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NOISE CONTROL FOR A BETTER ENVIRONMENT

# An Experimental Modal Analysis Methodology for the Vibro-Acoustical Identification of Coupled Enclosures

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**ABSTRACT**

Structures of machinery with practical importance, such as home appliances or transportation vehicles, can be considered elastic enclosures with acoustically coupled inner spaces and any noise control procedure to be applied requires a versatile approach for their vibro-acoustical identification. A combination of methods is developed for this purpose and a domestic type refrigerator, a comparatively complex yet typical structure, is used as a case study. The method is composed of a novel experimental modal analysis technique for the determination of cavity modes that uses a set of calibrated, low cost loudspeakers as volume velocity sources, whereas, a FEM approach is used for the verification. A special emphasis is given to the effect of temperature on the acoustical modal behavior of acoustical cavities. Furthermore, the noise transmission characteristics of the structure have been investigated by using near field sound intensity measurement techniques. Finally, possible transmission paths between the sources and receiving points have also been identified in accordance with low noise design requirements. These experimental methods are integrated into a coherent procedure of measurements that can be applied to any enclosure structure of practical importance, operating at frequency ranges below 1000 Hz without any limitation on the geometry.

**Keywords:** Noise, Enclosure, Measurement
**I-INCE Classification of Subject Number:** 32

## 1. INTRODUCTION

A noise control engineer working in consumers product industry is usually required to reduce the sound level of the product with a minimum or preferably no additional cost and/or design alterations. Hence, even for the most optimistic cases, acoustical enclosure principles as formulated in the waste literature of noise control engineering are hardly applicable as they almost always start directly from the design stage. (1) A principle such as “never enclose more volume than necessary” is almost a fundamental rule for such a designer, yet, what if due to some basic requirements of the machinery there will be large cavities to be part of this already existing enclosure? Furthermore, if it is not just one, but many and coupled? Therefore, the generalized problem under these conditions can be defined “given the structure, how can it be turned it into an effective enclosure?” Hence, an efficient system identification procedure becomes mandatory, since task is usually an iterative process until a directive or market goal achieved without, or at most with very small amount of additional cost.

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Transportation vehicles, such as automobiles, trains or airplanes are good examples with their interior volumes composed of cabinets for prime movers and for passengers. However, the passenger comfort in this case is one of the priorities of the total design, and noise control is included right at the starting point, and the other one being the strict environmental regulations and related standards.

On the other hand, there is a considerable amount of machinery, very effective in our daily lives, where "cavities" are there, as they are in need. A refrigerator, domestic or commercial is one good example of such a product. A typical no-frost refrigerator is composed of at least two main cabinets, one with an evaporator unit and usually an air-moving device at sub-zero temperatures, and a larger one for fresh-food at slightly above zero degree Celsius, and the cooling air is supplied from the upper deep-freezer section, in many cases by another air-moving device. Finally, there is a compressor-electric motor unit outside the main compartments but still an integral part of the whole assembly.

One interesting aspect of the problem is that the cavities of such an enclosure then assume different temperatures from each other, and also different from the ambient. This may look odd at first glance, however, for almost all enclosures for most of the prime movers, this could just be the case, hence, no-cavities is indeed very good advice.

Design of an effective acoustical enclosure requires a fundamental and if necessary a detailed analysis of the structure, hence, a hybrid method for the vibro-acoustical identification of coupled enclosures is developed and Experimental Modal Analysis (EMA) methodology is at the core of this methodology, which from some other perspective, can also be regarded as Operational Modal Analysis (OMA). The methodology will be presented for a case study of a refrigerator cabinet and then, the procedure will be used to interpret the measured Insertion Loss characteristics. Finally, an experimental noise path analysis presented for the predictive ability of the methodology.

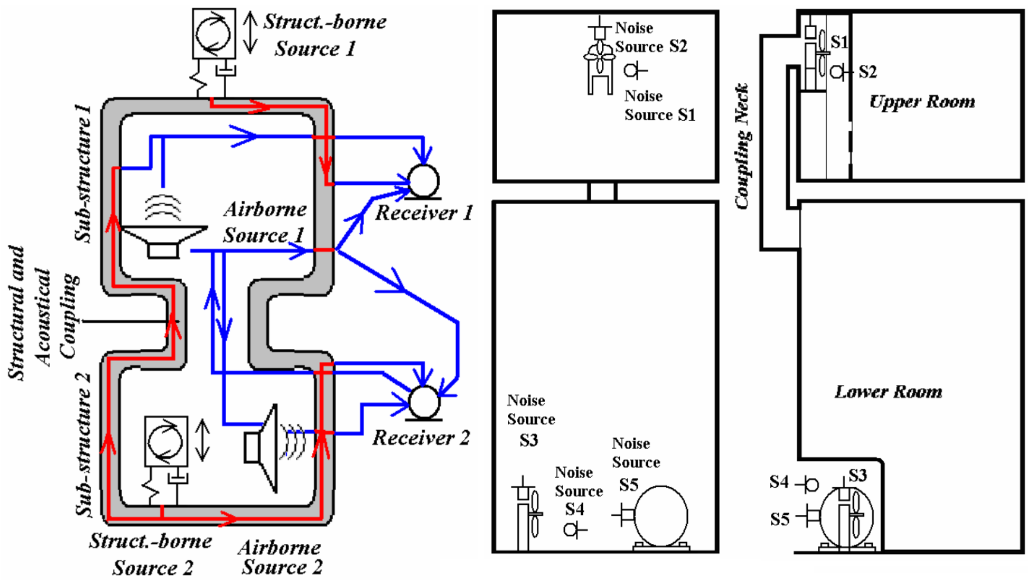

*Figure 1 – A Generalized Enclosure and Actual Enclosure with Simplified Geometry*

## 2. METHODOLOGY

### 2.1 Modeling Enclosure Cavities and Approaches of Analysis

A complex acoustical enclosure with two coupled acoustical cavities is depicted in Figure 1, both in its most generalized form and as a special case subject this study.

Acoustically, the simplest geometry for such an enclosure is two cavities connected with another neck element in between which serves as an air passage. The structure in this case study is a domestic refrigerator. The neck element served as a passage for the cooled air after forced by a small fan over the evaporator surfaces, given as noise source S1 in the Figure. Three other sources of noise S2-4 are also present in their most simplified versions, as the compressor, cooling fan and due to the flow of the coolant gasses. However, apart from the evaporator fan (S1), rest of the noise sources are either embedded into the cabinet, or outside. Hence, for cavity excitation the location of S1 is crucial for this particular case and for finding the cavity resonance modes of interest.

## 2.2 Analytical and FEM Models

As first things first, the enclosure volume is segmented in rectangular geometries as fine as possible and lumped acoustical elements are identified with the corresponding acoustical compliance and mass.(Figure 1) In this conjecture, it is only the neck volume that worth examining as a mass element, where the neck and the rest of the volumes have dominantly compliance characteristics.

Next to lumped parameter modelling, the same simplified cavity geometries are used for the exact solution of wave equation for rigid, parallel wall conditions in terms basic axial modes and frequencies.

For the Finite Element Model (FEM), acoustic cavities are modeled and meshed in IDEAS Solid Modeling Toolbox. The FEM analysis of the system is conducted by means of a commercial FEM solver (SYSNOISE). Linear-order (four-nodes), isoparametric, tetrahedral elements are chosen for meshing. The structure is compound of two sub-sections. The lower room is the first sub-section and, the rest of the cavity, that is the coupling neck and the upper cold sections, builds up the second subsection. Solutions of the software are exported in Universal File Format (UFF) to MATLAB. Consequently, data processing and visualization are done in the latter environment. The routines developed can be generalized to any geometry and solver provided that input data are created in UFF's, which is the commercial standard in Computer Aided Engineering (CAE) market.

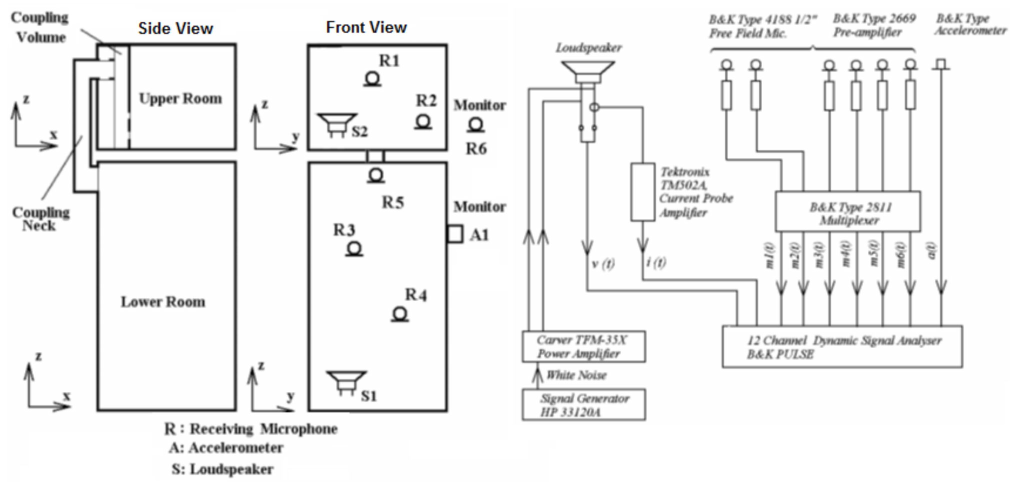


*Figure 2 – Microphone and Source Locations and Testing Equipment Set-up*

## 2.3 Experimental Acoustical Modal Analysis

The crucial part of the method is the selection of the excitation or the source for obtaining the necessary frequency response functions. For this purpose, apart from commercially available standard sources, a collection of cost effective mid-range speakers are tested and considerable number of them found for fair use, provided that they have reasonable phase characteristics and no adverse non-linear behavior. In such a case, with the linear, time invariant assumption a speaker as a transducer, the reciprocity holds valid as:

$$\left(\frac{Q}{V}\right)_{P=0} = \left(\frac{i}{p}\right)_{Q=0} \tag{1}$$

*where, Q is the volume velocity and p is the pressure at the diaphragm surface, and I is the current and V is the voltage driving the speaker. In a reactive sound field, as the case of the field inside a sealed enclosure cavity, the current fed into a speaker I, is correlated, or in constant phase relation with the sound pressure p, directly at the vicinity of the speaker diaphragm surface*.

Hence, the current signal served as a fair enough reference signal for Operative Modal Analysis purposes, as well as Experimental Modal Analysis, in case an absolute phase and magnitude relation holds. In this case study, many speakers with various properties are tested and quite a number of them are quite appropriate enough and the necessary transfer functions between the excitation and the response points are approximated as;

$$H_{M_k S_j} \sim \frac{G_{p_k i_j}}{G_{ii_j}} \tag{2}$$

*where, H is the transfer function, k,j are location indices for M receiving and S driving points, G is measured correlation functions p is sound pressure and "i" is the current fed to the speaker terminals*.

The measurement set-up, together with the instrumentation details are shown in Figure 2. It may worth mention that the voltage supplied to speakers is 2 Vpp, white noise in most of the successful measurements.

### 2.4 Transmission Characteristics

For an acoustical performance measure, the power based Insertion Loss, *IL* is defined as;

$$IL = L_{w0} - L_{w1} = 10log\left(\frac{W_0}{W_1}\right) \tag{3}$$

where $W_0$ and $W_1$ are sound power of a reference source at free field and inside the enclosure, $L_w$ is the related sound power levels. (2) The measurement of this parameter can be quite practically done by using a standard source, or measuring the sound power of some convenient sources of sound in free field and then repeating the measurement when they are inside the enclosure. On the other, a near-field acoustical intensity measurement gives a much detailed view of the acoustically transmitted and radiated field as depicted in Figure 3. A near-field measurement can also supply the particle velocity near the boundaries of the enclosure which can be further exploited for the radiation efficiency of the enclosure.

Finally, set-up for a noise transfer path investigation is given in Figure 4. For this purpose, a calibrated source as close as possible to an omnidirectional source is used for reciprocal measurements (microphone inside source points, sound source outside receiver points). The enveloping measurement surface and receiver points are constructed in accordance with the IEC 704, Sound Power Measurement Standard for Home Appliances, which eventually gives a perfect insight on performance of the cabinet in accordance with directives.

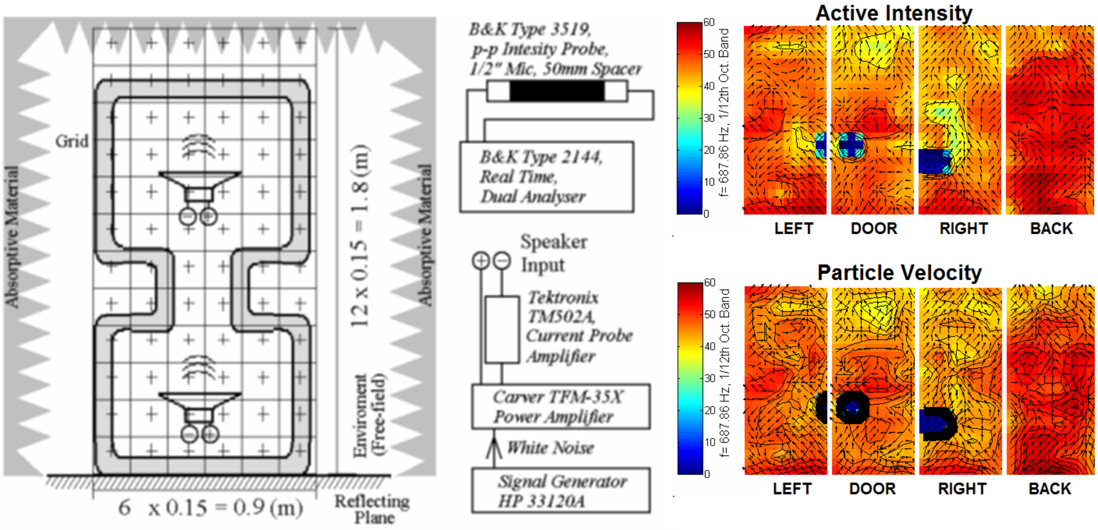


*Figure 3 – Experimental Set-up for IL Determination by means of Near-Field Acoustical Intensity Measurements and a Typical Result for a Resonance(F=688Hz)*

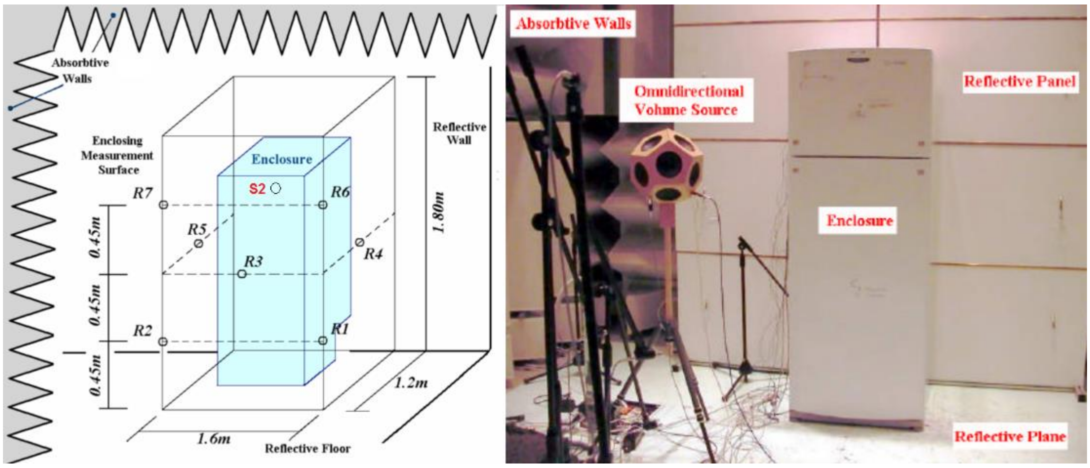


*Figure 4 – Noise Transfer Path Measurement Set-up*

## 3. RESULTS

### 3.1 Analytic Models

The resonance frequencies obtained from the lumped parameter and wave equation solutions (for rectangular shapes with rigid, parallel wall geometries) are consolidated in Figure 5. Furthermore, the FEM solutions are also superimposed and their grouping is done according to the cases used in FEM analysis. Labels indicating “Upper Only” and “Lower Only” for the upper and lower cavities respectively, and “Coupled” for the fully coupled enclosure inside volumes.

The lumped parameter solutions essentially indicates the frequencies where the system is behaving like a Helmholtz resonator and as expected, these frequencies are quite low, and predicted also well with the FEM model, with the exception of the coupling neck and small recession in the coupling are as this part is incorporated in the upper volume in FEM analysis. Apart from this, a very good picture of the cavity behavior has been obtained.

For a better insight, the cavity modal behavior is given for some of the resonance frequencies in Figure 6. For example, at 13 Hz and 127Hz, which are also calculated from the lumped parameter model, the cavity system acts like a Helmholtz resonator with different boundary and elements in each of these conditions.

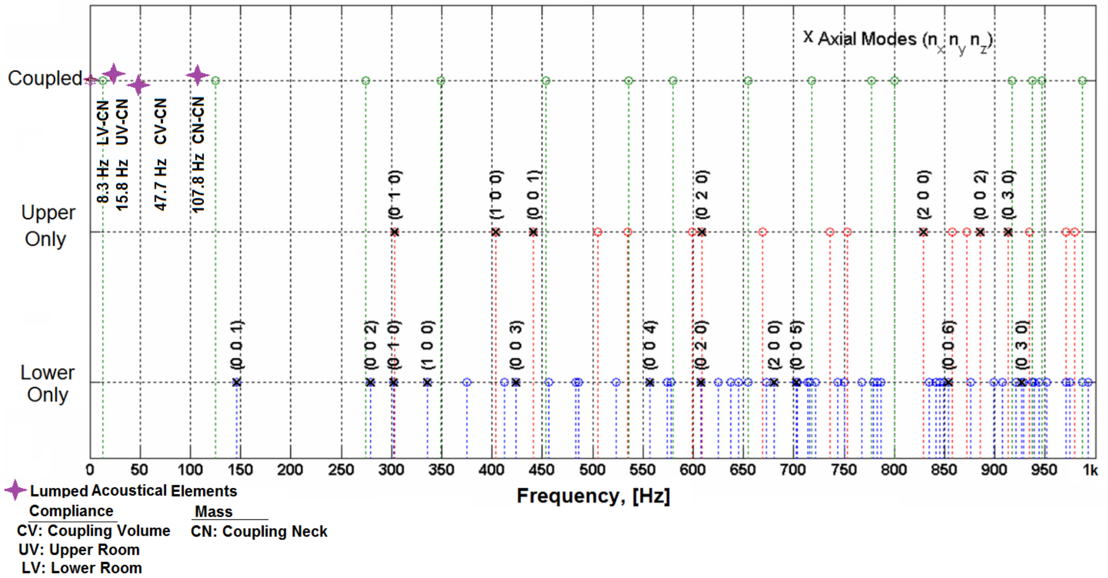


*Figure 5 – FEM Solutions Compared with Modes for Simplified Geometry*

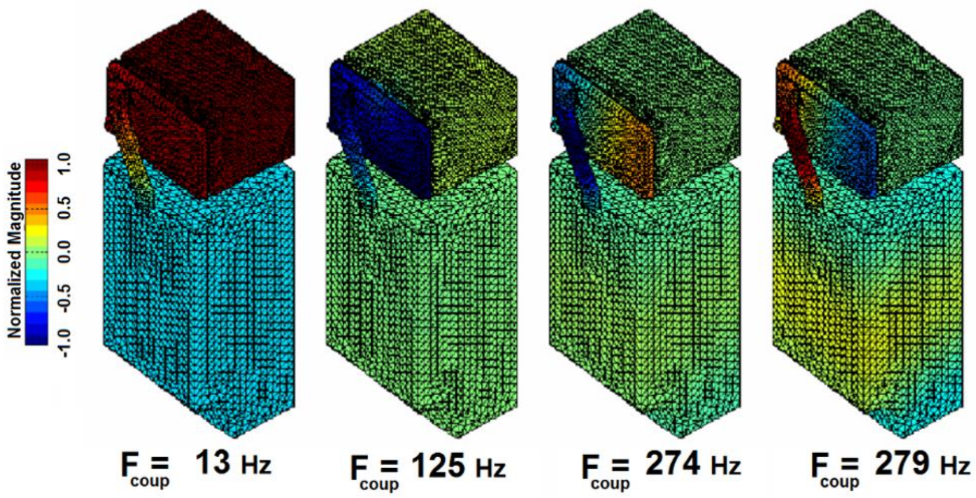


*Figure 6 – FEM Solutions for the First Modal Frequencies*

### 3.2 Experimental Acoustical Modal Analysis

Prior to EMA results, the effect of cavity temperature on the measurements of frequency response functions are presented in Figure 7. The measurements are started at $4^0$C (=277K) and kept going as the enclosure warms up to the ambient temperature at $21^0$C (=294K). As expected there is a substantial shift in resonance frequencies, noting that it is the modal frequency that changes due to the change in the speed of sound, but not the corresponding wave shape and this is just another operational way to detect these modal resonances.

In Figure 8, typical transfer functions for EMA measurements are given as bode plots, for two of the upper volume microphone pressure signals as outputs and the corresponding speaker current signal as input. After a peak detection algorithm, the possible resonances are indicated with black circles. These possible resonance frequencies are further checked by means of Nyquist plots, at around these frequencies and some typical graphs are presented in Figure 9.

The reason for this invaluable check is that a so-called "peak" point in a measured Frequency Response function does not necessarily correspond to a resonance. A smooth phase transition in +/180$^0$ is one best indicator for this, however, still due to overlapping resonance at higher frequencies these criteria also need careful analysis to conclude. Yet, cross-checking for these indicators at various positions for the microphones and sources, has given some quite robust results. For a practical representation, these EMA results are incorporated with the analytical and FEM analysis results of Figure 5, and they were presented altogether in a "enclosure region vs frequency" format as in Figure 10

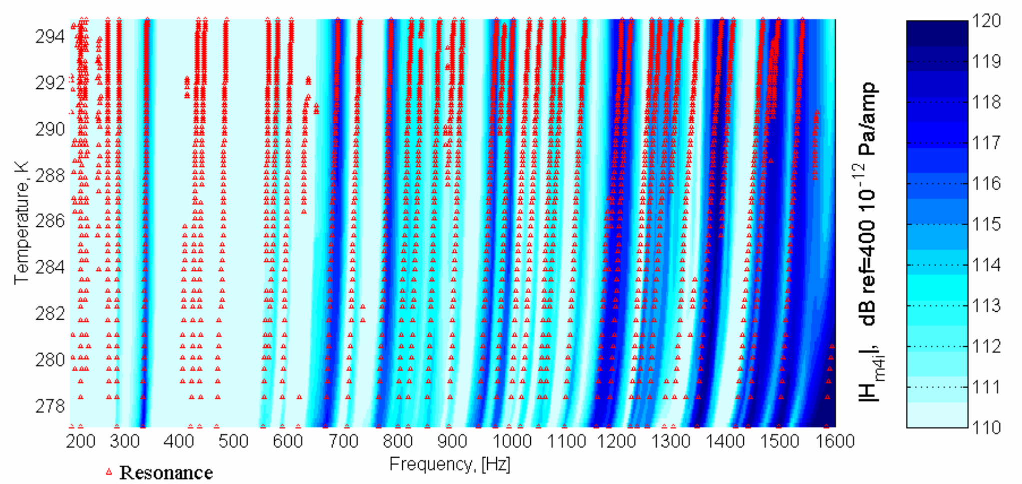


*Figure 7 – Effect of Temperature on Transfer Functions*

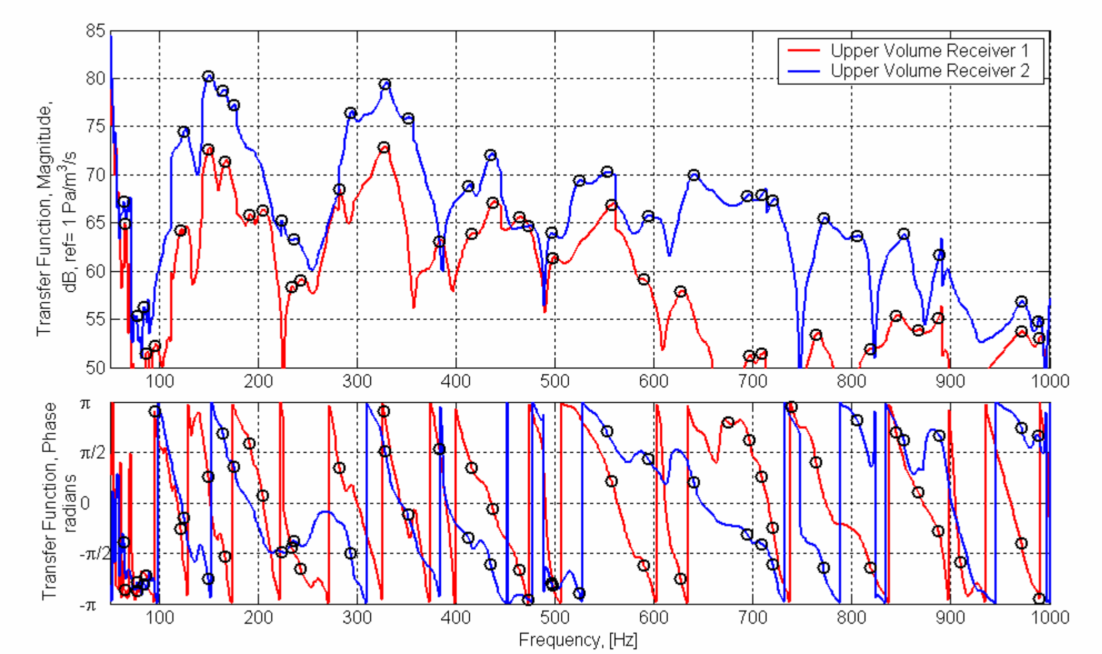


*Figure 8 – Typical Transfer Functions Measured for EMA, Bode Plot, Two of the Upper Volume Microphones, peak points indicated with black circles*

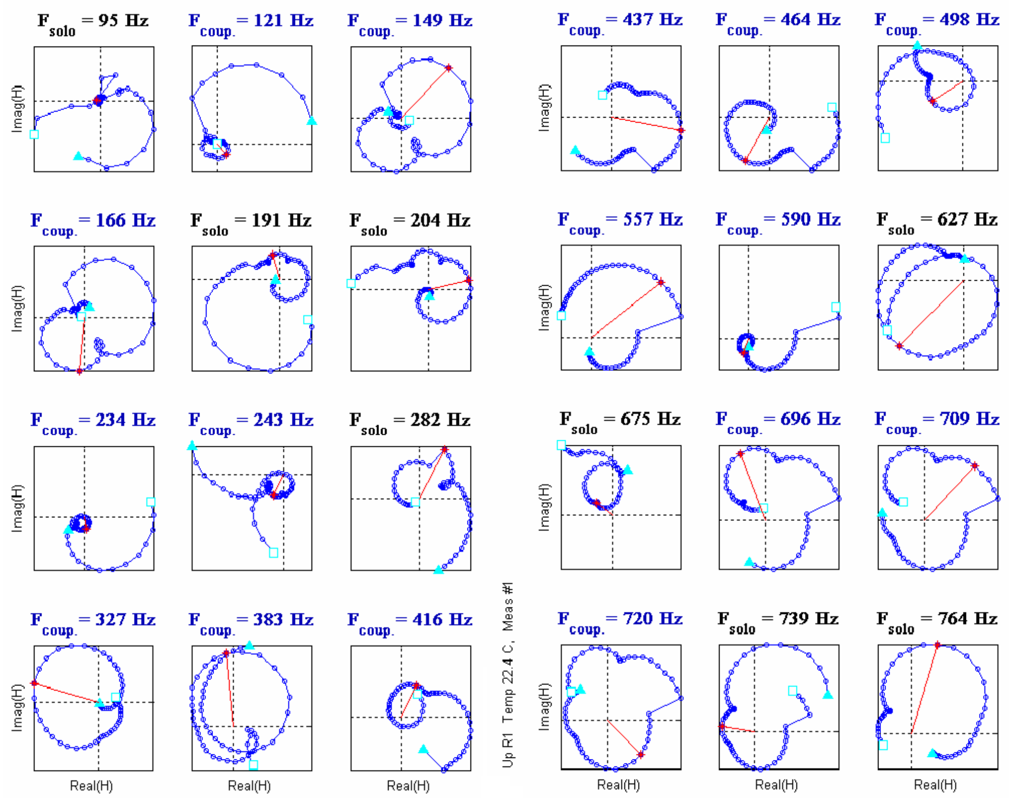

*Figure 9 – Nyquist Plots at Arround Possible Resonances of Transfer Functions (EMA), 2Hz resolution, red arrow: peak frequency, cyan square and triangle: start and end of the frequency window around the peak frequency*

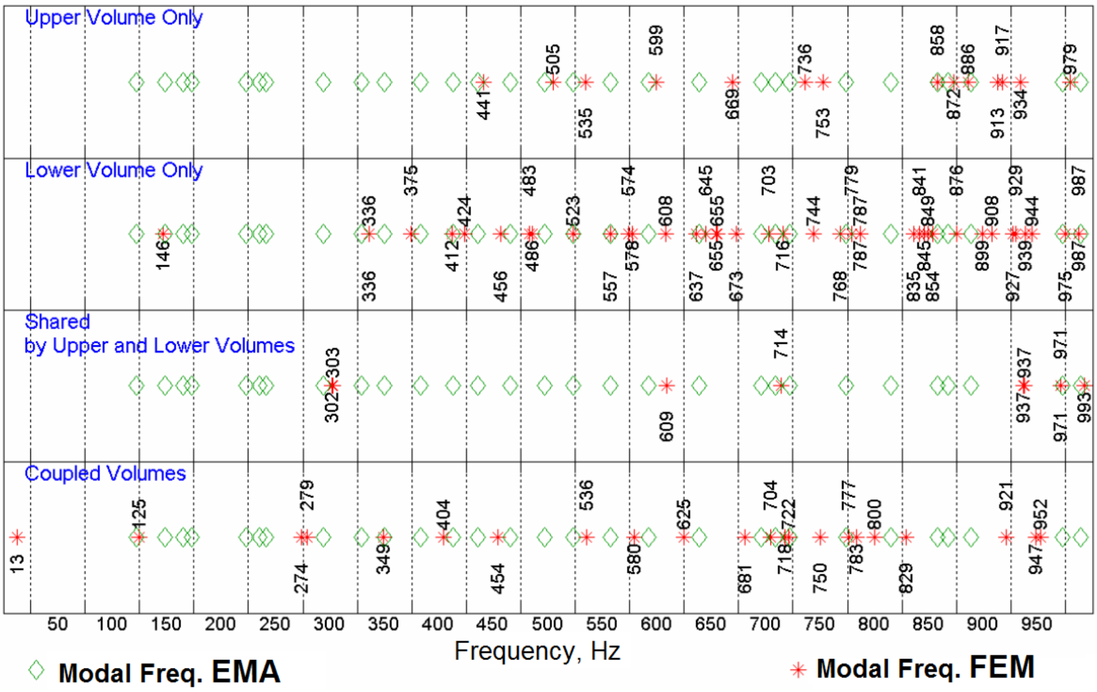

*Figure 10 – Final Comparison of Resonance Frequencies Obtained from FEM and EAM Methods*

## 4. DISCUSSION

The first and fundamental indicator for the performance of an enclosure is its noise transmission characteristics and the methodology applied has already been given for this study as Insertion Loss (IL) measurements. In Figure 11, one such measurement is discussed within the overall scope of the study presented so far.

Theoretically, a typical noise enclosure acts as low pass acoustical filter, that is, under a frequency of $F_{low}$ the enclosure is not effective at all and over a frequency of $F_{high}$ the enclosure has almost a flat response with a positive IL value. These two regions are named as “small enclosure” and “large enclosure” regions, respectively, and “intermediate” in between these frequencies. For the case study, these three regions are readily observable as $F_{low} < 200$ Hz and $F_{high} > 900$ Hz. In between, the enclosure IL is not only low but giving quite a number of deeps, that may cause severe noise problems if the sources enclosed have radiation at one of these trouble frequencies. Therefore, it would be invaluable to identify the causes of these mechanisms and relating them to the properties of the enclosure.

For the sake of convenience, the consolidated frequencies given in Figure 10, for the coupled volumes, that is the entire enclosure inside, are also given down below the IL frequency response plot. Notice that, due to analysis methodology, the IL values are obtained from 1/12 octave band, near-field acoustic intensity measurements, whereas, EMA measurements are conducted in FFT domain with 2Hz resolution, hence the frequency scale difference.

For convenience, IL deeps are related schematically to the resonance frequency clusters with indicators in red.

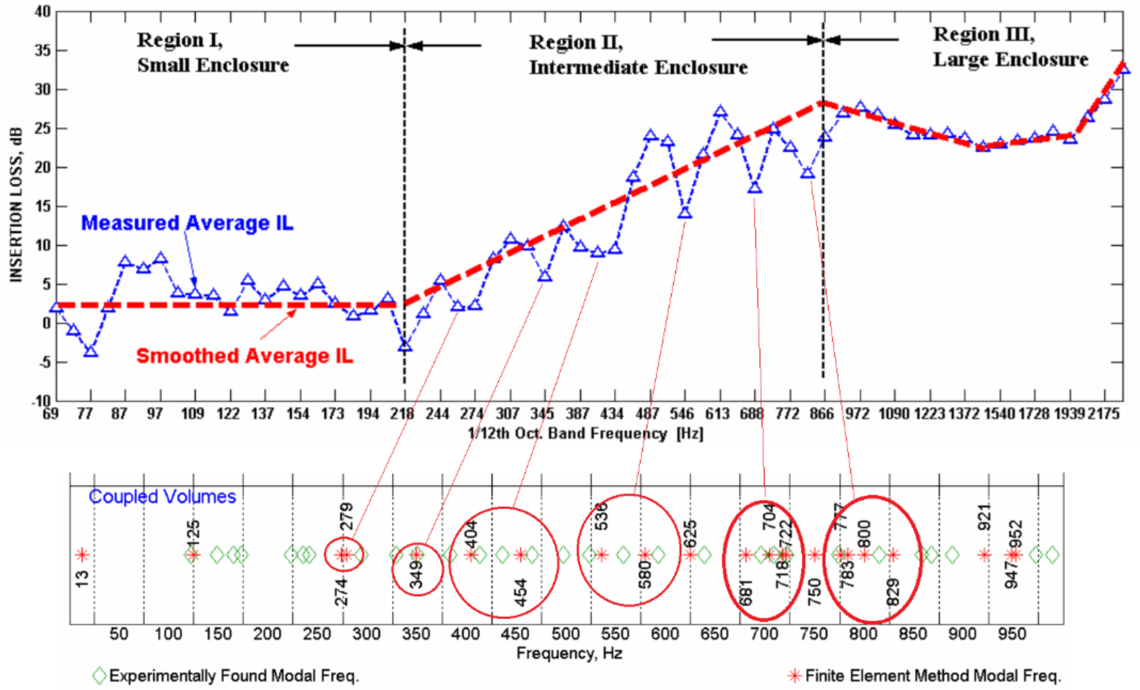


*Figure 11 – Insertion Loss (IL) for the Enclosure and Related Enclosure Cavity Resonances*

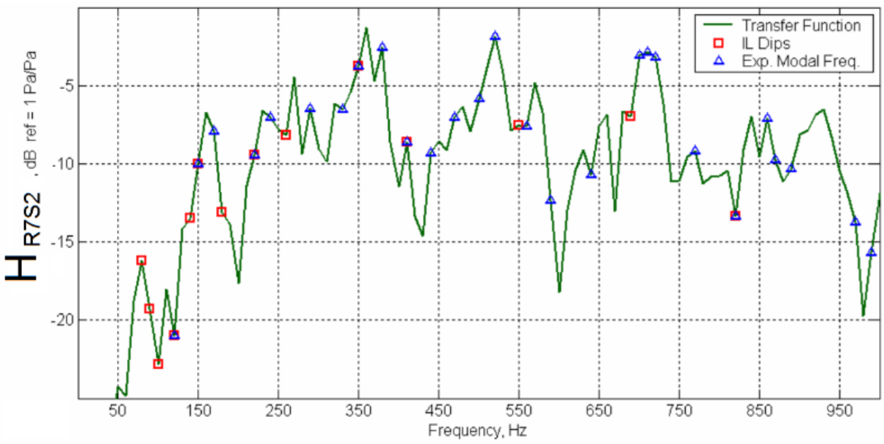


*Figure 12 – Noise Transfer Path, Source 2, Receiver 7, Reciprocal Measurement*

On the other hand, Figure 12 gives just one of these noise transfer functions, measured for a possible source position and receiver positions obtained reciprocally as explained before. S2 is a source point at the upper room and R7 is a standard measurement point outside the enclosure, almost at the same height (typical ear height, Figure 4) The IL deeps and EMA resonance frequencies are also overlapped on the graph. For this particular transfer path, some very practical conclusion can be made as, for example, modifying the enclosure cavities will be very effective for a source at around this position if it has some major radiation at 650-750 Hz region, whereas, will be no effect it operates for example at 800-850 Hz.

## 5. CONCLUSIONS

A study for the identification of cavity acoustic resonance frequencies and their effect on a possible enclosure transmission characteristics is carried out. The methodology is hybrid, combining basic analytic tools with FEM and Experimental Modal Analysis techniques. As a practical application, IL performance of a domestic refrigerator cabinet is investigated thoroughly and the effectiveness of the methodology is approved successfully.

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