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NUMERIC PREDICTION AND COMPARISON OF SOUND RADIATION FROM HERMETIC COMPRESSORS VARYING THE HOUSING THICKNESS

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Abstract. *The acoustic characteristics of hermetic compressors are critical parameters for the adequacy and acceptance of this product on the market. It is essential to predict the radiated noise by such equipment still in the design phase and to evaluate design changes aiming mitigate it. In this context, numerical modeling becomes feasible to reduce the cost associated with the manufacture of prototypes and design time. In this work, the TPA (Transfer Path Analysis) technique is used to predict numerically the sound power radiated by two different compressor housing thicknesses (3.2 mm and 2.8 mm). In order to validate the method, the results for 3.2 mm are compared with experimental measurements and showed good agreement. A numerical analysis of the vibro-acoustic energy trajectory between the sound source (four springs and discharge tube) and the receiver (microphones) is done in order to assess the contribution of different noise propagation paths. The results show that discharge tube path has the bigger contribution to the radiated noise and small changes in thickness are enough to change the level of noise radiated by the compressor in different frequency bands. The TPA is shown as a useful and viable tool for investigation of radiated noise by compressors and allows even identify operating ranges with more sensitivity. Thus, new changes in the acoustic design in order to minimize the noise may be proposed efficiently according to the most significant contribution to the receiver, cost and manufacturing process.*

Keywords: *hermetic compressors, TPA technique, radiation, simulation*

1. INTRODUCTION

In the consumer market, the request of low-noise levels of appliances has become essential to satisfy new expectations for environmental quality. At the same time, products with high performance and more energy efficient are more competitive. This has placed heavy demands on the refrigerator industry (Oliveira, 2006). In refrigeration systems, the compressor is the main source of noise and vibration, and is the focus of research in this manufacturing. In this way, new light weight refrigerator compressor has been developed, which has given excellent performance. Though, as the result, compressor noise and vibration may increase. Hence, great efforts have been made to solve the problem of noise radiation.

The compressor is basically composed of an electric motor and a single piston valve which are connected to the housing by four suspension springs and the tubing of discharge line, as shown in Fig. 1. The pressure change in the cylinder during a compression process excites the compressor mechanism. There is a high correlation between sound pressure amplitude and acceleration amplitude of the shell surface (Jeric *et al*., 2006)

The vibration of the compressor mechanism is carried through the tubing of discharge line and the internal spring suspension to the compressor housing which is a principal radiator of the compressor noise to the environment. The paths along which the acoustic energy flows from the source to the housing were divided into the air-borne path (cavity resonance) and structural-borne path (the suspension system and tubing of discharge forces) (Tojo *et al.*, 1980).

Transfer Path Analysis (TPA) is a technique for estimation and ranking of individual noise or vibration contributions by breaking it down to contributions from internal or external load paths to identify which paths are dominating the response. Once the dominant paths are identified, the problem is reduced to understanding how to minimize the contribution of these paths. The procedure to build a conventional TPA model typically requires two steps: (i) identification of the operational loads during in-operation tests; and (ii) estimation of the Frequency Response Functions (FRF's) from excitation tests (Janssens *et al*., 2014).

Experimental transfer path analysis (TPA) is a fairly well established technique. In general, however, this approach has not been broadly used as an analytical tool employing CAE-based data, even though there are a number of advantages to using numerical approach for TPA. For example, it is easy to estimate the FRF's using a finite element (FE) model, while direct measurement can be a difficult process.

In this work, a numerical methodology using TPA technique is presented to predict and compare the radiated noise for hermetic compressors housing and also to analyze the contributions of different noise propagation paths, using the software LMS Virtual.Lab®. Two different housing thicknesses were evaluated: 3.2 mm and 2.8 mm (the geometry and shape was maintained). The inputs of the methodology is the structural forces (four springs and discharge tube) which were estimated in previous work using hybrid (numerical and experimental) methodology. The FRF's, between four springs and tubing of discharge and virtual microphones, were predicted numerically. The numerical results were compared with experimental data and showed good agreement. TPA technique has proved to be a useful numerical tool for simulation, evaluation and comparison of vibro-acoustic paths, providing fast changes in the geometry of the structure in order to reduce the radiated noise and reducing cost with experimental tests.

2. THEORY BACKGROUND

2.1 Transfer Path Analisys (TPA) Technique

Transfer Path Analysis is a technique which allows to trace the flow of energy from a source, through a set of known structure-borne, air-borne and vibro-acoustic pathways, to a given receiver location (Padilha, 2006). Figure 1 illustrates the kinds of energy path.

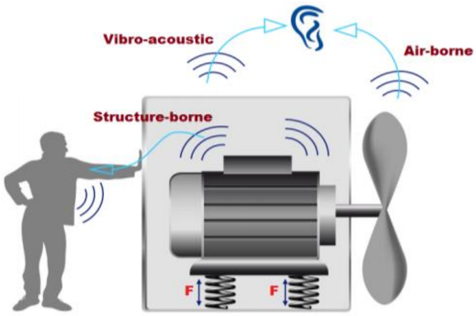


Figure 1. Types of energy path. Adapted from (LMS,2010)

Basically two quantities are required by the TPA method: (i) the frequency response functions (FRF'S) between the receiver and the inputs and (ii) the operational inputs (forces or volume velocities).

The FRF's can be obtained analytically by a finite element method (FEM) model, or experimentally: For structural-borne either hammer impact or shaker excitation can be used, to measure the corresponding accelerance frequency response function. To measure the acoustic frequency response functions, a volume velocity source can be used. The vibro-acoustic frequency response functions (between force and pressure) can be measured in a direct way using hammer or shaker excitation, or in a reciprocal way using a volume velocity source (LMS,2010). These FRF's are preferably measured when the source side is disconnected from the receiver side. In operational conditions the access to the inputs can be difficult.

In a similar manner to the difficulties encountered when measuring operational FRFs, the determination of operating forces is done indirectly (Plunt, 2005). Two alternative techniques can be used: the Complex Stiffness Method and Matrix Inversion. These techniques will not be discussed in this paper, but can be found on dedicated bibliographies (Padilha, 2006; Moura, 2010).

Assuming that the operating forces and transfer functions have been determined, TPA is based on the superposition principle that is valid for linear, time-invariant systems. The receiver sound pressure level (or acceleration level) is then explained as a superposition of partial results, each describing the contribution of the individual transfer paths, as in Eq. (1).

$$r_i(\omega) = \sum_{j=1}^{n} H_{ij}(\omega) \cdot S_j(\omega) \tag{1}$$

where $r_i(\omega)$ is the receiver spectrum as function of frequency, $H_{ij}(\omega)$ is the FRF between the receiver and the input *j* (force of volume velocity) applied at transfer path *i*, $S_j(\omega)$ is the operational force or volume velocity at transfer path *i* and *n* is the number of paths.

Thus it is possible to determine the relative importance of these paths in a given frequency range, checking which one has the most significant contribution to the receiver. Once the dominant paths are identified and classified, noise control methods can be applied more efficiently.

2.2 Acoustic Transfer Vector (ATV)

The methodology implemented in the software LMS Virtual.Lab® for calculating numerical FRFs uses the Acoustic Transfer Vector (ATV) approach. The concept of ATV is briefly described in this section.

In the classical approach, the Acoustic Boundary Element Method (BEM) is the numerical method for solving acoustic radiation problems in unbounded domains (Citarella *et al.*, 2007). The acoustic response is calculated by solving the system of equations for each loading condition, preferably in a multiload case solution sequence. In general this is very time-consuming since the BEM system matrices (full, symmetric, complex and frequency dependent) need to be assembled and solved at each frequency and for each set of load cases.

The Acoustic Transfer Vector (ATV) concept consists in the fact that every acoustic system can be considered as a linear system. Thus, it is possible to establish a linear input-output relationship between the input of an acoustic system (mechanical surface vibrations that generate sound waves), and the output (the sound pressure) at specific field points (Siano and Giacobbe, 2011).

Since only this normal component plays a role in the generation of sound waves, the ATV relates the normal vibration velocities $\{v_n(\omega)\}$ to the sound pressure $p(\omega)$ at a single microphone location as.

$$p(\omega) = \{ATV(\omega)\}^T \cdot \{v_n(\omega)\} \tag{2}$$

Where ω (angular frequency) highlights the dependence of frequency and $\{ATV(\omega)\}$ is the matrix of ATVs that relates the inputs and outputs of the system.

By definition, an ATV is completely determined by the characteristics of the acoustic system under consideration and depends only on the surface geometry, the fluid properties (density and sound speed) and any acoustic surface treatment (locally-reacting impedance). It does not depend on the loading (vibration boundary conditions, defined by the structural FE analysis). Therefore, the ATVs can be computed even before the structural response is known (Virtual.Lab, 2012). Thus, it is possible to store the ATV database and reuse it again to calculate and compare the acoustic performance for different situations, for example, to evaluate structural design parameters that do not change the acoustical model.

3. METHODOLOGY

The radiated sound pressure of a hermetic compressor model was predicted using the TPA technique for two different housing thicknesses: 3.2 mm and 2.8 mm (the geometry and shape was maintained). The structural forces (four springs and discharge tube) were estimated in previous work (Nunez *et al*., 2010) using hybrid (numerical and experimental) methodology. The vibro-acoustic frequency response functions (FRF's), between the four springs and tubing of discharge forces and virtual microphones pressure were predicted numerically using the ATV approach implemented in the software LMS Virtual.Lab®.

Due to the difficulties encountered when measuring operational forces, a hybrid (numerical and experimental) methodology was adopted by Nunez *et al*., 2010 to quantify the structural forces (four springs and discharge tube) which are responsible for the vibro-acoustic behavior of the compressor housing. In numerical step, it was used a BEM model of hermetic compressor to obtain the FRFs between unitary forces applied in the five points of excitation and sound pressure, measurement virtually externally to the compressor. In the experimental step, the sound pressure field of a compressor, with housing thickness of 3.2 mm, was measured in a semi- anechoic chamber on standard laboratory. The sound pressure levels were measured in 10 microphones distributed according to ISO 3744, illustrated in Fig. 2.

In possession of the FRF's (obtained numerically) and the sound pressures (experimentally measured), it is possible to solve the inverse problem and estimate the dynamic structural forces acting on the compressor, by minimum squares.

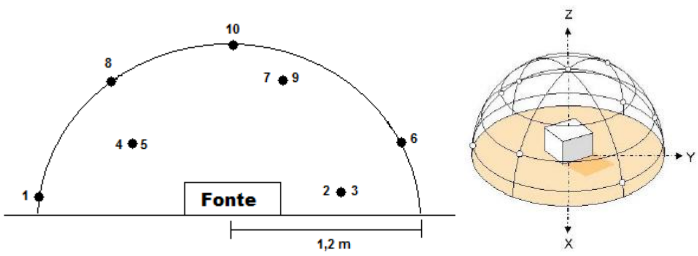
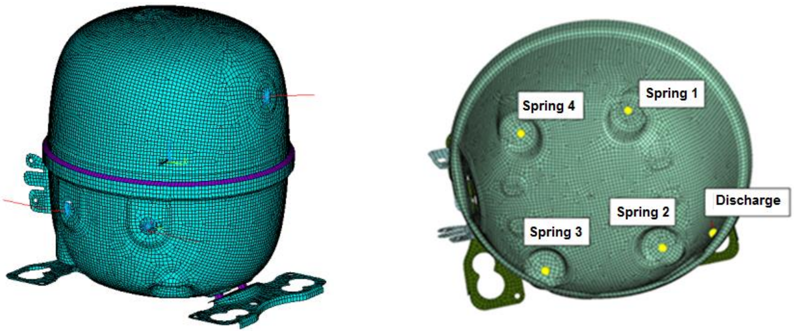


Figure 2. Microphones configuration according to ISO 3744

In this paper, the vibro-acoustic FRF (between force and pressure) was estimated numerically using the software LMS Virtual.Lab®. It is necessary to perform a structural analysis by finite element method in order to obtain the modal participation factors and an acoustic analysis, by boundary elements method (BEM), to calculate the ATV.

For structural analysis, the FE mesh was developed in Hypermesh®software and then exported to the software Ansys®. The model is composed by 28058 elements and 24396 nodes, as illustrated Fig. 3(a). The mesh was modeled mainly with *Shell63*, a linear shell element with six degrees of freedom each node. In this step the housing thickness is defined. A modal analysis has been performed, on the compressor operational frequency range from 0 Hz to 10000 Hz, in order to obtain the eigenfrequencies and eigenvectors.

A Modal-Based Forced Response, in the frequency range from 0 Hz to 10000 Hz, has been carried out in LMS Vitrual.Lab® *Noise and Vibration* module, in order to compute the modal participation factor of the compressor under unitary forces. The unitary forces were applied in four springs and discharge tube, as shown in Fig. 3(b), in the same directions adopted by the hybrid methodology above mentioned.



(a) (b)

Figure 3. Finite Element Modal (a) compressor mesh (b) detail of unitary force application points

The modal participation factors are calculated by LMS Virtual.Lab® according to Eq. (3), where $\{\Psi_i\}^T$ is the i-nth normal mode, $\{F\}$ is the unitary load vector, ω_{n_i} is the eigenfrequency corresponding to the i-nth mode shape and β represents the damping effect (β = 2ξ for structural damping and, where ξ is the critical damping ratio).

$$q_i(\omega) = \frac{\{\Psi_i\}^T \cdot \{F\}}{\omega_{n_i}^2 \cdot \left(1 - \left(\frac{\omega}{\omega_{n_i}}\right)^2 + j \cdot \beta\right)} \tag{3}$$

The acoustic model has been defined in LMS Virtual.Lab. The boundary element (BE) model of the compressor has been obtained starting from the finite element model, taking into account only the shell elements of the external skin. The compressor BE mesh is less discrete and composed by 1092 quadrilaterals element and 1094 nodes, as illustrated Fig. 4.

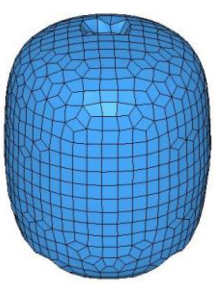
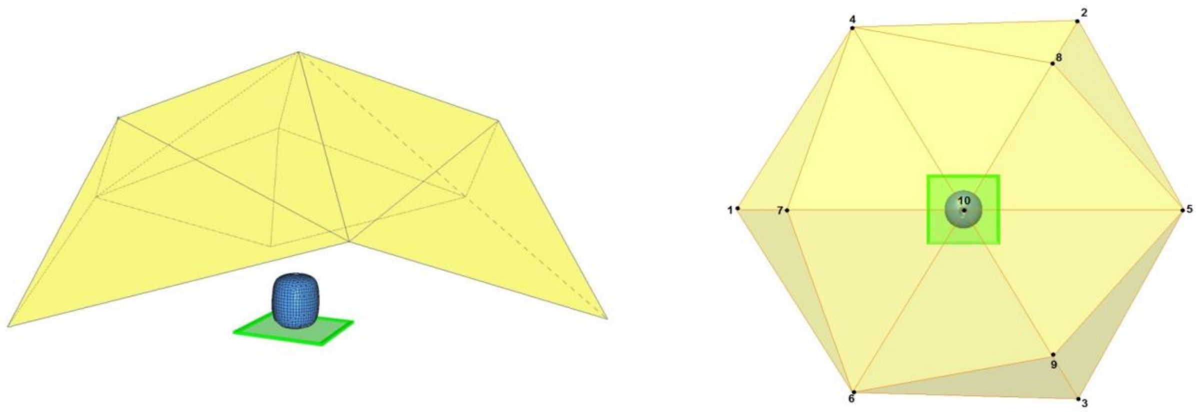


Figure 4. BEM compressor mesh

The field point mesh represents the virtual microphones positions in which the sound pressures should be calculated. Figure 5 shows the acoustic field point mesh (in yellow) and the plane to simulate the floor radiation (in green). The field point mesh is a hemispherical mesh, made up of 10 nodes, each one represents a virtual microphone, according to ISO 3744 (and experimental measurements). At each microphone location the ATVs is computed and stored.



(a) (b)

Figure 5. Field point mesh (yellow) and floor (green) (a) side view and (b) upper view

In the LMS Vitrual.Lab® software, after the modal participation factors, (Eq.3), and the ATV have been determined, the vibro-acoustic FRF are easily determined by Eq. (4) (LMS International, 2010).

$$FRF = \frac{P_k(\omega)}{F_i(\omega)} = \sum_{n=1}^{N} \frac{ATV_{ik}(\omega) \cdot \mathrm{q}_{n_i}(\omega) \cdot \mathrm{q}_{n_k}(\omega)}{[(\omega_n^2 - \omega) + j\omega\beta]} \tag{4}$$

Once the FRF and the operational forces have been determined, the acoustic pressure can be calculated, according to Eq. (1). The methodology to predict the acoustic pressure by TPA technique is summarized in the flowchart shown in Fig. 6.

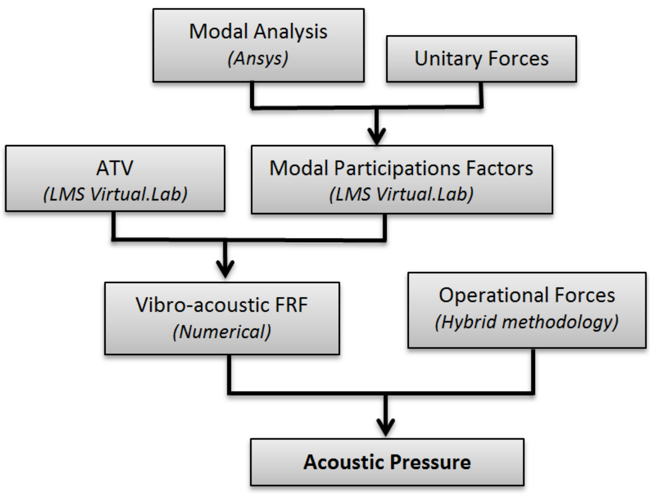


Figure 6. Methodology adopted in this work to predict the acoustic pressure

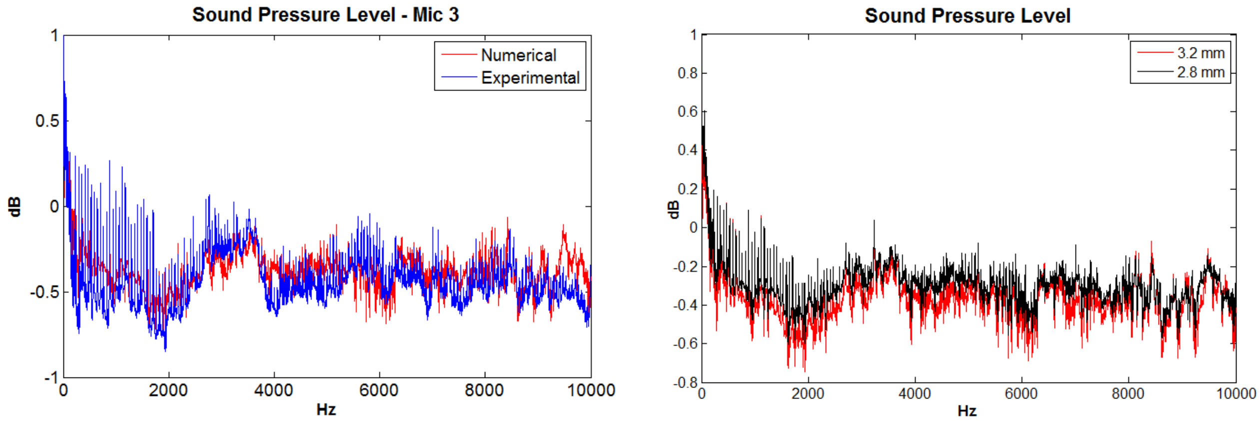
As mentioned, in this paper were considered two different thicknesses of the compressor housing: 3.2 mm and 2.8 mm. Thus, structural analysis was carried out twice, once for each case, while the ATV was computed just once, since the evaluated change does not alter the acoustic field, so it can be reused.

4. RESULTS AND DISCUSSION

In order to validate the results, the numerical acoustic pressures estimated for the compressor of 3.2 mm thickness were compared with the experimentally measured pressures (according to the hybrid methodology) To illustrate the results, Fig. 7(a) shows the numerical (in red) and experimental (in blue) sound level pressure measures at an randomly choose microphone 3. To preserve the confidentiality of data provided by the manufacturer of compressors, the results were normalized. It is possible to note a good agreement between the numerical and experimental curves. The difference between the amplitudes is expected, since the numerical model considers only the vibro-acoustic noise.

However, for the scope of this work the most important is that the tendencies of curves are the same, since future comparisons will be only between numerical results.

Figure 7(b) shows the average sound pressure levels (10 microphones) obtained by the numerical methodology for 3.2 mm housing thickness (in red) and the 2.8 mm housing thickness (in black). The global sound pressure level was 93.47 dB and 113.9 dB, for 3.2 mm and 2.8 mm, respectively. As expected, acoustic pressure radiated by the thinner housing is bigger, especially in low and medium frequencies.



(a) (b)

Figure 7. Sound Pressure Level curves (a) numerical (red) and experimental (blue) for 3.2 mm housing thickness (b) numerical for 3.2 mm (red) and 2.8 mm (black)

Figure 8 shows through a color graph, the sound pressure (dB (A)) estimated in each microphone to the thickness of 3.2 mm. The intensity of the colors is the sound pressure level - the maximum sound pressure values are indicated by red, as indicated in the legend on the right. The x-axis shows the frequency range, the y-axis indicates the microphone in which the pressure was computed. It can be observed that higher noise levels occur mainly at low frequencies (up to 1000 Hz) and in the range of 2500 Hz -4500 Hz. It is also noted that the microphone 03 has, in general, higher values for the sound pressure level.

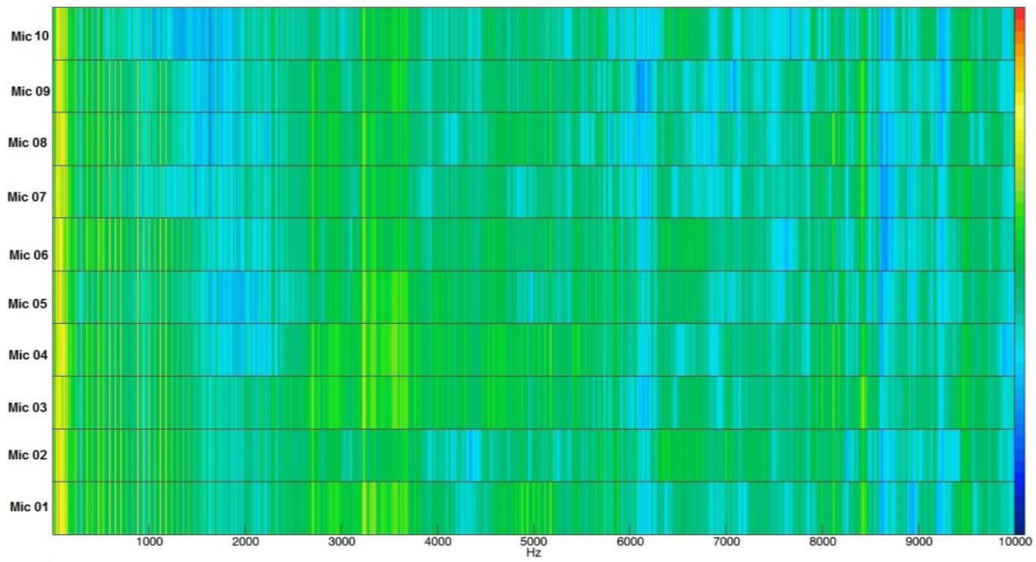


Figure 8. Sound pressure level estimated for each microphone in dB (A)

As TPA output, it is possible to evaluate the individual contribution of each vibro-acoustic energy transfer path (four springs and the discharge tube) in relation to microphone 03. For best viewing results, the frequency range of 2500-4500 Hz were selected and shown in Fig. 9.

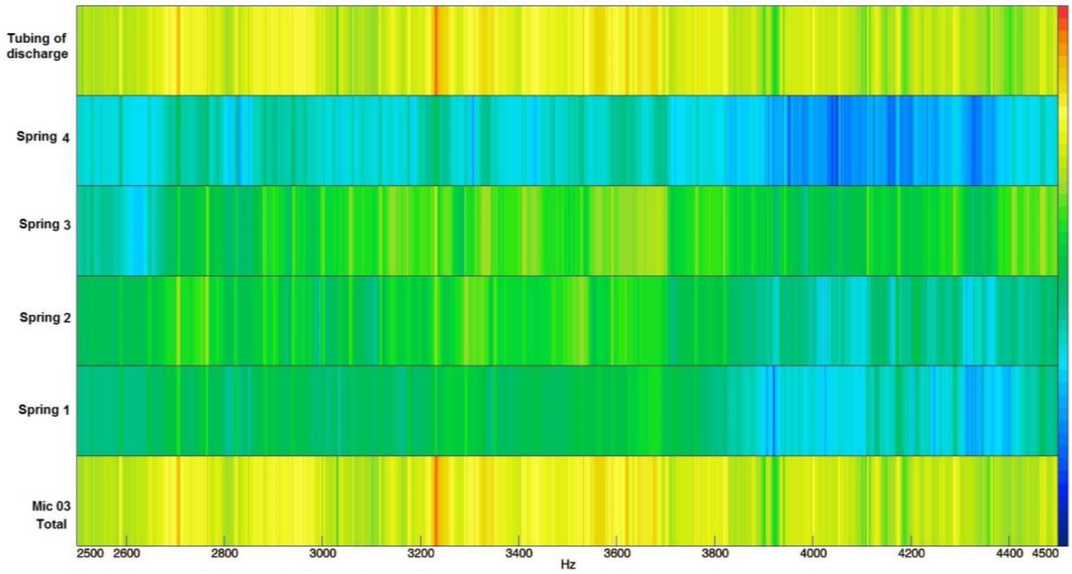
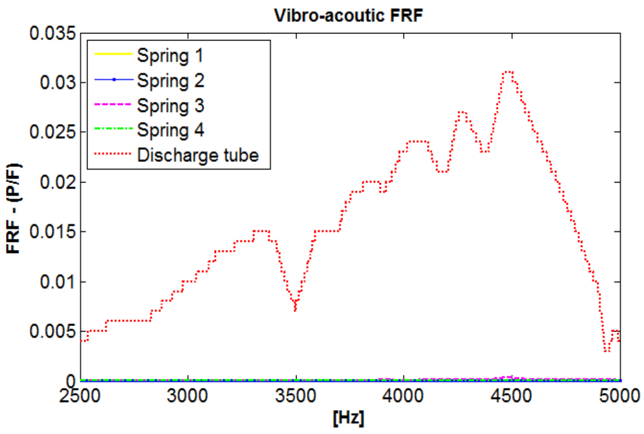


Figure 9. Transfer paths contributions for the sound pressure level at microphone 03

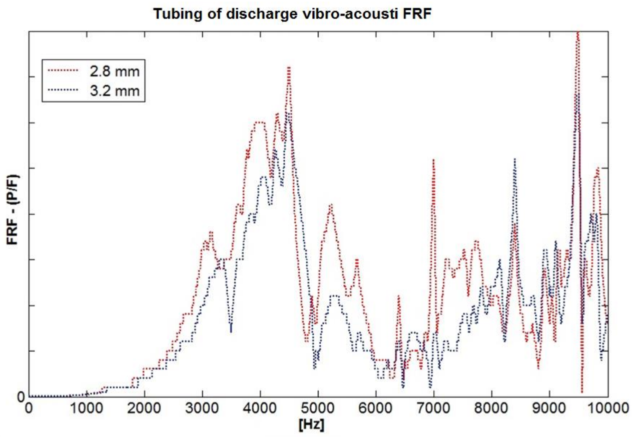
By the graph, it is clear that the principal contribution to the sound pressure on the microphone 03 is derived from the discharge tube in which even there is a pressure peak near to 3230 Hz. Though, the least contribution is from Spring 4, mainly between 3900 Hz and 4400 Hz. Also, it is interesting to note that the contribution of an individual path changes according to frequency. For example, Spring 01 is significant until 3800 Hz, and then decreases, while the contribution of the Spring 03 becomes greater.

Analyzing the FRFs, Fig. 10(a) shows the estimated vibro-acoustic FRFs between the 05 excitation points and microphone 03. The results show that response function of the discharge leads the greatest contribution of this path comparing to the springs. Thus, even small excitation forces are able to cause great responses.

Figure 10(b) shows the FRFs between the discharge tube and the microphone 03, for each case of the compressor housing thickness evaluated in this study (2.8 mm – in red and 3.2 mm in blue). It is observed that, in general, the thickness variation of 04 mm is enough to reduce the FRF in the frequency range considered. On the other hand, induces peaks at some frequencies.



(a)



(b)

Figure 10. (a) Estimated FRFs between the 05 excitation points and microphone 03 (3.2 mm) (b) FRFs between the discharge tube and the microphone 03 for thickness housing of 2.8 mm (red) and 3.2 mm (blue)

It is emphasized that the TPA results shown here were restricted to the microphone 03 for space restriction. But the technique and analysis are easily extended to other microphones.

5. CONCLUSION

This work presents a numerical method using the TPA technique to predict and compare the radiated noise for hermetic compressors housing. Two different housing thicknesses were evaluated: 3.2 mm and 2.8 mm.

The procedure to build a TPA model requires the identification of the operational loads that excited the system; and estimation of the Frequency Response Functions (FRF's). In this paper, the operating forces were estimated by a compressors company using a hybrid methodology (numerical and experimental). The vibro-acoustic FRF's, between the four springs and tubing of discharge forces and virtual microphones pressure, were predicted numerically using the methodology implemented in the software LMS Virtual.Lab®: A structural analysis was performed by FEM in order to obtain the modal participation factors and an acoustic analysis, by BEM, to calculate the ATVs. As the ATVs are completely independent from these design parameters, they only had to be calculated once, and can be reused for the different housing thickness. Models were evaluated in a frequency range of 0 Hz to 10000 Hz.

The estimated numerically sound pressure was compared with experimental data obtained in the manufacturer's laboratory. There was good agreement between the numerical and experimental curves, since both showed the same trend. As expected, the acoustic pressure radiated by the thinner housing is bigger, especially in low and medium frequencies.

Transfer Path Analysis allows quantifying and classifying the vibro-acoustic path between the excitation points and the receiver (virtual microphones). The results shown in relation to the microphone 3, reveal that the discharge pipe has the biggest contribution to the sound pressure level in comparison to other pathways (springs). Once the dominant paths are identified, structural design modifications can be proposals more efficiently, and the acoustic response can be easily evaluated at each design cycle at low computational cost and without prototypes or experimental tests.

6. REFERENCES

Citarella R., Federico L., Cicatiello A., 2007, "Modal Acoustic Transfer Vector Approach in a FEM-BEM Vibro-Acoustic Analysis", *Engineering Analysis with Boundary Elements*, p. 248

Janssens K. H. B. , P. Mas , P. A. Gajdatsy , H. Van der Auweraer , and L. J. P. Gielen , 2009, " Transfer path analysis," U.S. patent 2009/0276197 A1

Jeric A., Govekar E., Gradisek J. and Grabec I., 2006, "Influence of the Housing on the Noise Emitted by a reciprocating Compressor", In *The Thirteenth International Congress on Sound and Vibration*. Vienna, Austria.

LMS International, 2010, "ATV and Panel Acoustic Contribution Analysis.", Application notes, Not published, p. 25.

Moura, H. G., 2010, "Métodos de identificação de forças em sistemas vibroacústicos", D. thesis, Universidade Federal de Santa Catarina, Florianópolis, Brazil.

Nuñez, I. J. C., Marqui, A. L. L., Nunes, M. A. A., Arruda, J. R. F., 2010, "Metodologia híbrida para estimativa de forças estruturais aplicadas à vibroacútica de compressores herméticos", In *VI Congresso Nacional de Engenharia Mecânica – CONEM 2010*. Paraíba, Brazil.

Oliveira A., 2006, "Adição de amortecimento estrutural usando materiais viscoelásticos", Master dissertation, Universidade Federal de Santa Catarina, Florianópolis, Brazil.Padilha, P. E. F., 2006, "Comparação de técnicas de análise de caminhos de transferência vibroacústicos", Master dissertation , Universidade Estadual de Campinas, Campinas, Brazil.

Plunt, J., 2005, "Finding and Fixing Vehicle NVH Problems with Transfer Path Analysis", *Sound & Vibration*. Vol. 39, p. 12

Sisson, T. and Simpson, F., 1984, "Analysis and Control of Hermetic Compressor Sound Power Levels". In *International Compressor Engineering Conference*. Indiana, Unitated States.

Siano D., Giacobbe S., 2011, "Radiated engine noise prediction using multi-body simulation". In *8th International Conference on Structural Dynamics – EURODYN 2011*, Leuven. Belgium.

Tojo, K., Saegusa, S., Machida, S., Hirata, T., Sudo, M., 1980, "Noise reduction of refrigerator compressors" In *International Compressor Engeniring Conference*. Indiana, Unitated States.

Virtual.Lab 11 SL1, 2012, "Manual – Online Help", LMS

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