VIBRATION AND ACOUSTIC ANALYSIS OF A REFRIGERATOR CABINET BASE PLATE

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ABSTRACT

VIBRATION AND ACOUSTIC ANALYSIS OF A REFRIGERATOR CABINET BASE PLATE

This research study deals with the vibration and acoustics analysis of a refrigerator cabinet base plate by using Theoretical Modal Analysis by Finite Element Method and Experimental Modal Analysis.

In this study, a cabinet base plate is considered to investigate the titled problem. This cabinet base plate has several non-uniform geometrical properties such as holes, stiffeners and mounting places. Due to the aforementioned non-uniformity, numerical methods are used to find the vibration and acoustics properties such components.

The aim of the first important part of this thesis is to provide the theoretical background to the readers. Other parts of this thesis are presented for theoretical modal analysis of cabinet base plate which is carried out by using solid element in ABAQUS and experimental modal analysis that is accomplished by using dB4-1 Mobil Analyser and its connections.

The modal parameters found from both numerical and experimental methods are compared and good agreement is found. After determining the agreement, the proper modification has been done for the system which generates undesired sound due to the natural frequency.

ÖZET

BİR BUZDOLABI KABİN TABAN PLAKASININ TİTREŞİM VE AKUSTİK ANALİZİ

Bu araştırma çalışması, bir buzdalabı kabin taban plakasının Sonlu Elemanlar Metodu ile yapılan Teorik Modal Analizi ve Deneysel Modal Analizi aracılığı ile titreşim ve akustik analizi ile ilgilidir.

Bu çalışmada, kabin taban plakası başlıktaki problemi araştırmak üzere göz önüne alınmştır. Kabin taban plakası delikler, direngenlik arttırıcılar ve montajlama yerleri olmak üzere çeşitli düzgün olmayan geometrilere sahiptir. Belirtilen bu düzgünsüzlükler nedeni ile böylesi bir parçanın titreşim ve ses özelliklerini bulmak için sayısal yöntemler kullanılır.

Bu tezin birinci önemli bölümünün amacı okuyuculara teorik bilgi sağlamaktır. Bu tezdeki diğer bölümler, ABAQUS da katı elemanlar kullanılarak yapılan kabin taban plakası teorik modal analizi ve dB4-1 Mobil Analyser ve onun bileşenleri ile başarılan deneysel modal analiz için sunulmuştur.

Sayısal ve deneysel metodlar ile bulunan modal parametreler karşılaştırılmış ve iyi bir uyum gözlenmiştir. Bu uyumun tesbiti sonrası, doğal frekans nedeni ile ses oluşturan sistemde uygun değişiklik yapılmıştır.

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LIST OF SYMBOLS

$A(\omega)$	Accelerance FRF
$[B]$	strain matrix
c	damping coefficient
$[D]$	damping matrix
E	Young's modulus
f_{i}	i^{th} natural frequency
$\{f(t)\}$	force vector
$\{F(\omega)\}$	complex amplitude of harmonic force
j	complex number
k	stiffness coefficient
$[K]$	stiffness matrix
m	mass
$[M]$	mass matrix
$[N]$	assumed displacement functions matrix
R and R^*	residues of the receptance
T	kinetic energy
U	strain energy
$\{u_i\}$	i^{th} vibration mode shape vector
$\{x(t)\}$	displacement vector
$\{X(\omega)\}$	complex amplitude of displacement
$Y(\omega)$	Mobility FRF
$[Z(\omega)]$	dynamic stiffness matrix
$\alpha(\omega)$	frequency response function
λ and λ^*	complex poles
ρ	density
ξ	damping ratio
v	poisson's ratio
ω	frequency
ω_i	i^{th} natural radian frequency

CHAPTER 1

GENERAL INTRODUCTION

Vibration is the motion of elastic systems about equilibrium positions. It is generally undesired motions for most of the machines and structures. It causes higher stresses, wear of machine parts, damage to machines and undesired sounds.

In the past two decades, modal analysis has become a major technology in the quest for determining, improving and optimizing dynamic characteristics of engineering structures. Finite element analysis as a computer modeling approach has provided engineers with a versatile design tool, especially when dynamic properties need to be perused. Computer modeling alone cannot determine completely the dynamic behavior of structures, because certain structural properties such as damping and nonlinearity do not conform to traditional modeling treatment (He and Fu 2001).

The experimental techniques are nurtured by the theory of modal analysis and in turn provide new impetus to it. Modal analysis is the process of determining the inherent dynamic characteristics of a system in forms of natural frequencies, damping factors and mode shapes, and using them to formulate a mathematical model for its dynamic behavior. The formulated mathematical model is referred to as the modal model of the system and the information for the characteristics is known as its modal data. Modal analysis embraces both theoretical and experimental techniques. The practice of modal testing involves measuring the FRFs or impulse responses of a structure (He and Fu 2001).

The invention of the fast Fourier transform (FFT) algorithm by J.W. Cooly and J.W. Tukey in 1965 finally paved the way for rapid and prevalent application of experimental technique in structural dynamics. With FFT, frequency responses of a structure can be computed from the measurement of given inputs and resultant responses. The first, and perhaps the most significant, method of experimental modal analysis was proposed by C.C. Kennedy and C.D. Pancu in 1947 before FFT was conceived. Their method was largely forgotten until FFT gave life to experimental modal analysis. Since then, numerous methods have been proposed and many have been computerized, including those time domain methods which are based on free vibration of a structure rather than its frequency responses (He and Fu 2001).

Bishop and Gladwell (1963) described the state of the theory of resonance testing which was considerably in advance of its practical implementation. 10000 papers published on modal testing and analysis since 1984 to 2000 (Ewins 2000).

Finally, Deveci et al. studied on the freezer fan noise of a no-frost type refrigerator by numerical and experimental methods. A coupled numerical model of the freezer fan compartment was created with ABAQUS finite elements analysis and simulation software. An experimental modal analysis was performed on the evaporator cover and the fan louver. Accelerometers (Brüel&Kjaer Type 4507B, Sensitivity: 1 mV/ms2), a modal hammer (Endevco Type 2302, Sensitivity: 11.4 mV/N), a portable 4-channel data acquisition unit and general purpose measurement software (Brüel&Kjaer Pulse Type 3560D) were used to conduct the experimental modal analysis.

Nowadays, the most critical parameters for refrigerators are the energy consumption and the acoustic properties.

Therefore, noise-free refrigerators become the preferred choice of customers. Noise emitted by refrigeration, one or more different part of the product occurs due to vibrations that occur during operation.

- Three types of refrigerators are available in the market:
- Static: cooling provided by evaporation pipes behind the wall.
- Brewed: a static refrigerator equipped with fan located inside.
- No-frost: a fan (embedded in the back wall) pushes air to flow over the evaporator before entering into the refrigerating compartment.

Compressors mounts on refrigerator cabinet is the vibration source for whole body of refrigerators. Therefore, cabinet base plate is very critical structural component on sound power level of a refrigerator.

The modification on cabinet base plate is very effective for the desired acoustic properties for the majority of some kind of refrigerators which have no more excitation sources such as extra fans etc.

In this study, the main interest is the theoretical and experimental modal testing of existing cabinet base plate which causes vibration and acoustics problems in the products in VESTEL.

CHAPTER 2

THEORETICAL BACKGROUND

2.1. Introduction

In this chapter, theoretical and experimental modal analysis, finite element modeling are summarized. Theoretical modal analysis includes FRF of SDoF and MDoF system, graphical displays for sample FRFs. Finite element background is provided for strain and kinetic energy for arbitrary shaped solid, element mass/stiffness matrices. Equation of motion, natural frequencies and mode shapes are introduced. Finally, measurement hardware, signal processing, modal data extraction are given.

2.2. Equation of Motion

The general differential equation of a forced vibration is given by

$$[M]\{\ddot{x}(t)\}+[D]\{\dot{x}(t)\}+[K]\{x(t)\}=\{f(t)\} \tag{2.1}$$

where $[M]$, $[D]$ and $[K]$ are mass, damping and stiffness matrices, respectively. Also, $\{x(t)\}$ is displacement vector and $\{f(t)\}$ is force vector.

2.2.1. Natural Frequencies

Considering the harmonic motion for displacement and force vectors in the general differential equation, the following generalized eigenvalue equation is written:

$$([K]-\omega_i^2[M])\{u_i\}=\{0\} \tag{2.2}$$

where ω_i is i^{th} natural frequency of the system.

2.2.2. Mode Shapes

$\{u_i\}$ in Equation given in Section on "Natural Frequencies" is the i^{th} vibration mode shape vector.

2.3. Theoretical Modal Analysis

2.3.1. Frequency response functions of an SDoF system

Some mechanical and structural systems can be idealized as SDoF systems. The SDoF system having a mass, a spring and a viscous damper is considered. For a harmonic force $f(t) = F(\omega)\, e^{j\omega t}$, the response of the system is another harmonic function $x(t) = X(\omega)\, e^{j\omega t}$ where $X(\omega)$ is a complex amplitude. The frequency response function (FRF) of the system is given by (He and Fu 2001)

$$\alpha(\omega) = \frac{X(\omega)}{F(\omega)} = \frac{1}{k - \omega^2 m + j\omega c} \tag{2.3}$$

or in different forms as

$$\alpha(\omega) = \frac{1/k}{1 - \dfrac{\omega^2}{\omega_0^2} + j2\dfrac{\omega}{\omega_0}} \tag{2.4}$$

$$\alpha(\omega) = \frac{1/m}{\omega_0^2 - \omega^2 + j2\omega\omega_0\xi} \tag{2.5}$$

Mobility FRF for this system is given as

$$Y(\omega) = \frac{\dot{X}(\omega)}{F(\omega)} = \frac{j\omega}{k - \omega^2 m + j\omega c} \tag{2.6}$$

Accelerance FRF for this system is given as

$$A(\omega)=\frac{\ddot{X}(\omega)}{F(\omega)}=\frac{-\omega^2}{k-\omega^2 m+j\omega c} \tag{2.7}$$

It is known that α(ω), Y(ω), and A(ω) have the following relationships:

$$|A(\omega)|=\omega|Y(\omega)|=\omega^2|\alpha(\omega)| \tag{2.8}$$

The reciprocals of α(ω), Y(ω), and A(ω) of an SDoF system also used in modal analysis. They are respectively:

$$Dynamic\ stiffness=\frac{1}{\alpha(\omega)}=\frac{force}{displacement} \tag{2.9}$$

$$Mechanical\ \ impedance=\frac{1}{Y(\omega)}=\frac{force}{velocity} \tag{2.10}$$

$$Apparent\ \ mass=\frac{1}{A(\omega)}=\frac{force}{acceleration} \tag{2.11}$$

The FRF of an SDoF system can be presented in different forms to those in previous equations. The receptance FRF can be factorized to become:

$$\alpha(\omega)=\frac{R}{j\omega-\lambda}+\frac{R^*}{j\omega-\lambda^*} \tag{2.12}$$

where

$$R=\frac{1}{2m\omega_0 j} \tag{2.13}$$

$$\lambda=(-\xi+\sqrt{1-\xi^2}\,j)\omega_0 \tag{2.14}$$

Conjugate coefficients R and R^* are called residues of the receptance. λ and λ^* are the complex poles of the SDoF system.

2.3.2. Graphical display of a frequency response function

2-D graphical display of an FRF is possible in different approaches. They can be listed as follows:

1. Amplitude–phase plot and log–log plot
2. Real and imaginary plots
3. Nyquist plot
4. Dynamic stiffness plot

Details of the listed items are available in the textbook written by He and Fu (2001). A sample frequency response function with linear-linear plot is shown in Figure 2.1.

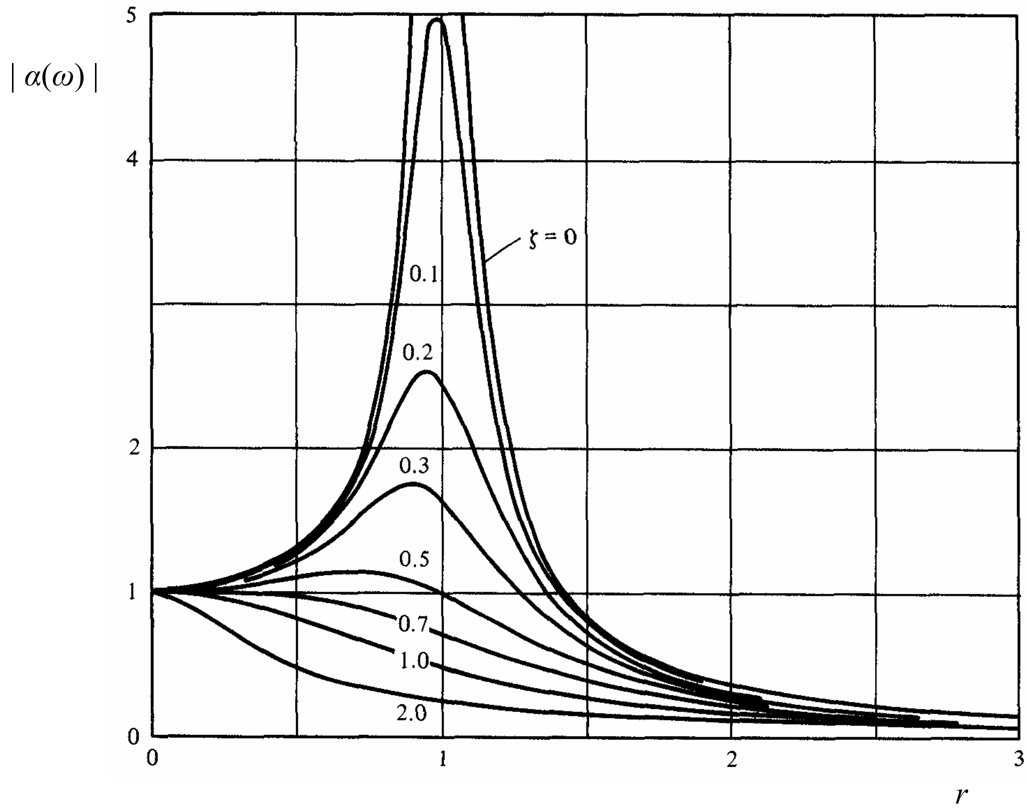

Figure 2.1. A sample for frequency response function

2.3.3. Frequency response functions of a damped MDoF system

The equations of motion of a MDoF system is written in matrix form as:

$$[M]\{\ddot{x}(t)\}+[D]\{\dot{x}(t)\}+[K]\{x(t)\}=\{f(t)\} \qquad (2.15)$$

where $[M]$, $[D]$ and $[K]$ are mass, damping and stiffness matrices, respectively. Also, $\{x\}$ is displacement vector and $\{f(t)\}$ is force vector.

Considering the harmonic motion for displacement and force, the following equation can be written:

$$[Z(\omega)]\{X\} = \{F\} \tag{2.16}$$

where $[Z(\omega)] = ([K] - \omega^2[M])$ is known as the dynamic stiffness matrix of an MDoF system. Similar to SDoF system, the inverse of dynamic stiffness matrix gives the receptance FRF matrix of the system and is denoted by $[\alpha(\omega)]$. It is in open form as:

$$[\alpha(\omega)] = ([K] - \omega^2[M])^{-1} \tag{2.17}$$

or

$$[\alpha(\omega) = \begin{bmatrix} \alpha_{11}(\omega) & \alpha_{12}(\omega) & \ldots & \alpha_{1n}(\omega) \\ \alpha_{21}(\omega) & \alpha_{22}(\omega) & \ldots & \alpha_{2n}(\omega) \\ \ldots & \ldots & \ldots & \ldots \\ \alpha_{n1}(\omega) & \alpha_{n2}(\omega) & \ldots & \alpha_{nn}(\omega) \end{bmatrix} \tag{2.18}$$

$\alpha_{ij}(\omega)$ is the frequency response function when the system only has one input force applied at coordinate 'j' and the response is measured at coordinate 'i'. A sample plot for receptance $\alpha_{11}(\omega)$ of the 4DoF system is shown in Figure 2.2. It is seen from Figure 2.2 that there are 4 peaks due to the degrees of freedom of the system.

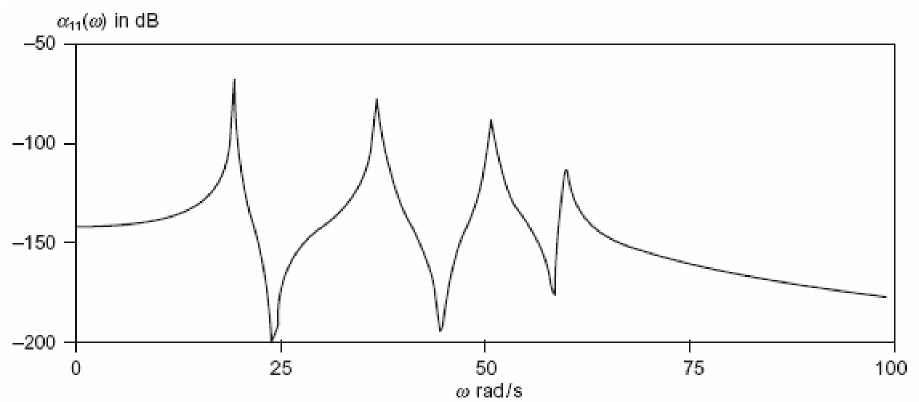

Figure 2.2. A sample for receptance $\alpha_{11}(\omega)$ of the 4DoF system

2.4. Finite Element Modeling

2.4.1. Strain Energy for Arbitrary Shaped Solid

For a three-dimensional solid of volume *V* shown in Figure 2.3, the strain energy is given by the following equations

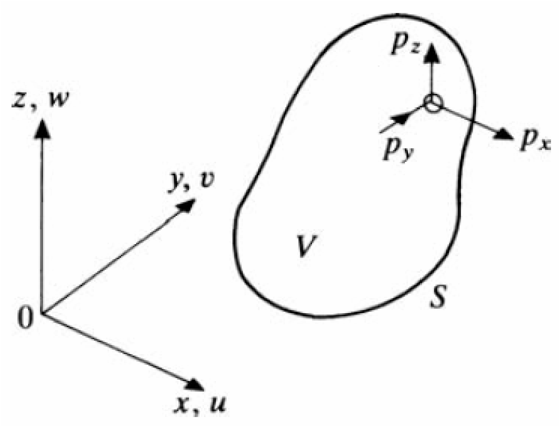

Figure 2.3. Three-dimensional solid.
(Source: Petyt 2010)

$$U = \frac{1}{2}\int_V \{\sigma\}^T \{\varepsilon\}\, dV \tag{2.19}$$

where

$$\{\sigma\}^T = \{\sigma_x \quad \sigma_y \quad \sigma_z \quad \tau_{xy} \quad \tau_{xz} \quad \tau_{yz}\} \tag{2.20}$$

$$\{\varepsilon\}^T = \{\varepsilon_x \quad \varepsilon_y \quad \varepsilon_z \quad \gamma_{xy} \quad \gamma_{xz} \quad \gamma_{yz}\} \tag{2.21}$$

The stress–strain relationships take the form

$$\{\sigma\} = [D]\{\varepsilon\} \tag{2.22}$$

where [*D*] is a symmetric matrix. For an anisotropic material it contains 21 independent material constants. In the case of an isotropic material it is

$$[D] = \frac{E}{(1+\nu)(1-\nu)} \times \begin{bmatrix} (1-\nu) & \nu & \nu & 0 & 0 & 0 \\ & (1-\nu) & \nu & 0 & 0 & 0 \\ & & (1-\nu) & 0 & 0 & 0 \\ & & & \frac{1}{2}(1-2\nu) & 0 & 0 \\ & & & & \frac{1}{2}(1-2\nu) & 0 \\ sym & & & & & \frac{1}{2}(1-2\nu) \end{bmatrix} \tag{2.23}$$

where *E* is Young's modulus and *v* is poisson's ratio. Substituting the stress–strain relationships into the strain energy expression gives

$$U = \frac{1}{2}\int_V \{\varepsilon\}^T [D]\{\varepsilon\}\, dV \tag{2.24}$$

where

$$\{\varepsilon\} = \begin{bmatrix} \frac{\partial u}{\partial x} \\ \frac{\partial v}{\partial y} \\ \frac{\partial w}{\partial z} \\ \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x} \\ \frac{\partial u}{\partial z} + \frac{\partial w}{\partial x} \\ \frac{\partial v}{\partial z} + \frac{\partial w}{\partial y} \end{bmatrix} \tag{2.25}$$

2.4.2. Strain Energy for Arbitrary Shaped Solid

The kinetic energy is given by (Petyt 2010)

$$T = \frac{1}{2}\int_V \rho\,(\dot{u}^2 + \dot{v}^2 + \dot{w}^2)\,dV \tag{2.26}$$

where ρ is density.

2.4.3. Element Mass and Stiffness Matrices

The mass matrix of an element is given by the following integral (Petyt 2010)

$$[m]_e = \int_{Ve} \rho [N]^T [N]\,dV \tag{2.27}$$

where [N] is a matrix having assumed displacement functions.

The mass matrix of an element is given by the following integral (Petyt 2010)

$$[k]_e = \int_{Ve} [B]^T [D][B]\,dV \tag{2.28}$$

where [B] is strain matrix.

2.5. Experimental Modal Analysis

2.5.1. Measurement Hardware

The hardware elements required consist of

- a source of excitation for providing a known or controlled input to the structure,
- a transducer to convert the mechanical motion of the structure into an electrical signal,
- a signal conditioning amplifier to match the characteristics of the transducer to the input electronics of the digital data acquisition system,
- an analysis system (or analyzer), in which signal processing and modal analysis programs reside.

Figure 2.4 shows a diagram of a basic test system configuration.

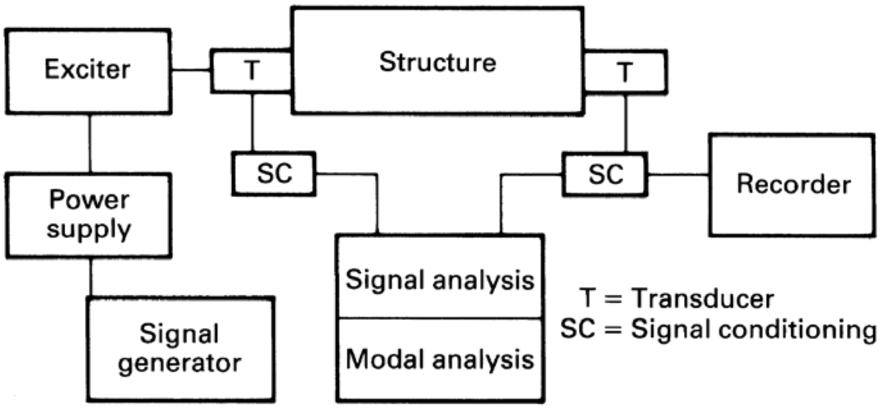

Figure 2.4. General test configuration
(Source: Inman, 2006)

2.5.2. Signal Processing

The analyzer is used to convert the analog time domain signals into digital-frequency-domain information. The analyzer perform the required computations by Fourier transform to alter an analog signal $x(t)$, into frequency-domain information x_n. Figure 2.4.

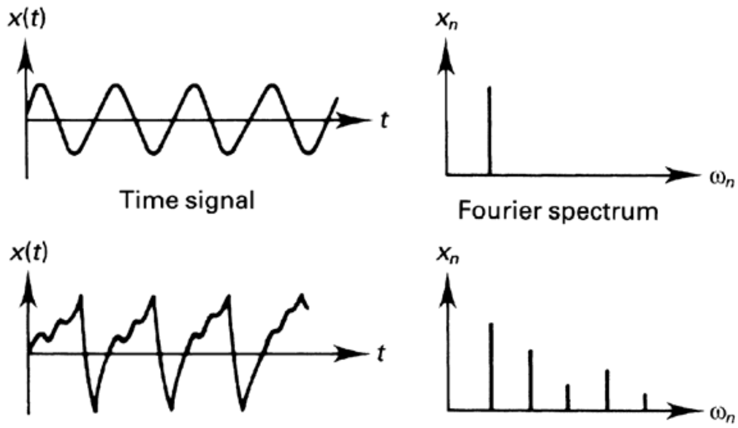

Figure 2.5. Some signals and their Fourier Spectrum
(Source: Inman, 2006)

2.5.3. Modal Data Extraction

After obtaining the FRF (frequency response function) $\alpha(j\omega)$ shown as in Figure 2.6, the task is to compute the *natural frequencies* and *damping ratios*. There are several ways to examine the measured FRF to extract these data. To examine all of them, interested reader should consult well-known textbook written by Ewins (2000). SDOF Method or Peak Amplitude Model is summarized below:

This method works adequately for structures whose FRF exhibit well- separated modes. The method is applied as follows:

(i) First, individual resonance peaks are detected on the FRF plot, and the frequency of one of the maximum responses taken as the natural frequency of that mode (ω_r).

(ii) Second, the local maximum value of the FRF is noted and the frequency bandwidth of 'half-power points' is determined ($\Delta\omega$)).

(iii) The damping of the mode in question can now be estimated from one of the following formulae

$$2\xi_r = \frac{\omega_a^2 - \omega_b^2}{2\omega_r^2} \cong \frac{\Delta\omega}{\omega_r} \tag{2.29}$$

(iv) Last, the estimated modal constant of the mode is calculated as

$$A_r = 2\xi_r\omega_r^2\left|\alpha(j\omega)\right| \tag{2.30}$$

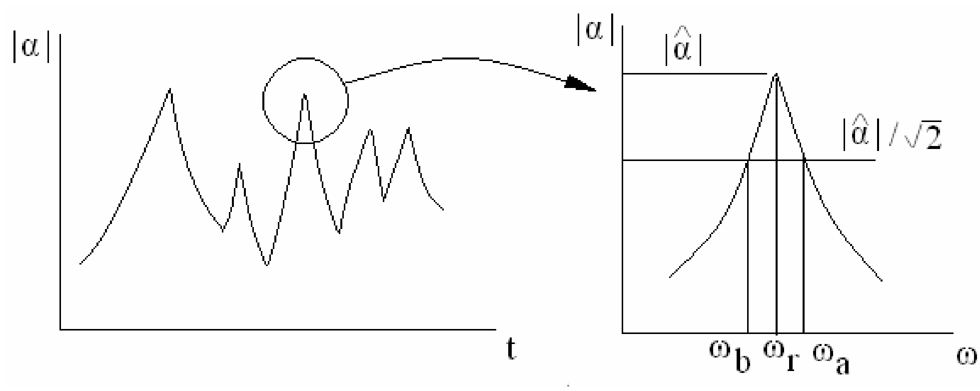

Figure 2.6. FRF (frequency response function)

(Source: Ewins, 2000)

CHAPTER 3

THEORETICAL MODAL ANALYSIS

3.1. Introduction

The original cabinet base plate has three long holes. It has rezonance problem about 50 Hz. This situation can be eliminated by design modifications. In this chapter, cabiner base plate with different number of holes are analyzed. The material properties of the cabinet base plate are given as follows: Density ρ= 7.85 10^{-9} ton/mm^3, Young modulus E=205.8 kN/mm^2 and Poisson's ratio υ=0,28. The width, length and thickness of the plate are measured, respectively, as: 120 mm, 585 mm, and 1.5 mm.

3.2. Finite Element Model

Cabinet base plate is created in ABAQUS by using about 30000 3D tetrahedral stress elements for finite element model. Cabinet base plate with compressor is shown in Figure 3.1.

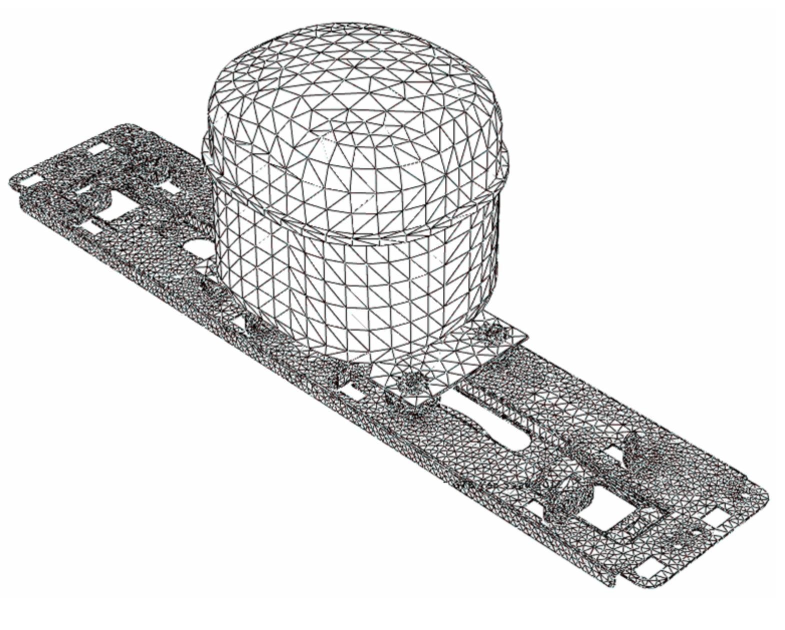

Figure 3.1. Finite element model of a plate

3.3. Natural Frequencies and Mode Shapes

The cabinet base plates with one, two, and three long holes are shown in Figure 3.2. The natural frequencies of the cabinet base plates are found from the finite element models generated in ABAQUS and given in Table 3.1. Also, the mode shapes of the cabinet base plate with two holes are shown in Figures 3.3-3.8.

Figure 3.2. Cabinet base plates with one, two and three long holes

Table 3.1 Theoretical Modal Analysis Frequencies (Hz) by ABAQUS

Natural frequencies	Cabinet base plate with one hole	Cabinet base plate with two holes	Cabinet base plate with three holes
f_1	38.9	40.3	43.6
f_2	42.3	46.8	49.8
f_3	*50.0*	53.8	62.6
f_4	55.5	67.2	81.8
f_5	76.2	87.3	88.6
f_6	90.0	95.1	105.5

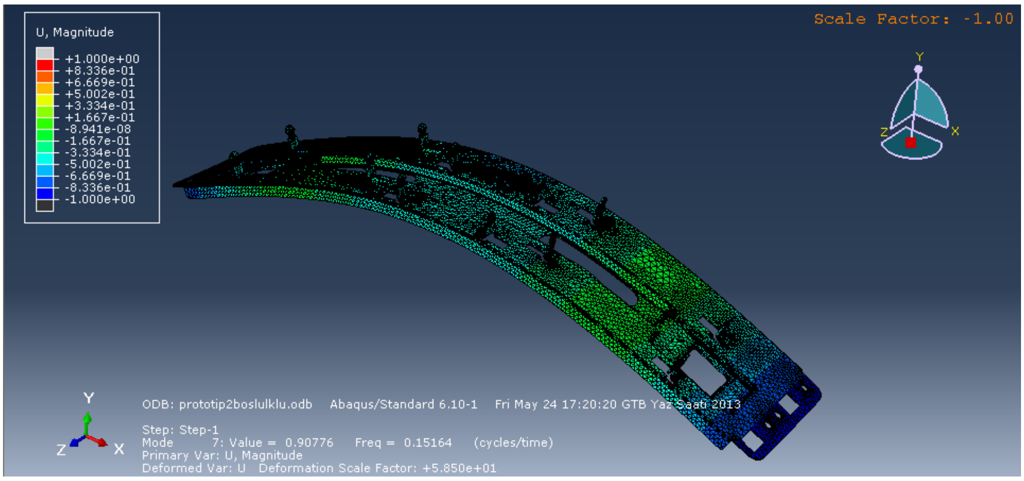

Figure 3.3. First mode shape of cabinet base plate

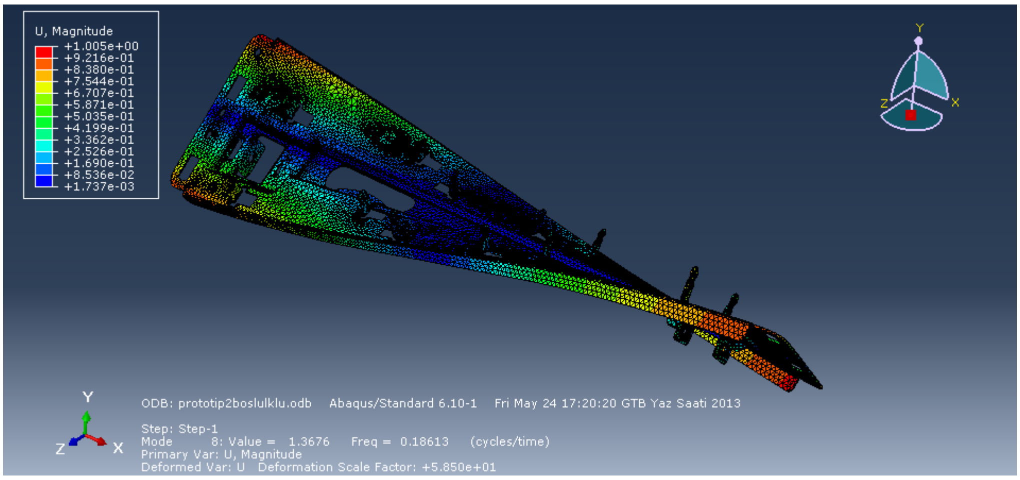

Figure 3.4. Second mode shape of cabinet base plate

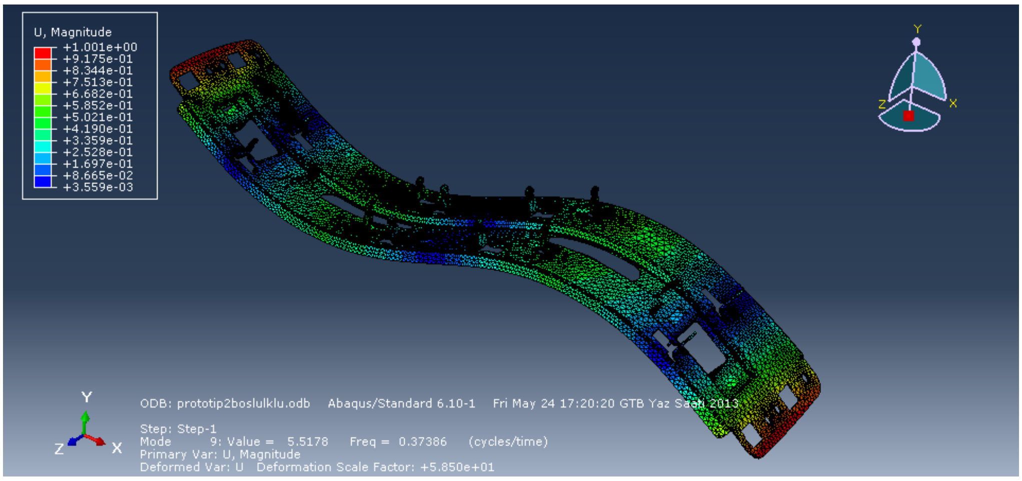

Figure 3.5. Third mode shape of cabinet base plate

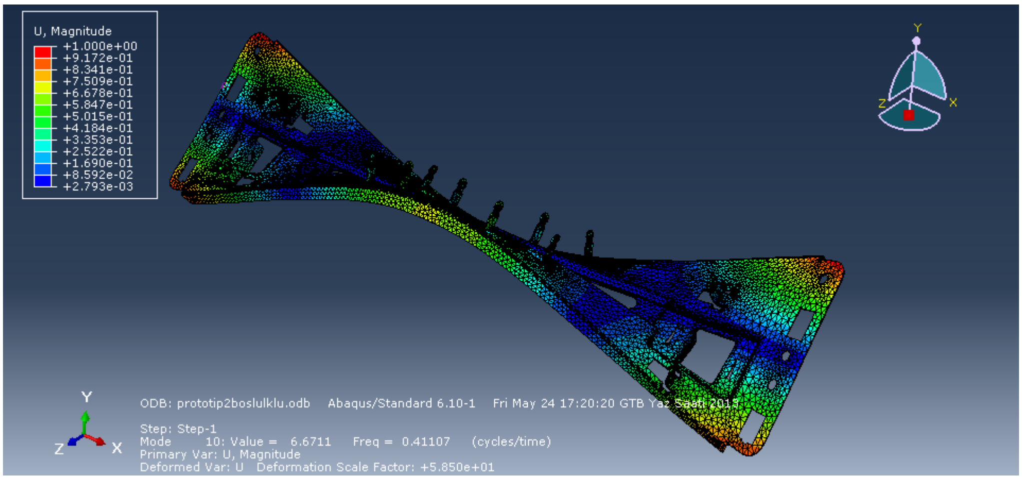

Figure 3.6. Fourth mode shape of cabinet base plate

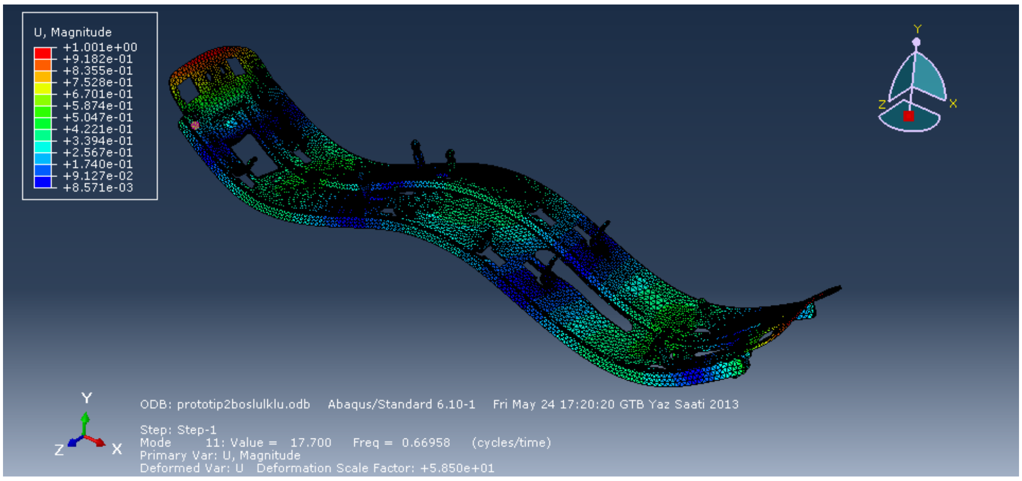

Figure 3.7. Fifth mode shape of cabinet base plate

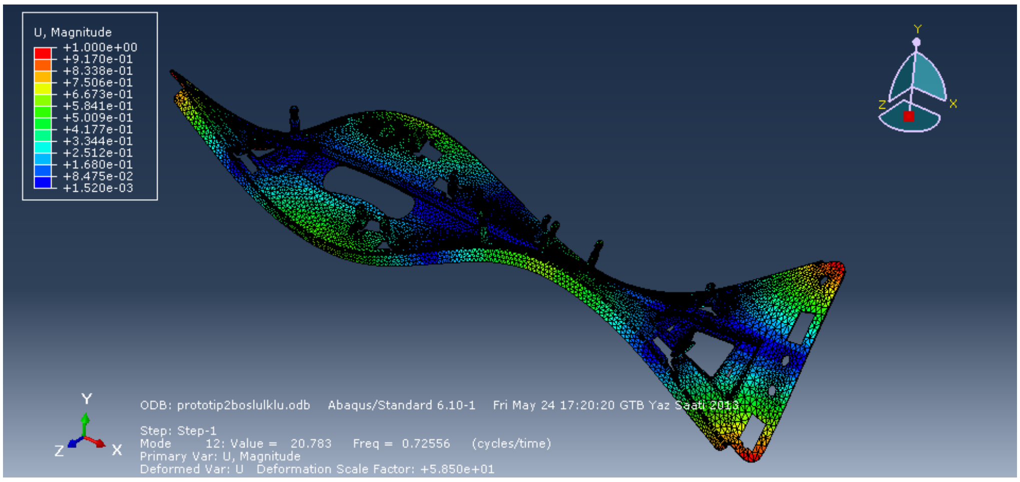

Figure 3.8. Sixth mode shape of cabinet base plate

CHAPTER 4

EXPERIMENTAL MODAL ANALYSIS

4.1. Experimental Setup

4.1.1. Introduction

The experimental modal analysis set-up shown in Figure 4.1. It includes hammer, signal analyzer, cabinet base plate with accelerometer. The details of the components of this system are given in the next subsections. Cross-spectrum test set-up is shown in Figure 4.2.

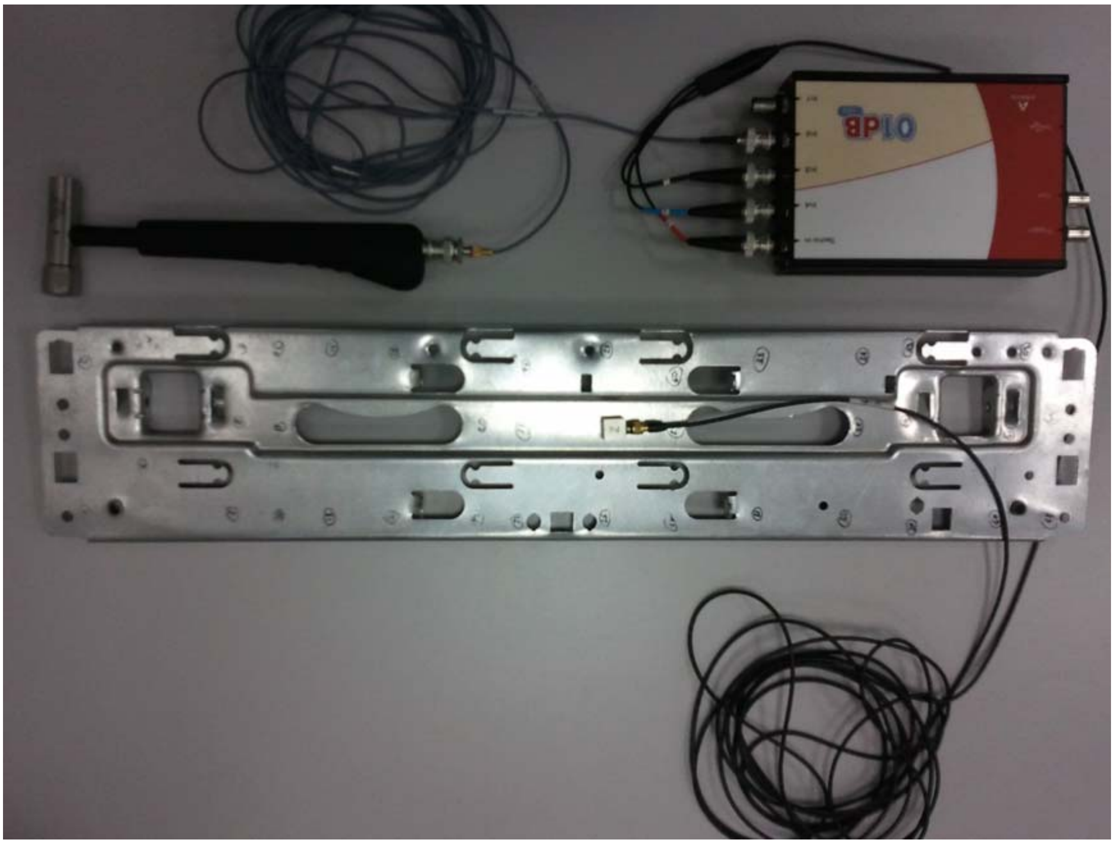

Figure 4.1. Experimental setup for cabinet base plate

Figure 4.2. Cabinet base plate cross-spectrum test set-up

4.1.2. Exciter

As an exciter, the impact hammer shown in Figure 4.3 is used. Dimensions of the hammer is shown in Figure 4.4. The main properties of this hammer are as follows:

- Four types with sensitivity from 1 to 22 mV/N
- Three replaceable tips
- Acceleration compensated

Figure 4.3. B&K Type Impact Hammer Type 8206-002

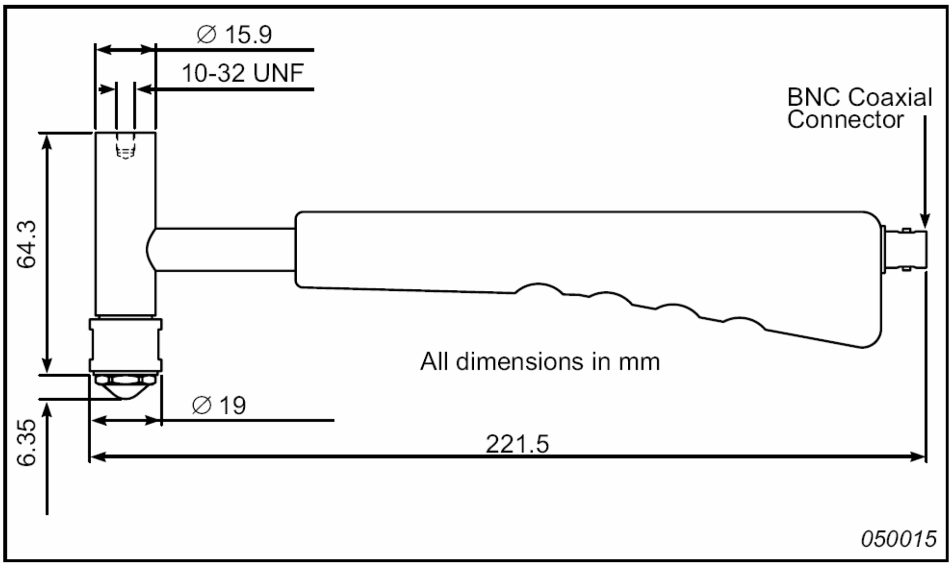

Figure 4.4. B&K Type Impact Hammer Type 8206-002 dimensions

4.1.3. Accelerometers

To have the displacements at specified points in the cabinet base plate, the accelerometer shown in Figure 4.5 is used. The properties of this accelerometer as follows:

Sensitivities for each direction are different as:

103.3 mV/g for *x*,

101.5 mV/g for *y*

106.4 mV/g for *z*

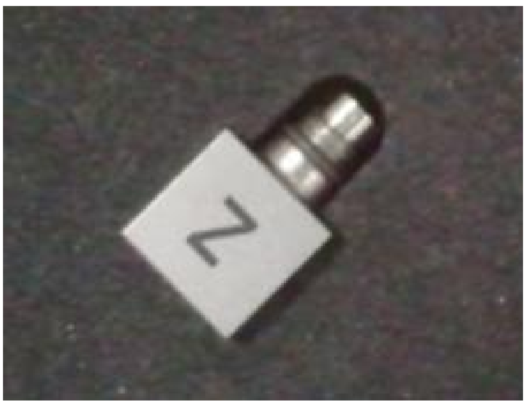

Figure 4.5. B&K 3D Accelerometer Type 8206-002.

4.1.4. Signal Conditioners

dBFA Suite 4.9 software is used as signal conditioner software of which interface is shown in Figure 4.6.

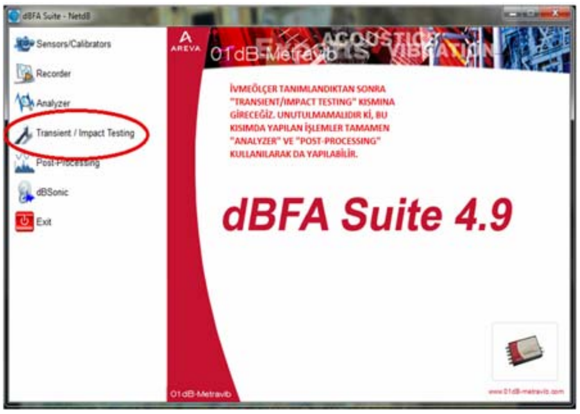

Figure 4.6. dBFA Suite 4.9 Software interface.

4.1.5. Recorder

dBFA Suite 4.9 Software is used as a signal recorder software.

4.1.6. Signal Analyzer

dB4 4-Channel analyzer shown in Figure 4.7 is used. It is a portable noise and vibration analyzer consisting of a USB2.0 4-channel acquisition unit combined with one of the software program of the 01dB range. It offers 4 synchronized 24-bit ICP inputs.

Figure 4.7. dB4 4-Channel analyzer

4.2. Natural Frequencies and Mode Shapes

Experimental modal test frequencies for the cabinet base plate with three holes found from the set-up are given in Table 4.2.

Table 4.2 Experimental frequencies (Hz) for the cabinet base plate with three holes

f_1	44,7
f_2	49.5
f_3	62,5
f_4	82.0
f_5	89.0
f_6	109.0

In order to obtain the mode shape of the cabinet base plate for the desired frequencies, the measurement points are determined as shown in Figure 4.1. By using the modal hammer introduced in Section 4.1.2, vibration data are collected. Then, the first mode shape shown in Figure 4.8 is found which is the same with Figure 3.3.

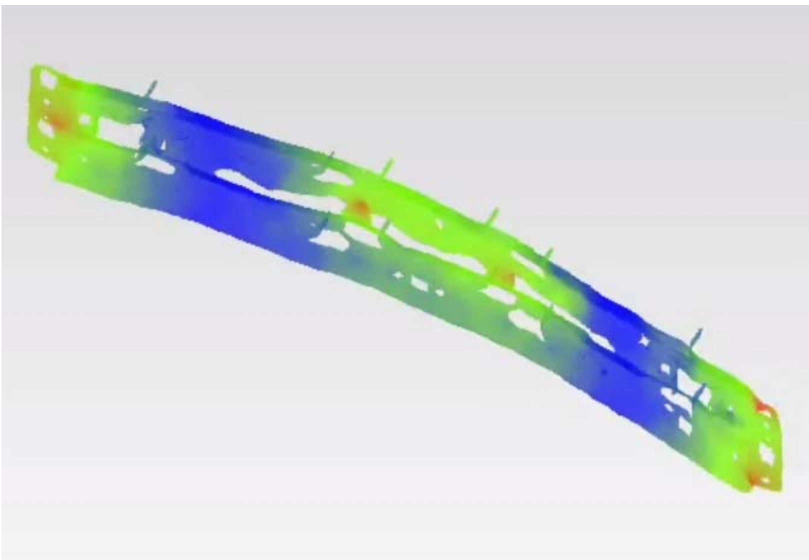

Figure 4.8. First mode shape of cabinet base plate obtained by experimentally

CHAPTER 5

MODIFICATION AND DISCUSSION OF RESULTS

5.1. Introduction

Compressor excites the cabinet base plate at 50 Hz while refrigerator operates. It can be seen from Table 3.1 and Table 4.2 that second natural frequency of cabinet base plate with three holes is calculated 49.8 Hz from ABAQUS and validated by experimentally as 49.5 Hz. Therefore, there is a resonance problem for cabinet for aforementioned condition. Because of this undesired situation, cabinet base plate must be modified to change its vibrational characteristics.

5.2. Modification of Vibration Characteristics

Modifications may be carried out by changing the following properties:

- Material properties of the cabinet base plate,
- Bitumen Based Damping Sheets,
- Thickness of cabinet base plate,
- Number and geometries of hole existing on cabinet base plate.
- Stiffeners can be added to cabinet base plate

The easiest solution is decreasing the hole number because of the cost. Design modification is checked for natural frequencies theoretically, again by using finite element model recreated in ABAQUS. On the other hand, acoustical improvements is observed experimentally.

Cabinet base plates with two and three long holes shown in Figure 5.1 are the parts in the problem after and before modification, respectively.

The natural frequencies of cabinet base plate with two and three holes are given in Table 5.1. It is seen from Table 5.1 that second natural frequency is shifted from 49.8 Hz to 53.8 Hz for the cabinet base plate with two holes. Since the excitation is at 50 Hz, cabinet base plate with two hole is in safe condition for this excitation.

Figure 5.1. Cabinet base plates with two and three long holes

Table 5.1 Theoretical Modal Analysis Frequencies (Hz)

Natural frequencies	Cabinet base plate with two holes	Cabinet base plate with three holes
f_1	40.3	43.6
f_2	46.8	49.8
f_3	53.8	62.6
f_4	67.2	81.8
f_5	87.3	88.6
f_6	95.1	105.5

5.3. Acoustic Improvement

While the sound power level was 48.3 dB(A) measured on 10 samples before modification, it was reduced to 45.6 dB(A) after modification done on 10 samples. To reach this results, several design modifications have been done.

Reverberation room used to test the sound power level by using 6 microphones in diffuse field is shown in Figure 5.2.

Figure 5.2 Reverberation room to test the sound power level

CHAPTER 6

CONCLUSIONS

In introduction, vibration of plates with holes by Finite Element Method and modal testing techniques are surveyed. As theoretical background, the following details are given:

- Finite Element Method for Vibration Analysis of Plates,
- Theoretical Modal Analysis,
- Experimental Modal Analysis. Vibration analysis of cabinet base plate are presented as Natural Frequencies and Mode Shapes by Finite Element Method,

As a result,

- Modal Test and Modal Analysis results validation are performed,
- Design improvement for acoustic resonance is performed,
- Declarations of Sound Power Levels for all refrigerator models (Width 60cm) rearranged with new acoustic improvement for costumers.

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