repeated for ten appliances in order to obtain reliable results. Different improvements have been carried out and the results were presented in terms of acoustic behaviour at the defrost process.

#### **5.2 Defrost Principle in a Refrigerator**

In the defrost process, since the compressor is not working, the evaporator coils warm in a short period of time. At this instant, ice blocks on the evaporator drop into the evaporator bowl. On the other hand, due to this temperature change, large plastic

surfaces of the refrigerator and also food containers begin to dilate. These two phenomena create an impulsive type of noise.

Defrosting is controlled by an electric or electronic timer and its time period varies according to the model of the appliance. To speed up the ice thawing process a defrost heater is mounted around the evaporator of all no-frost models, such as in top-freezer, bottom-freezer and also side-by-side models. For different periods of compressor operation, defrost heater has different operation periods, e.g. 6, 8, 10 and 14 minutes. Therefore defrost period is considered as the operation period of the heater. Defrost period of the present refrigerator is 14 minutes. A defrost thermostat opens the heater circuit when the evaporator temperature rises above a preset temperature, 40°F (5°C) or more, thereby preventing extreme heating of the freezer compartment. The defrost process is set up such that either the compressor or the defrost heater is on, but not both at the same time. Although in former refrigerators, compressors have been running continuously; in newer designs, due to heat and energy considerations, this is not done.

### **5.3 Noise Status During Defrost**

It is known that the major root cause of the defrost cracklings is the thermal expansion of the plastic components around the evaporator area. Also, de-icing that is falling of ice into evaporator bowl is another reason of defrost noise. There are many factors affecting the defrost noise, such as evaporator surface area, plastic surface area and power of the heater.

For defrost measurements, the appliances that have high defrost noise have been operated through 72 hours before the tests. Because, the refrigerators are set to start the defrost operation for each 72 hours. The measurements have been performed at the reverberation chamber defined in Table 3.1.

Since there is no standard for the defrost noise, a measurement criteria have been described for the present measurements. Cracking noise is an impulsive noise therefore it is necessary to measure its peak level together with the equivalent level

in 14 minutes. Another essential data for the defrost noise is the frequency of the peaks, which means how often the user might hear the defrost cracks. Therefore, the defrost noise has been assessed considering the explanations shown in Table 5.1.

Table 5.1 Explanations of measurement values

	Explanations
$L_{Weq}dB(A)$	Equivalent sound power level of the cracks in the defrost interval
$L_{Wmax}dB(A)$	Maximum sound power level of all the cracks measured
NC	Number of cracks ( $L_W > 40 dB(A)$ ) measured

$L_{Weq} dB(A)$  value is calculated as,

$$L_{Weq} = 10 \log \left[ \frac{1}{T} \sum_{i=1}^{i=n} \frac{W_i}{W_{ref}} \right] dB(A) \quad (5.1)$$

where,  $T$  is the defrost period and  $W_i$  is the measured A-weighted sound power level for each cracking noise.

The data for the original state of the appliances have been recorded as reference tests and presented in Table 5.2. The results of reference test for appliance 1 is given in Figure 5.1. Table 5.2 shows that, throughout the ten measurements with different appliances of the same type, maximum sound power levels of the defrost process are always higher than 50 dB(A). Cracks occur 8 to 12 times for each defrost cycle. According to these results, the refrigerators have a defrost noise as crack noise and it is absolutely disturbing both in type and level for human hear.

Table 5.2 Reference values for the defrost noise

Appliance Number	$L_{Weq}dB(A)$	$L_{Wmax}dB(A)$	Number of Cracks
1	32.9	58	10
2	32.9	58	8
3	29.7	52	10
4	32.2	55	11
5	29.8	52	10
6	30.5	53	11
7	31.3	56	11
8	29.5	51	10
9	32.9	56	11
10	32.4	56	12

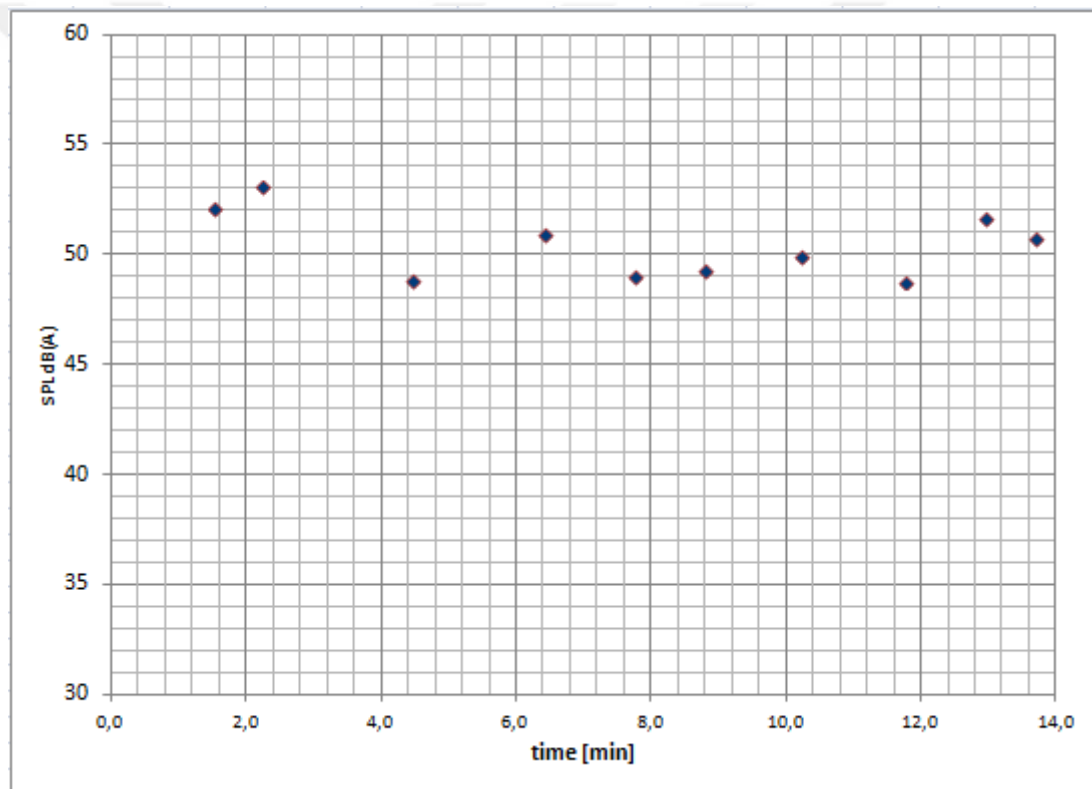


Figure 5.1 Reference test for the appliance number 1

#### 5.4 Modifications for Defrost Noise

In this part of study, in order to decrease the defrost noise with regard to number of cracks, their maximum and equivalent values; some experimental studies were carried out on these ten appliances. These experimental approaches are explained in sequence.

#### 5.4.1 Changing the Material of the Freezer Back Plastic (FBP)

FBP material could have an influence on defrost noise. Originally, FBP parts were being made of Styrene Butadiene (SB) material and the reference tests were conducted with this material. As an alternative, another plastic material (PPTV) as FBP material was used since this is slightly softer than the previous one. Eventually, no improvement was seen with PPTV as shown in Table 5.3.

Table 5.3 Effect of the material of FBP on the defrost noise

Appliance Number	$L_{Weq} dB(A)$		$L_{Wmax} dB(A)$		NC	
	reference test	with PPTV material	reference test	with PPTV material	reference test	with PPTV material
1	32.9	33.6	58	59	10	10
2	32.9	33.4	58	58	8	8
3	29.7	30.1	52	52	10	10
4	32.2	32.4	55	55	11	11
5	29.8	30.5	52	52	10	10
6	30.5	31.7	53	54	11	11
7	31.3	32.6	56	56	11	11
8	29.5	30.5	51	51	10	10
9	32.9	33.3	56	56	11	11
10	32.4	32.9	56	56	12	12

#### 5.4.2 Reducing the Freezer Back Plastic (FBP)'s Screw Diameters

The freezer back plastic part is mounted to the cabinet by two screws as seen in Figure 5.2. Reducing the contact between the cabinet and the FBP might suppress the thermal stresses causing cracklings during cooling down and heating up processes of the compartment. For this reason, dome diameters were reduced from 10 mm to 9 mm. However, no noise improvement was observed after this modification as shown in Table 5.4.

Table 5.4 Effect of the dome diameters of FBP on the defrost noise

Appliance Number	$L_{W_{eq}}dB(A)$		$L_{W_{max}}dB(A)$		NC	
	reference test	With 9 mm	reference test	With 9 mm	reference test	With 9 mm
1	32.9	33.4	58	58	10	10
2	32.9	33.4	58	58	8	8
3	29.7	30.5	52	52	10	11
4	32.2	33.4	55	55	11	11
5	29.8	30.5	52	52	10	10
6	30.5	31.4	53	53	11	11
7	31.3	31.4	56	56	11	11
8	29.5	30.2	51	51	10	10
9	32.9	32.9	56	56	11	12
10	32.4	33.1	56	56	12	12

### 5.4.3 Removing the Freezer Back Plastic (FBP)

It was supposed that, one of the sources generating the cracking noise was the FBP. Because, this part has a plastic surface and it is affected by a rapid change of temperature directly. The FBP is mounted to the cabinet by two screws as seen in Figure 5.2. The user faces with the front part of the FBP, called freezer compartment area. The back side of FBP corresponds to the evaporator area. Therefore, a rapid change of temperature in the evaporator is significant on FBP.

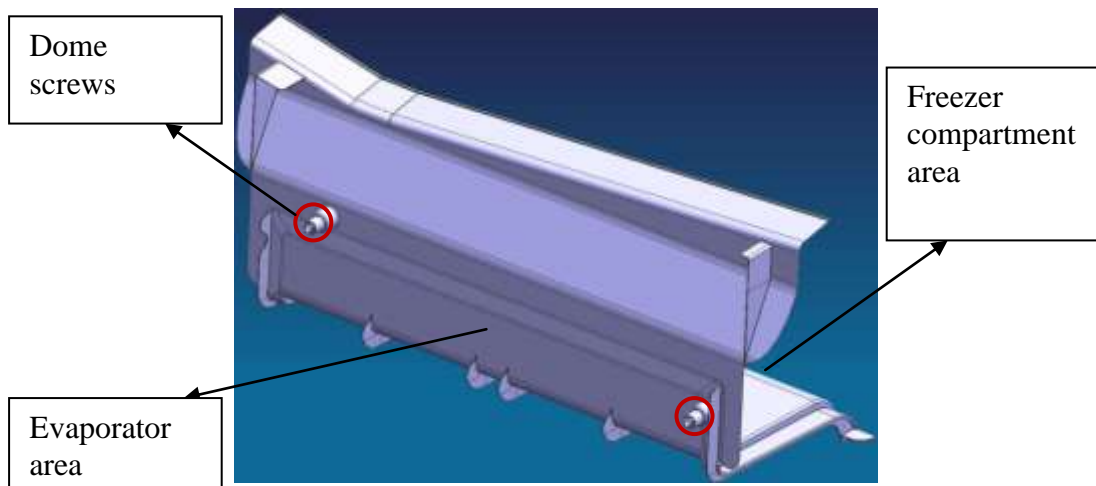


Figure 5.2 Freezer back plastic assembly

Table 5.5 Effect of the presence of FBP on the defrost noise.

Appliance Number	$L_{Weq}dB(A)$		$L_{Wmax}dB(A)$		NC	
	reference test	without FBP	reference test	without FBP	reference test	without FBP
1	32.9	33.1	58	58	10	11
2	32.9	32.7	58	58	8	8
3	29.7	30.4	52	52	10	10
4	32.2	32.2	55	55	11	11
5	29.8	30	52	52	10	10
6	30.5	29.8	53	53	11	12
7	31.3	31.4	56	56	11	11
8	29.5	30.5	51	51	10	10
9	32.9	32.9	56	56	11	11
10	32.4	32.8	56	56	12	12

The tests were conducted to verify this claim by removing FBP from the appliances. Results are compared with the reference tests in Table 5.5.

There is no meaningful change in  $L_{Weq}$  levels after removal of FBP.  $L_{Wmax}$  values for two cases are the same. Number of cracks increased for appliance numbers 1 and 6. Therefore, no improvement was recorded by this implementation.

#### 5.4.4 Modifying the Defrost Parameters

Some modifications were made in the defrost parameters as shown in Table 5.6. First, in order to dissolve the ice from the evaporator surface rapidly, during the defrost time, the electrical power level was increased. Also, the standard defrost time was lowered. Then, the defrost end temperature measured on the evaporator inlet surface by using a thermocouple were lowered. The influence of these modifications on defrost noise is presented in Table 5.7.

Table 5.6 Defrost parameters

	initial parameters	modified parameters
electrical power	270 Watt	320 Watt
standard defrost time	14 min	11 min
defrost end temperature	12 °C	10 °C

Table 5.7 Influence of the modified parameters on the defrost noise

Appliance Number	$L_{Weq} dB(A)$		$L_{Wmax} dB(A)$		NC	
	reference test	modified parameters	reference test	modified parameters	reference test	modified parameters
1	32.9	31.3	58	56	10	8
2	32.9	31.0	58	56	8	8
3	29.7	29.5	52	51	10	8
4	32.2	31.4	55	53	11	10
5	29.8	29.1	52	50	10	9
6	30.5	29.6	53	52	11	9
7	31.3	30.1	56	54	11	9
8	29.5	29.1	51	49	10	9
9	32.9	32.0	56	53	11	10
10	32.4	31.5	56	53	12	10

According to the test results which were carried out with modified parameters, the  $L_{Weq}$  level was reduced between 0.2 dB(A)-1.9 dB(A) whereas  $L_{Wmax}$  was reduced between 1 dB(A)-3 dB(A). Moreover, the number of cracks was measured between 8 to 10 times after the modification. Obviously, there is an improvement due to the new defrost parameters that avoids formation of the peaks.

## 5.5 Conclusion

Maximum values of defrost noise have considerably high levels. Since defrost time is 14 minutes, the computed equivalent levels of defrost noise are not so much. This noise is significant, disturbing and unwanted for it is impact type. Especially, the users suffer from the crack noise during the night.

Although it is a significant type, there is no published research and standard on the defrost noise. Standards might be helpful in order to determine the boundary conditions and restrictions of the measurements. These conditions were determined here by using the measurement experience. Maximum, equivalent values and number of cracklings with sound power levels higher than 40 dB(A) were considered.

Four different attempts were performed to attenuate the defrost noise: removal, material change and reduction of the screw diameters of freezer back plastic part and also modification of the defrost parameters. The last alternative that includes



increasing the electric power and reducing the defrost time was resulted with gain in acoustic performance.

The defrost noise is still a potential problem required to be studied and solved in the future for the no-frost technology.



## **CHAPTER SIX**

### **FLOW NOISE IN REFRIGERATORS**

#### **6.1 Introduction**

Vibration and noise problems due to fluid flow occur in many refrigerators. The flow-related vibrations are known as “Flow-Induced Vibrations”. In the case of piping connected to reciprocating fluid machines, the oscillating (fluctuating) flow in the piping generates excitation forces causing piping vibration (Kaneko et al., 2008). “Flow-Induced Vibrations” cause to “Flow Noise”, known as “Fluctuating Noise”. In a refrigerator, refrigerant-induced noise is a combination of fluid-acoustic and vibro-acoustic phenomena. Turbulence, compressor-induced pulsations, phase change and throttling are the main sources of acoustic excitation in the refrigerant flow (Neto et al., 2014).

#### **6.2 Flow-Induced Vibrations**

Kaneko et al. (2008) have presented a classification based on the flow type as given in Table 6.1. Flow-induced vibrations may arise due to steady, unsteady and two-phase flows. For steady flow, the interaction between fluid and structure causing to increasing vibration amplitudes such as resonance is the most commonly observed phenomenon. For unsteady flow, turbulent forces constitute dominant source of structural vibration excitation. Two-phase flow is a mixture of two fluids with different densities. Therefore, the variations of flow momentum and pressure in time are sources of excitation for the structural vibration.

Some definitions and explanations about vibration mechanisms in Table 6.1 are given below (Kaneko et al., 2008):

Table 6.1 Classification of flow induced vibrations (Kaneko et al., 2008)

Fluid and flow	Flow field	Vibration mechanism	Example
<b>Single-phase flow</b>			
<b>Steady flow</b>	External flow	vortex induced vibration	resonant vibration
			forced vibration
		acoustic resonance	acoustic resonance
			cavitation
	fluidelastic vibration	wing flutter and galloping	
		Fluidelastic vibration of tube arrays	
Internal flow	Surging	compressor surge	
	vibration of piping	pump surging piping, bellows, collapsible tubes	
<b>Unsteady flow</b>	Turbulent flow	random vibration	buffeting
			vibration of reactor internals
	Sudden change in flow	pressure pulsation	valve vibration
			water hammer
Pulsating flow	forced vibration	vibration due to internal fluid oscillations	
	acoustically induced vibration	combustion - induced vibration	
<b>Two-phase flow</b>	Bubble-induced vibration		sloshing
	Thermal-hydraulic vibration with phase change		vibration caused by condensation
			Instability caused by boiling
Vibration of piping by two-phase flow			

### 6.2.1 Vortex – Induced Vibration (VIV)

VIV is the motion induced on bluff bodies interacting with an external flow. Resonance may occur when the vortex shedding frequency of the oscillating flow becomes close to one of the structural frequencies of the body.

### **6.2.2 Acoustic Resonance**

Acoustic resonance may occur in heat exchangers and boilers with tube bundles when a critical level of the flow rate in the duct is exceeded. A high level of noise is produced, in extreme cases preventing plant operation and causing structural damage.

### **6.2.3 Fluidelastic Vibration**

Fluidelastic vibration, also known as self-induced oscillations, is observed for multiple cylinders. When the flow velocity is high, cylinders oscillate with large amplitudes. This is the most dangerous vibration and the main cause of many problems in industry.

### **6.2.4 Surging**

Surging may occur in a pipeline if a tank or an accumulator is located downstream of the pump and a resistive component, such as a flow-regulating valve, is located further on downstream. When the fluid contains many air bubbles, the bubbles can play the same role as installed accumulators.

### **6.2.5 Vibration of Piping**

Oscillating and two-phase fluid flows are causes of pipe vibrations. Gas-liquid, two-phase flow producing vibration may have different flow patterns shown in Figure 6.1 and explained as following:

- Bubbly flow: In this situation, the flow – induced forces are small and the gas flow rate is low. Therefore, bubbles occur in main liquid flow in the pipe as shown in Figure 6.1.a. A unit bubble can be modeled as a single degree of freedom system (Strasberg, 1956; Minnaert, 1933). The equation of motion and natural frequency of a bubble are given in Table 6.2.

- Stratified flow and wavy flow: In the case of horizontal piping liquid collocates at the lower side and gas flows at the upper side. If the interfacial surface is smooth, the flow is called a stratified flow (Figure 6.1.b). If interfacial surface is wavy, the flow is called a wavy flow (Figure 6.1.c).
  
- Plug and slug flow: If the equivalent radius of the bubble is larger than the radius of the pipe the spherical shape of the bubble deforms (Figure 6.2). In this case bubbles may be in bullet like shape. This flow is called plug flow (Figure 6.1.d). If large bubbles and liquid slugs flow alternately, this flow is known as slug flow (Figure 6.1.e).
  
- Froth flow: In this case, liquid slugs contain many bubbles but their shapes are not uniform. This flow is highly turbulent so flow-induced forces are large (Figure 6.1.f).
  
- Annular mist flow: This flow consists of an unstable liquid flow at the bottom and gas flow at the center of the piping. The turbulence in the liquid flow may cause to pipe vibrations (Figure 6.1.g).
  
- Mist flow: The main flow is the gas flow and liquid droplets flow like mist. This flow is the opposite of the bubbly flow (Figure 6.1.h).

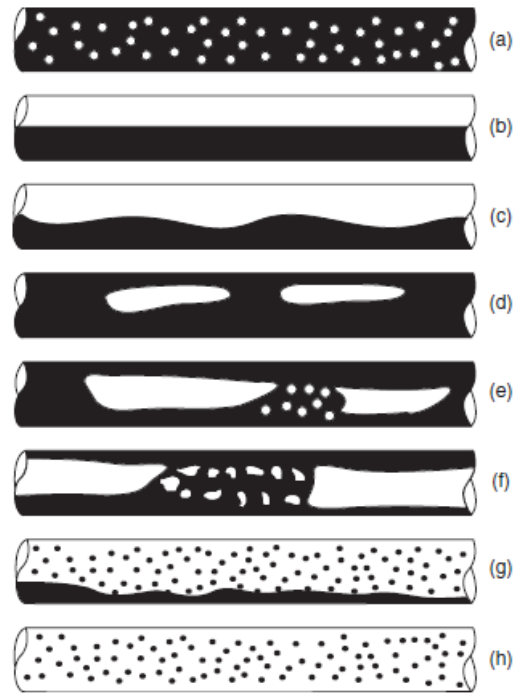


Figure 6.1 Flow patterns in horizontal pipes: (a) bubbly flow, (b) stratified flow, (c) wavy flow, (d) plug flow, (e) slug flow, (f) froth flow, (g) annular mist flow and (h) mist flow (Kaneko et al., 2008)

Table 6.2 Vibration characteristics of bubble

Strasberg (1956)	$m\ddot{v} + b\dot{v} + k(v - V_0) = P_A e^{j\omega t}$ <p>where,</p> $m = \frac{\rho}{4\pi R_0}, k = \frac{\gamma p}{V_0} \text{ and } v = V_0 + \alpha \sin \frac{2\pi t}{T}$ <p><math>m</math>: equivalent mass of the bubble  <math>v</math>: volume of the bubble  <math>b</math>: radiation resistance of the bubble  <math>k</math>: equivalent stiffness of the bubble  <math>V_0</math>: initial volume of the bubble  <math>R_0</math>: equivalent radius of the bubble  <math>P_A</math>: acoustic pressure of the bubble  <math>\omega</math>: radius frequency  <math>t</math>: time  <math>\rho</math>: density of the liquid surrounding the bubble  <math>\gamma</math>: specific heat ratio  <math>p</math>: pressure of the liquid surrounding the bubble  <math>\alpha</math>: amplitude of oscillation  <math>T</math>: period of oscillation</p>
Minnaert (1933)	$f_n = \frac{1}{2\pi R_0} \sqrt{\frac{3kp}{\rho}}$ <p><math>f_n</math>: natural frequency of the oscillating bubble</p>

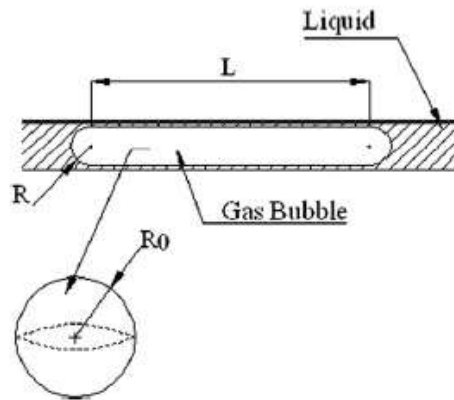


Figure 6.2 Deformation of the bubble (Han et al., 2010)

### **6.2.6 Forced Vibration**

As self-excited vibration, forced vibration is an important cause of pressure pulsations. Pulsations damage to machines and plants directly and may cause secondary damage to surroundings.

### **6.2.7 Acoustically Induced Vibration**

Thermal processes such as combustion, steam condensation and boiling are causes of acoustic vibration and noise.

### **6.2.8 Random Vibration**

Vibration is considered random when it doesn't conform to the vortex-induced or fluidelastic vibrations. This phenomenon includes the excitation due to vorticity generated by upstream structures.

### **6.2.9 Pressure Pulsation**

A specific frequency component may become dominant when flow changes due to effect of fluid machinery. If this frequency equals to the acoustic natural frequency

of any equipment, then high flow oscillations and related pressure pulsations occur. This phenomenon is also known as “flow oscillations”.

### **6.3 Literature Survey**

Flow noise has become one of the key factors on household refrigerators especially on recent years. Therefore, this subject is being examined and discussed increasingly in the open literature.

Han et al. (2009) have presented a study on the root-causes of refrigerant-induced noise in refrigerators. They concluded that bubbles in an intermittent flow have the most important effect on this noise. Therefore, flow patterns in region of two-phase flow should be considered more carefully. The flow pattern may be made steady by increasing mass flux or mass flow rate. These cyclic conditions are especially important in the evaporator inlet pipe.

Han et al. (2010) have examined the flow pattern of the refrigerant in an evaporator-inlet pipe; and estimated flow patterns also using different maps. They have monitored the characteristics of the flow in their experiments. The authors have discussed the relation between the flow pattern and flow noise; and suggested the shape and layout of the evaporator inlet to keep away from intermittent flow pattern.

Hartmann & Melo (2013) have presented an experimental study on popping noise in a refrigerator. Popping noise is a type of flow noise that occurs after some short time of the compressor start-up. The main cause of this strange and unpleasant noise has been explained by “condensation induced shock”. The authors have made some attempts to decrease this noise. These are, changing position and diameter of the driers, changing mass flow rate of the compressor and using an additional heat exchanger. The popping noise is completely eliminated when the dryer is positioned horizontally. But this solution is not suitable for applying to all refrigerators. The other effective solution is using a heat exchanger. They have concluded that this solution had benefit to eliminate the flow noise and the energy consumption of the appliance didn't change.



Hartmann & Melo (2014) have made an experimental study on capillary tube flow. They used a dual-evaporator, bottom-mount refrigerator with a variable speed compressor. The research has been focused on the effects of the flow pattern at the entrance of the capillary tube on the acoustical behavior of household refrigerators. They have stated that, the evaporator inlet pipe receives a high speed, two phase flow from the capillary tube; therefore, vapor bubbles are produced by oscillating, coalescing and collapsing. They have made different attempts to attenuate the fluctuating noise by producing steady, single phase flow at the capillary inlet, by increasing the compressor speed, increasing the refrigerant charge and by increasing the capillary tube restriction. The authors have concluded that the first two alternatives were not successful. Using capillary tubes with smaller diameters might be a solution. However their supply was limited and also installations were not practical

Neto et al. (2014) have analyzed fluid borne noise in refrigerators with experimental study. They have measured the acoustic pressure and the acceleration signals on different points that they determined on cooling circle system. They have found that the main source of acoustic excitation is the pulsations of the compressor. The other cause of refrigerant-induced noise was the high-speed superheated flow creating turbulent excitation. This excitation was more significant above 500 Hz frequencies. High-intensity spikes due to vapor bubbles were observed at the entrance of the capillary tube. They have recommended smaller radius tubing to recover the noise due to two-phase intermittent flow. Acoustic filters at the suction and discharge pipes of the compressor and also at the exit of the capillary tube were also recommended.

Capillary tube is the inlet part of the evaporator. One of the main reasons of the flow noise in refrigerators is the two-phase refrigerant flow at the capillary tube. Acoustic behavior of household refrigerators is greatly affected by the flow pattern of the capillary tube. The more oscillating this flow pattern is, the stronger the evaporator vibrations will be. When evaporator is vibrating severely, noise may be

noticeable outside of the appliance. Therefore, the flow at the inlet and outlet of the capillary tube is significant for the vibration and noise of the refrigerator. In Figure 6.3, the capillary tube inlet of an evaporator is shown.



Figure 6.3 Capillary tube inlet of evaporator

The previous studies have investigated the capillary geometry, mass velocity, mass quality of the refrigerator and bubble geometry. Since two-phase flow at the evaporator inlet is an important source of the flow noise in refrigerators, the design of the capillary should be effective on this physical phenomenon. However, no study has been published to date reporting on the effect of capillary design on the flow noise. The main idea of this research is based on this fact. Therefore, in the course of this flow noise study, the effect of the capillary position was examined by varying capillary insert length. The capillary position is kept horizontal as shown in Figure 6.4. In order to examine the flow characteristics and also flow noise, samples with different capillary lengths were produced and these were assembled to the same appliance. Then, acoustic tests were performed for these refrigerators.

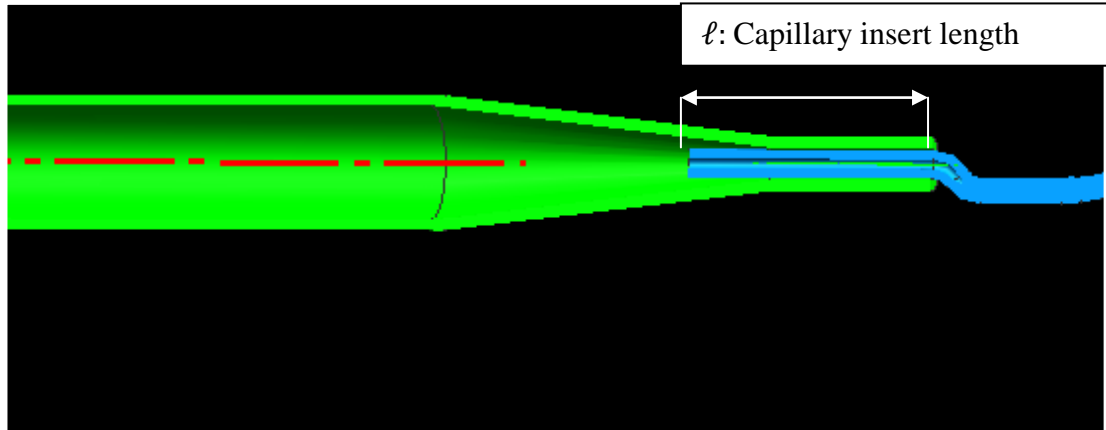


Figure 6.4 Capillary position of injection area

#### 6.4 Experimental Studies on Capillary Insert Length

In this study, it has been detected that the original capillary length of the refrigerator with flow noise was short (7 mm). Therefore, the two-phase flow at the evaporator inlet was being suddenly directed to a narrower cross sectional area. Refrigerant striking the walls was exerting high levels of noise. For this reason, some design changes were applied to increase the capillary insert length.

The refrigerator was a typical no frost, bottom-mount type, equipped with 1600 rpm compressor, running with the use of R-600a. Appliance was placed in a reverberation chamber shown in Figure 3.2 with acoustic characteristics given in Table 3.1. To determine the relationship between noise and cycle conditions, sound power level was determined simultaneously with the cycle temperature.

The first acoustic tests were performed for the refrigerator with original capillary length. The result was obtained in a fluctuating shape as shown in Figure 6.5. The mean sound power level was 39.8 dB(A) with  $\pm 2.5$  dB(A) fluctuations between the compressor start-up and up-down periods. The acoustic behavior is unsteady and large fluctuations proceed along all the compressor time. Figure 6.6 shows the variation of the cyclic temperature and electric power of refrigerator at the operating time. Sound power level variation in time and its 1/3 octave frequency spectrum are shown together in Figure 6.7. It can be noted that the flow noise has wide frequency content and considerable contributions are in the range of 200 Hz-1000 Hz.

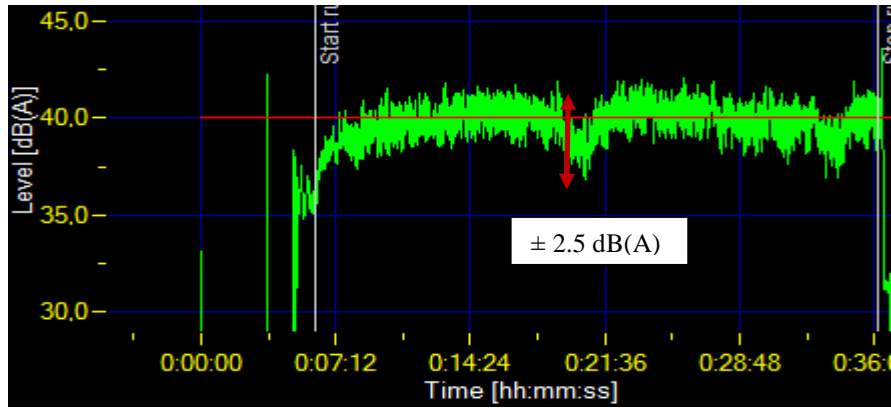


Figure 6.5 Sound power level variation of the refrigerator

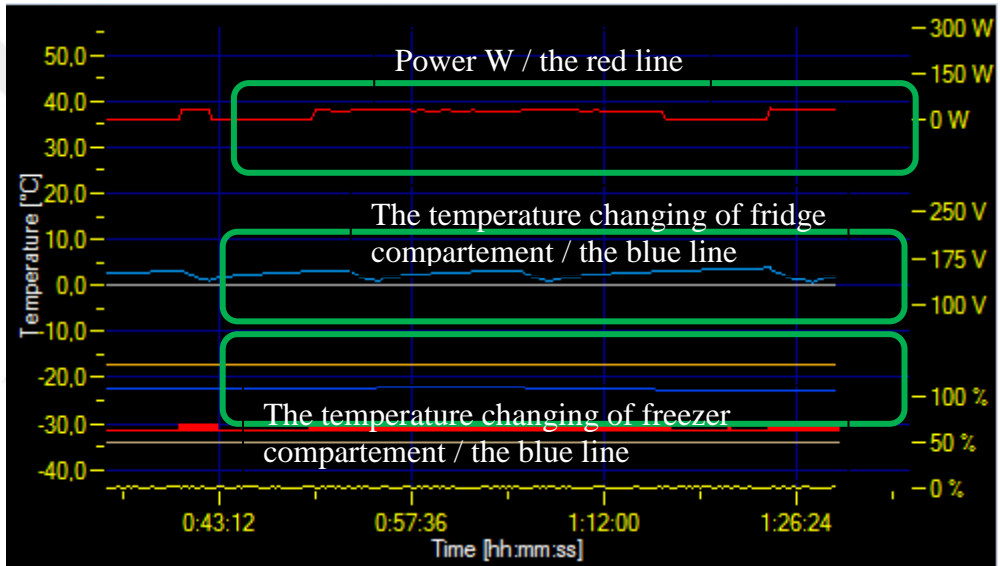


Figure 6.6 Cyclic temperature and electric power of the refrigerator

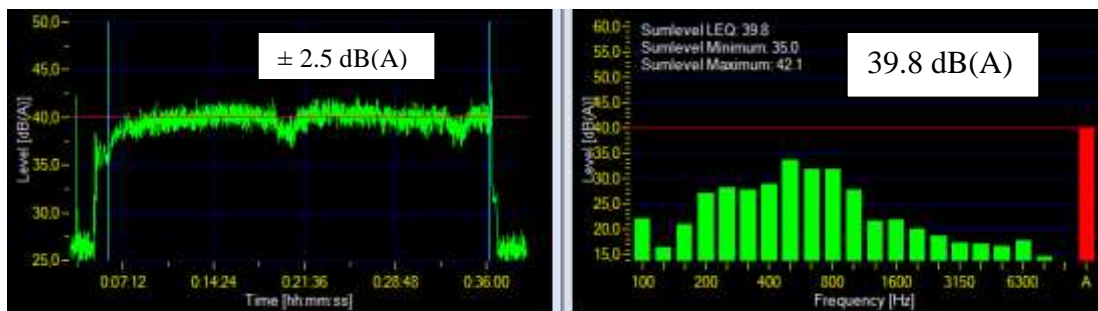


Figure 6.7 Sound power variations in time and frequency spectrum for  $\ell = 7$  mm

The fluctuations of sound power indicate that the noise is mainly due to flow. Nevertheless, the compressor and fan noise level should be investigated in order to gain a better understanding of the flow induced part of the noise. Therefore, the compressor and fan of the refrigerator were subjected to solo noise level measurements. The results of these measurements are shown in Figures 6.8 and 6.9 as overall sound power level dB(A) and 1/3 octave band spectrum. As it is seen, compressor and especially fan have narrow frequency bands.

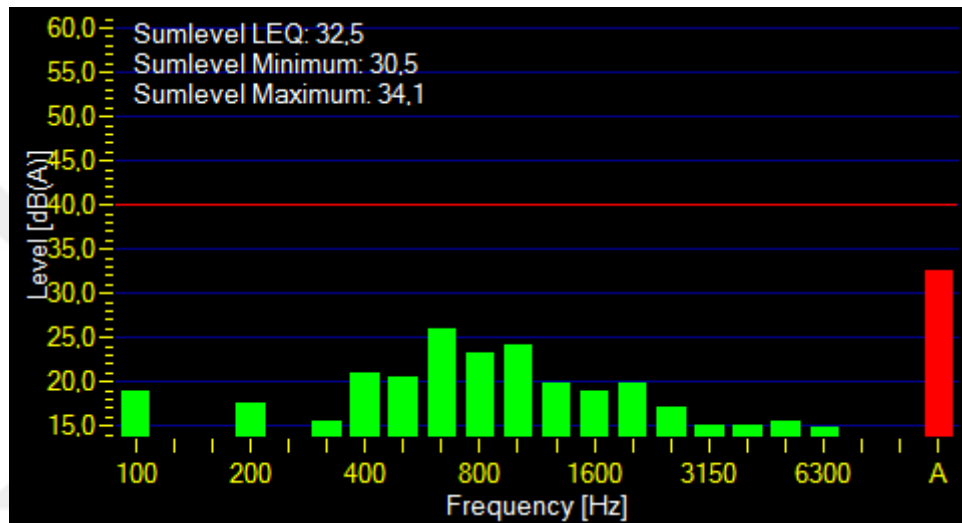


Figure 6.8 Solo compressor sound power level and frequency spectrum

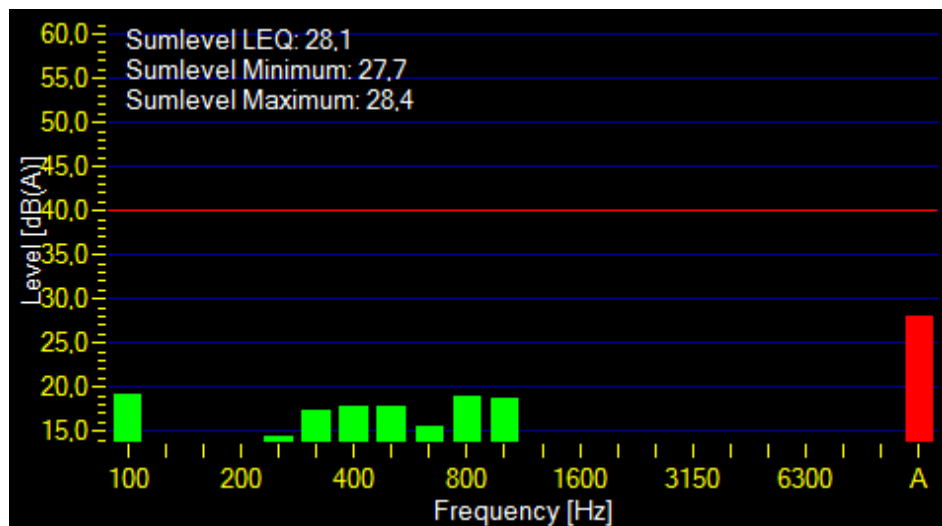


Figure 6.9 Solo fan sound power level and frequency spectrum

Using the decibel summation formula to obtain the total sound power level of compressor and fan,

$$L_A + L_B = 10 \log_{10} (10^{\frac{L_A}{10}} + 10^{\frac{L_B}{10}}) \quad (6.1)$$

where,

$L_A$ : The solo compressor noise level (32.5 dB(A))

$L_B$ : The solo fan noise level (28.1 dB(A))

$L_A + L_B = 33.84$  dB(A)

is obtained. Consequently, the difference between the total sound power level 39.8 dB(A) and the total solo level 33.84 dB (A) gives the contribution of flow noise.

In order to examine the flow noise, three different samples were prepared and tested. These samples have capillary insert lengths 15 mm, 17 mm and 20 mm. The sound power variation levels and 1/3 octave frequency spectra of the refrigerators with these samples are shown in Figures 6.10, 6.11 and 6.12 respectively. Abstract of the measurement is presented in Table 6.3. Results show the positive effect of increasing capillary insert length on both the level and variation of the sound. The fluid noise has been completely eliminated for the capillary tube with 20 mm insert length. For this case, the measured level 34.0 dB(A) with almost negligible variation is only the combined noise of the compressor and fan.

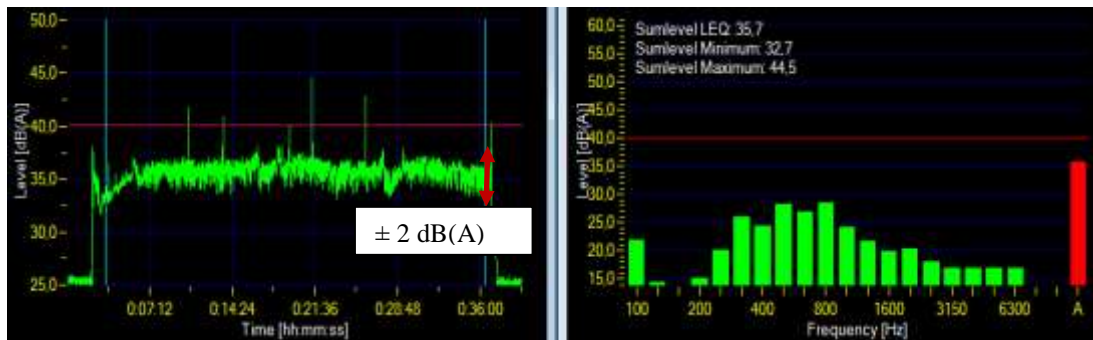


Figure 6.10 Sound power and frequency spectrum for  $\ell = 15$  mm.

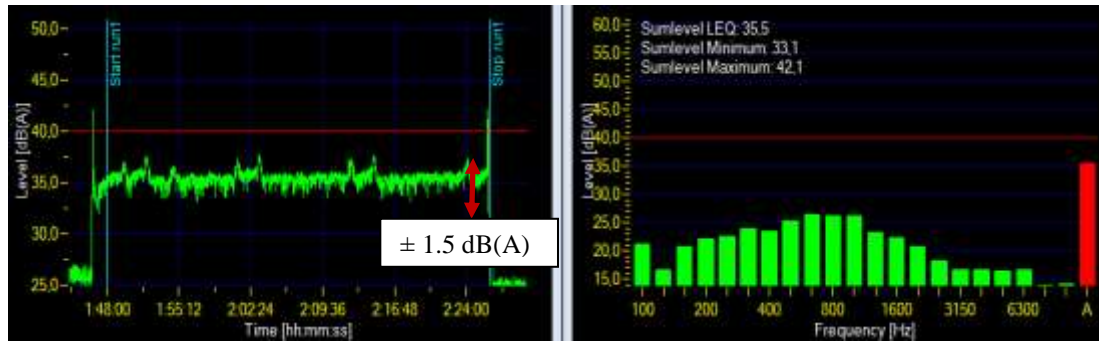


Figure 6.11 Sound power and frequency spectrum for  $\ell = 17$  mm.

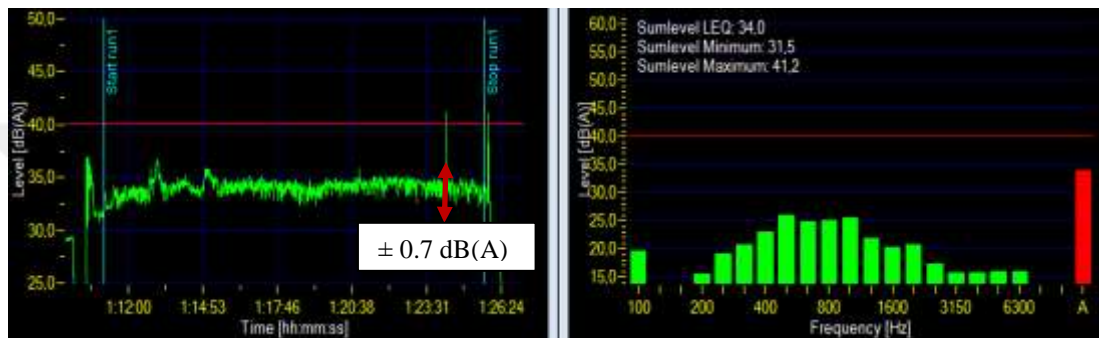


Figure 6.12 Sound power and frequency spectrum for  $\ell = 20$  mm.

Table 6.3 Abstract of measurements for different capillary insert lengths

Measurements	The capillary insert length (mm)	Sound Power Level dB(A)	The variation of sound power level dB(A)
1	7	39.8	$\pm 2.5$
2	15	35.7	$\pm 2$
3	17	35.5	$\pm 1.5$
4	20	34.0	$\pm 0.7$

## 6.5 Conclusion

-One reason of the flow noise is throttling of the refrigerant passing through the capillary insert of the refrigerator system. This process produces a two-phase flow which can even reach critical conditions in terms of the acoustic behavior.

-In order to control the flow noise of the refrigerator, the capillary design was improved by elongating capillary insert length.

-The new capillary insert length completely eliminated the flow noise of the refrigerator.

-The variation of sound power level with time was almost disappeared by this modification.





## **CHAPTER SEVEN**

### **CONCLUSION**

Vibro-acoustics has become a fundamental concept for design of state-of-the-art products. Refrigerators even if not considered to be so noisy products, as continuously operating domestic sound sources, they needed to be improved both in terms of technological requirements and environmental comfort.

This thesis is devoted to vibro-acoustic improvement of different refrigerators with several types of noise, using various approaches. These approaches were supported with tests repeated many times to make sure of the repeatability.

High-capacity refrigerators examined had high general noise level. As the first solution, a new sound insulation material composed of fenolic felt and sawdust was formed. Condenser fans were localized as the main sound sources of these refrigerators. Therefore as the second solution, these standard fans were replaced by a horizontal cylindrical fan. The use of this fan was approved also by the energy considerations. Consequently, these modifications were applied to refrigerators and a great deal of noise reduction was attained.

Defrost noise was examined for standard no-frost refrigerators. Although different alternative were studied, the only positive result was performed for the modification of defrost parameters including the increase of electrical power and the decrease of standard defrost time.

Flow noise was investigated also for standard no-frost refrigerators. The two-phase flow is one of the main reasons of the flow noise. Capillary insert length was modified to make the two-phase flow more annular. These studies were resulted in the removal of the flow noise completely.

These studies were intended to specific type of refrigerators. With respect to the developing technology and corresponding appliances new studies and solutions

including also defrost noise will be performed. Hence, vibro-acoustic research of refrigerators will continue in the future.



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## APPENDIX

Table A.1 Features of equipment used in the measurements

Number	Model	Firm	Description	Chapter
1	Type 4943	Bruel & Kjaer	½ inch diffuse – field microphone, 3.15 Hz to 10 kHz	5, 6
2	Type 4950	Bruel & Kjaer	½ inch free – field microphone, 3.15 Hz to 10 kHz	4
3	Type 4178	Bruel & Kjaer	¼ inch sound intensity microphone, 4 Hz to 100 kHz	4
4	Pulse Software	Bruel & Kjaer	Sound pressure, power measurement, FFT analysis, order analysis	4, 5, 6
6	Pulse Software	Bruel & Kjaer	Sound intensity mapping	4
7	Type 2690	Bruel & Kjaer	Nexus microphone conditioner	4, 5, 6
8	Type 3654	Bruel & Kjaer	Sound intensity probe kit	4
9	Agilent 34970A	Sigma Data	Datalogger	5, 6
10	Thermocouple	Sigma Data	Temperature sensor	4, 5, 6