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SOUND FIELD NUMERICAL ANALYSIS OF A HERMETIC COMPRESSOR CAVITY

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ABSTRACT

The intense sound field in the cavity of hermetic compressors represents one of the major shell excitations and its consequent noise radiation. This field is mainly excited by the inlet muffler pulsation as well as by the block and electric motor surfaces radiation. This work describes a numerical analysis of the cavity sound field considering different geometric simplifications in the block and electric motor surfaces, numeric solutions and the coupling to the shell. Results were experimentally validated, indicating reasonably good agreement. This model has shown to be adequate for the parameter influence analysis of the sound energy transmission to the shell, and its radiation to the external environment.

Keywords: *Hermetic Compressor; Noise; Vibration.*

1. INTRODUCTION

Hermetic compressors are largely used in domestic appliances and are responsible for a significant contribution to the overall noise radiation. Shell vibrations are excited by the vibration energy generated in the compressor / electric motor assembly and transmitted through the springs and discharge tube. The strong sound field in the cavity represents also an important shell vibration excitation mechanism. The cavity sound field is mainly excited by the inlet muffler pulsation, as well as by the compressor / electric motor and moving parts surface radiation. This shows the importance of the cavity as an energy transmission path to the shell. Its acoustical behavior must therefore be fully understood in the development of new compressors. This work describes a numerical analysis for the prediction of the sound field in the cavity. Results for different numerical models are presented and the influence of geometry details is discussed.

2. SIMPLE GEOMETRY CAVITY

Previous attempts to modeling the sound field in the cavity have shown the strong influence of the geometry details upon the accuracy of the numerical results. This is particularly important at higher frequencies as the acoustic wavelength becomes comparable to the cavity irregularities. The detailed geometry model preparation and the numeric solution for the frequency range of interest can be very time consuming for a given tolerance requirement. However, the numeric model and the boundary conditions may also present uncertainties in the results. In order to investigate these effects, an analysis was initially carried out on a simple geometry cavity. The analysis was later carried out on a real compressor cavity. All numeric models results presented in this work were obtained with software Sysnoise 5.6.

2.1 Cylindrical cavity

A cylinder made of steel, having 160mm in internal diameter and 140mm length was used for this first analysis. The thickness of the cylinder wall was 30mm to avoid the influence of background noise as well as its coupling with the internal sound field. Disks with different thicknesses were prepared. They were tightly bolted to the ends of the cylinder to simulate a clamping condition. It was then possible to excite one disk at one end and analyze the disk response at the opposite end, excited by the sound field in the cavity.

In a first analysis one of the disks was chosen 15mm thick to simulate the behavior of a rigid wall. The second disk was 3mm and was excited by a point force. It was experimentally observed that its vibration response is not significantly affected by the presence of the cavity sound field. This led to the conclusion that the numerical analysis does not require a fluid-structure coupling, i.e., the disk vibration can be obtained without the coupling effects to the sound field. Sound field can therefore be analyzed considering the disk vibration distribution as a boundary condition only.

The cavity sound field was modeled by FEM and the response calculated by the Modal Superposition approach. Damping values were experimentally obtained by the decay rate method. Pressure transducers were placed at the cylinder wall for the sound pressure signal acquisition. Disk vibration response was measured at 65 points by a small accelerometer. These values were used in the modal expansion approach for the calculation of the velocity response at all mesh nodes of the disk. This set of vibration data was then used as excitation in the numeric model of the cavity sound field.
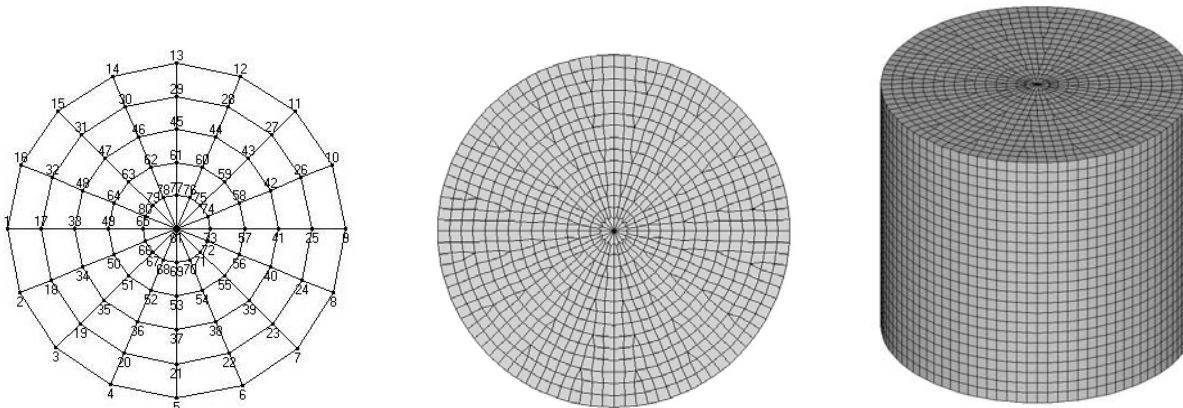

Figure 1 – Surface velocity measurement grid (left), disk mesh (center) and cylindrical cavity mesh (right).

Figure 2 shows a typical comparison between measured and calculated discrete frequency sound pressure results. Good agreement is observed throughout the spectrum. The cavity model used 23000 elements yielding to at least 6 elements per wavelength and the disk was represented by 950 shell elements.
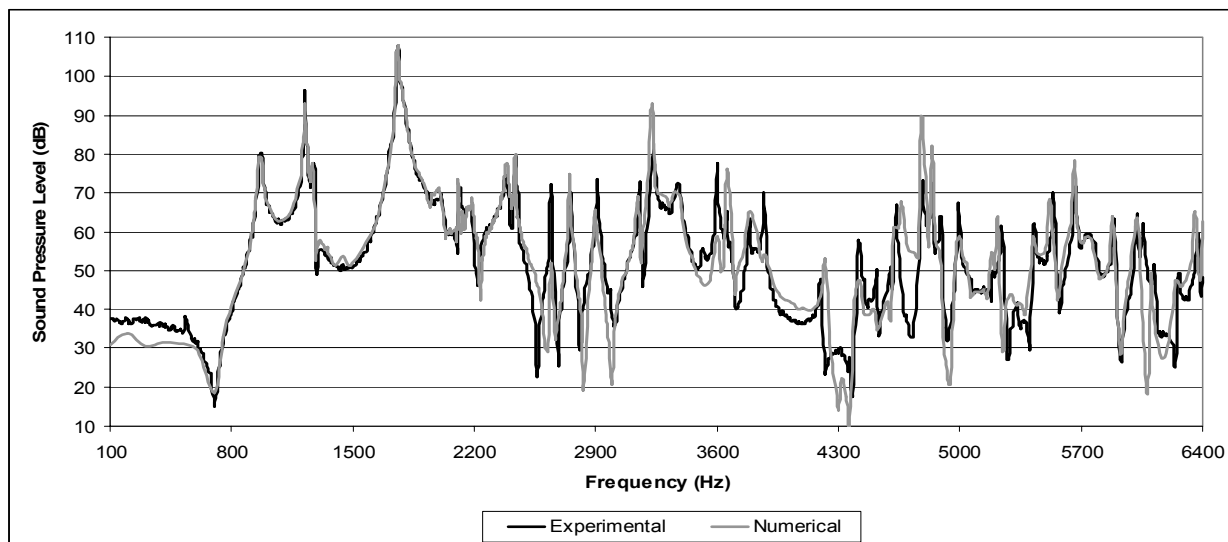

Figure 2 – Comparison between experimental and numerical sound pressure results for a simple geometry cavity.

2.2 – Disk vibration response to cavity excitation

The same experimental set up was used for the fluid-structure interaction analysis, i.e., disk vibration response excited by the cavity sound field. A 3mm thick disk made of steel was placed at one end of the cylinder, as described in item 2.1. Vibrations were excited by a small shaker and the response measured at 65 points. These data were used for the excitation of the cavity sound field. The cavity, this time, was coupled to a second disk placed at the opposite side of the cylinder. In order to obtain clear experimental vibration signals the second disk (made of steel) was chosen 0.75mm thick and the response measured by a laser doppler vibrometer, thus avoiding mass loading and damping addition effects.

Shell elements were used in the FE representation of the thinner disk at the opposite side of the exciting disk (3mm). Cavity fluid elements were linked to the disk structural elements. The thinner disk response was also obtained by the modal superposition approach. Figure 3 shows a comparison between numerical and experimental results, indicating good agreement for frequencies above l kHz. Large discrepancies are expected at low frequencies since the cavity has no resonances in this region. In order to verify this, the cavity was filled with acoustic material and the response of the thinner disk measured again. A significant reduction in the response at high frequencies was observed. This indicates that the disk is excited mainly by the sound field and vibrations transmitted trough the cylinder walls are negligible.

This procedure is considered adequate for the representation of the sound energy transmission through the cavity to the shell, as is the case in hermetic compressors. Block and electric motor surface vibrations can be used as velocity boundary condition in the cavity model, and this used for shell response determination. However, detailed geometry and relative vibration informations at compressor operation conditions are required. The effect of geometry will be discussed in the following section.
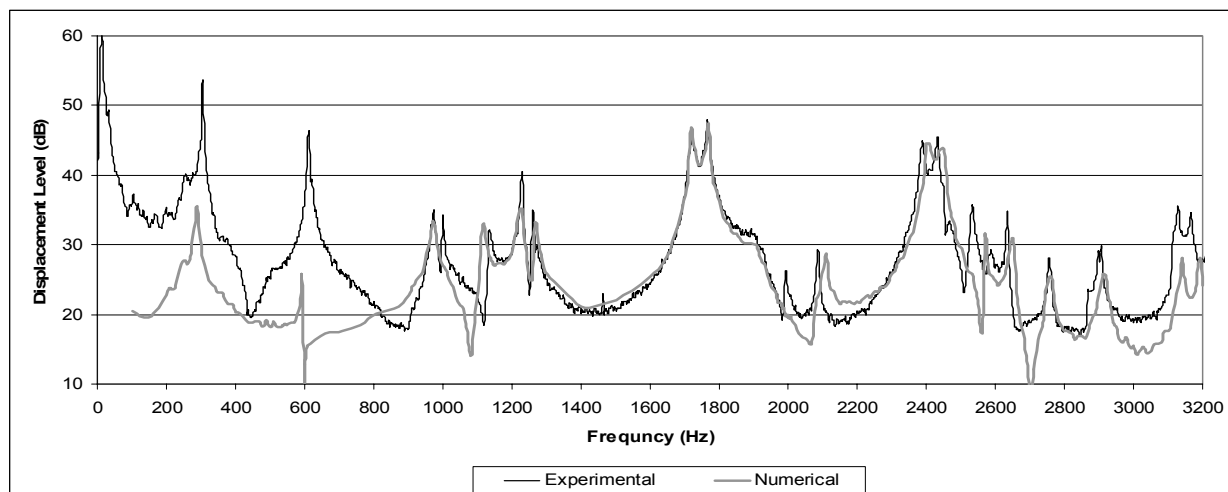
Figure 3 – Comparison between experimental and numerical vibration response of 0.75mm steel disk excited by the sound field in the cavity.

3. SOUND FIELD IN A REAL COMPRESSOR CAVITY

The sound field analysis on a real compressor shell was carried out considering initially a simple geometry representation for the block and electric motor. This aims at a deeper understanding about the sensibility of geometry details upon the calculated response accuracy. In these analyses cavities were modeled by BEM and solved by Padé expansion method. First of all the sound field was modeled for an empty shell, decoupled from the shell vibrations. The cavity was modeled with 7500 elements type TRIA3 which allows analyses up to 10 kHz. Unitary values for the particle velocity were prescribed in a region of elements at the top of the cavity, as shown in figure 4, and the sound pressure response was calculated at several points.
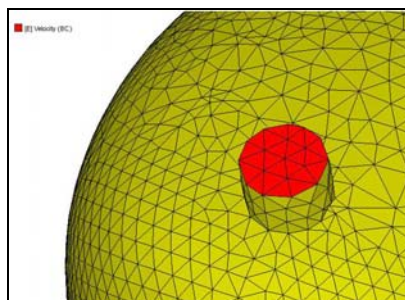

Figure 4 – Particle velocity unitary values boundary condition.

An experiment was set up for validating this model, as shown in figure 5. The particle velocity was measured by means of a Microflown transducer and the pressure response measured at the indicated points. Figure 6 shows the sound pressure frequency response function at one point relative to the prescribed unit particle velocity values. A reasonable good agreement is observed for frequencies up to 10kHz, for the cavity filled with air. Figure 7 shows a comparison in 1/3 octave bands. One notices an average difference of about 3 dB, in 1/3 octave values, however in some bands the difference can be as high as 9 dB.

Particle velocity transducer
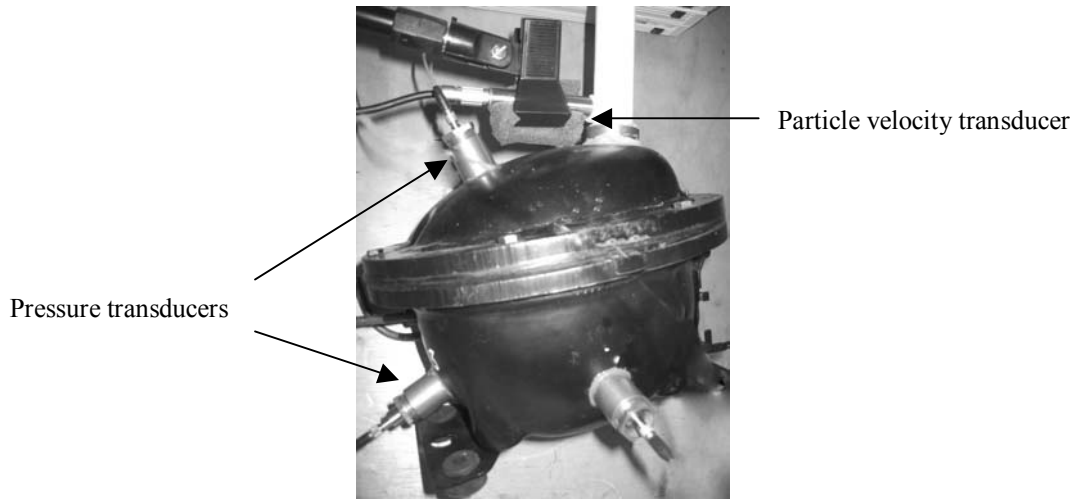
Pressure transducers
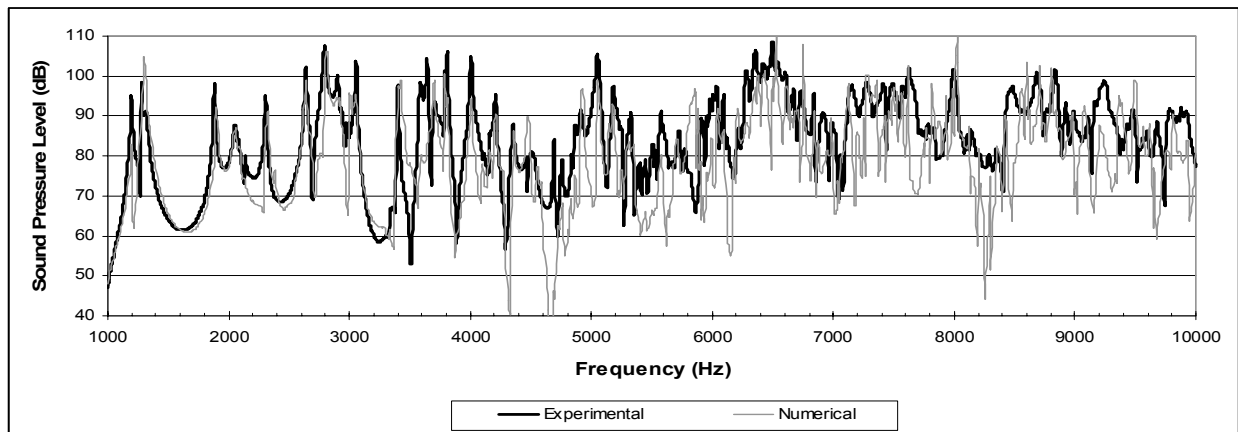
Figure 5 - Experimental set up

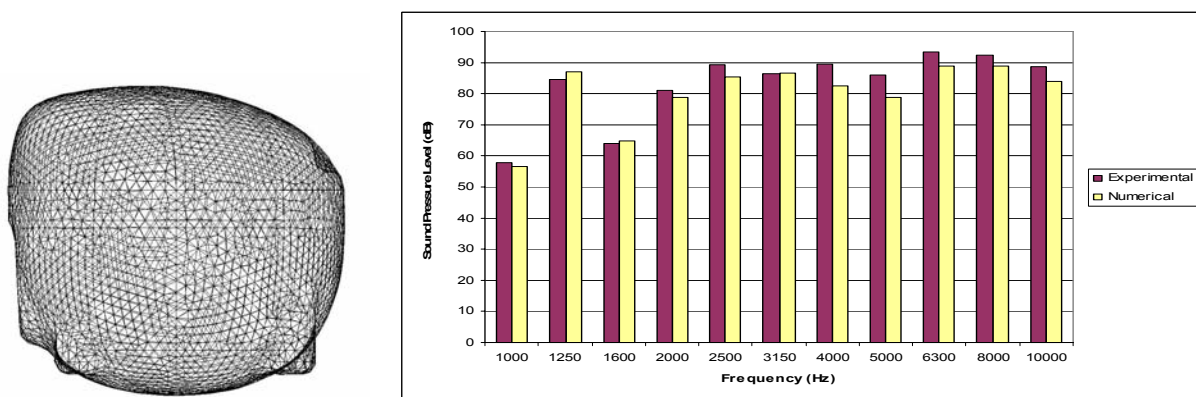Figure 6 – Comparison between numerical and experimental sound pressure results for an empty compressor cavity.

Figure 7 - Empty cavity model; sound pressure response comparison in 1/3 octave bands.

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In a second experiment the stator of the electric motor was represented by a similar simple geometry component made of nylon supported on four small solid steel cylinders, representing the springs, as shown in figure 8. This figure also shows a comparison between numerical and experimental results in 1/3 octave bands. The average difference is also of the order of 3 dB, with maximum values of 9.3 dB.
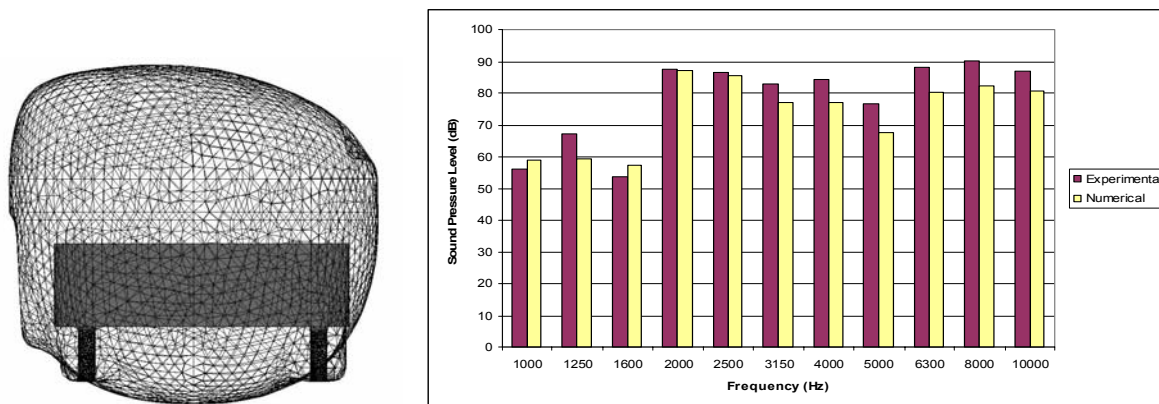

Figure 8 – Cavity model containing a simple geometry stator; sound pressure response comparison in 1/3 octave bands.

In a third experiment the effects of copper wires winding were represented also in a simple geometry by nylon pieces placed above and under the stator as shown in figure 9. A comparison between numerical and experimental results indicates an averaged 3dB difference and 10 dB maximum difference values. One notices that larger differences are consistently observed at 4 kHz and 5 kHz bands. This is attributed to uncertainties in the experimental procedure.
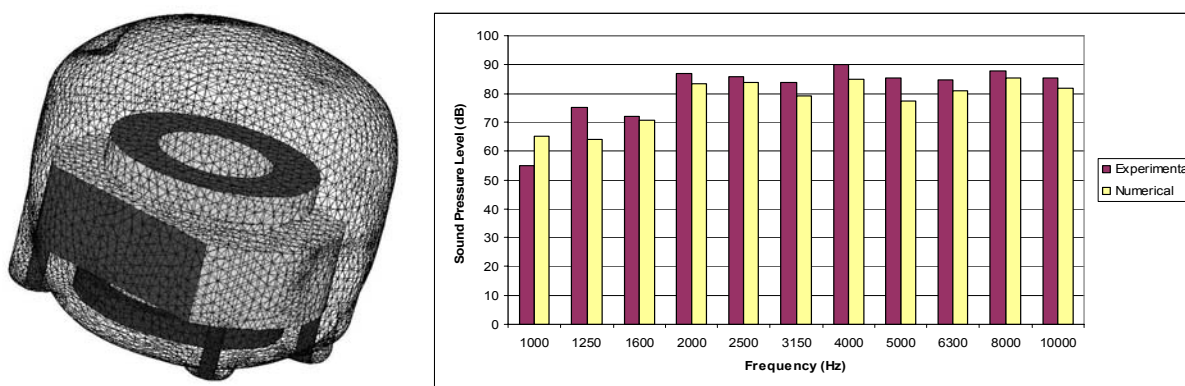

Figure 9 – Cavity model representation containing stator with copper wires; sound pressure results comparison in 1/3 octave bands.

In a last experiment the cavity of a compressor containing the real block and electric motor assembly was excited and the sound pressure measured at several points. A numeric model considered the real geometry with simplifications to reduce the number of boundary elements which would be required to fully represent all the geometry details. A comparison between numeric and experimental results indicated similar tendencies as in the previews analyses with averaged errors of the order of 3dB and maximum of about 10dB.

4. CONCLUSIONS

Numeric models for simple geometry cavities have shown that the fluid-structure interaction with the dynamic behavior of a 3mm thick disk can be neglected. It can therefore be considered as a rigid boundary from the cavity point of view. The excitation of the sound field can be represented by a relatively small number of surface velocity measurement points. Extension to all other surface model nodes can be made by the modal superposition method. This requires a reliable structural dynamic model. In case of block and electric motor assembly a structural model may present significant inaccuracies. Therefore, a careful and detailed surface velocity measurement at operation condition is recommended.

Simple geometry representation for block and electric motor has led to cavity sound field numeric models with accuracies of the order of 3dB throughout the spectrum in 1/3 octave bands except for frequencies close to 4 kHz and 5 kHz. This is attributed to uncertainties in the experimental results. The efforts required for a sufficiently detailed representation of the real geometry make this approach time consuming. The use of Statistical Energy Analysis may be an attractive alternative method considering the high modal density of the cavity and the shell at high frequencies.

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