# DEVELOPMENT OF A METHODOLOGY TO IDENTIFY NOISE SOURCES IN A ROTARY COMPRESSOR

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***Abstract.*** *Rotary compressors are widely applied on air conditioning devices due to the high efficiency and pressure rates. In the other hand, a great problem of these compressors is the irradiated noise. The main noise sources of rotary compressors are internal turbulence, valve impacts, bearings, electric motor and roller-vane friction. Many of these sources can excite the compressor housing that is one of the largest sound radiation surface areas in rotary compressors. The purpose of this paper is to propose a methodology to identify the sources position and evaluate the treatment effects of noise on the generated power levels. Structural forces were estimated to be used as input parameters for a sensitivity analysis, dynamic displacements were measured on some points of compressor and applied on a finite elements model FEM to calculate the sound field resulted.*

**Keywords**: *Rotary compressors, vibroacoustics, estimated force.*

## 1. INTRODUCTION

Nowadays compressors market requires products with low noise and vibration levels, since these sources cause discomfort to users (Jeric, 2006).

Rotary vane compressors are widely applied on air conditioning devices due to the high efficiency and pressure rates (Crocker, 2007). The main noise sources of these compressors are internal turbulence, valve impacts, bearings, electric motor and roller-vane friction. The suction accumulator that is connected to the housing compressor can be another secondary source (Zhu, 2008). So, the main internal sources are: a) the pressure variation between suction tube and discharge collector. This pressure fluctuation excite the compressor housing, and any other structure attached to it, which can vibrate and irradiate noise (Crocker, 2007); b) the transmitted impact by kinetic energy of the valves closing (Leyderman, 1993); c) the electromagnetic forces on the motor stator; d) the gas pulsation into compression chamber, which excites mechanical parts of the structure, irradiating noise to external environment (Kim, 1998).

The sound spectrum of a rotary vane compressor is very complex. In a general approach, in 60 revolutions per second, the generated sound in the frequency range of 600 to 900 Hz can be explained by the gas flow and its pulsation; in the frequency range of 2 to 2.5 KHz by valve impacts; and above 3.5 KHz mainly due to the friction forces (ASHRAE, 2008).

In accordance to Crocker (2007), high vibration levels in different frequencies are observed in regions above and below of motor stator, on accumulator support, and close to soldering line of suction. In his work three methods were applied to reduce noise and vibration:

1. Vibration dampers around the bottom of compressor housing, close to regions of high vibration magnitude, resulting in a reduction of 2.5 dB(A) in overall spectrum;
2. Changes in the geometry of suction channel, resulting in 2 dB(A) of reduction due to a smooth flow,
3. Re-design of the rotor e roller manufactured with polyamide material, resulting in reduction of 2 dB(A).

The mechanisms and transmission paths of noise in these compressors are very complex, but all generated noise is from transmitted vibration to compressor housing, allowing the use of source identification techniques such as sound intensity probes and beam forming (Egea et all, 2010) or particle velocity sensors (Honschoten, 2010), for example. The main problem of these techniques is that, besides the high cost, they only indicate the sound intensity and position of noise sources, without any quantitative information about the amplitudes of the efforts that have generated the acoustic field.

The objective of this work is to propose a methodology to identify the sources position and evaluate the amplitudes of the inputs responsible by the generated power levels.

The technique is based on the measurement of dynamic displacements on some points of compressor and the application of them on a finite elements model FEM, used to calculate the sound field resulted by the imposed displacements.

In this approach, the FEM model is used to interpolate the measured displacements, although the quality of interpolation will be proportional to the ability of the FEM to represent high order modes. The great advantage of the proposed methodology is that it allows the estimation of structural forces, which generate the acoustic field. These forces can be used as input parameters for sensitivity analysis and structural optimization, for example. In this work, a sensitivity analysis was performed by means of vibration levels reduction in some interest regions of the compressor and Sound Power Levels (SWL) was analyzed.

This paper is organized as follows: section 2 discusses the paper methodology, in section 3 the experimental procedure is presented, section 4 describes the numerical and experimental results and finally in section 5 the necessary conclusions will be presented.

## 2. METHODOLOGY

Figure 1 shows a typical sound power levels (SWL) spectrum of studied compressor, in 1/3 of octave bands from frequency 100 Hz to 10000 Hz. In accordance to figure there is a predominance of high frequencies in the SWL of generated noise. For this reason, in this paper the proposed methodology was applied in the 1/3 of octave bands from 630 Hz to 2.5 kHz.

The hybrid methodology presented in this paper consists of:

1. Measurement of vibration levels generated by the compressor housing and accumulator at several points. To ensure the phase between the measurements points, a reference point i was used. So the Frequency Response Functions FRFji was estimated for other points j in the frequency band of interest.
2. Construction of an acoustic model FEM with a mesh that meets the requirements of the frequency band analyzed, at least six nodes per wavelength and infinite contour, or the air impedance of at a distance of at least two lengths wave of vibrant surfaces.
3. Validation of acoustic model.
4. Application of measured displacements in the FEM model using only frequencies of interest and simulation of acoustic field to calculate the SWL. The jth harmonics of real and imaginary displacements are obtained by the product between the ith harmonic of displacement (obtained by fft of vibration response) and the FRF Hji. The experimental measured displacements of compressor were applied on their respective nodes of FEM.
5. Reduction of vibration amplitude in some parts of compressor and recalculation of sound power levels for purpose of sensitivity analysis.

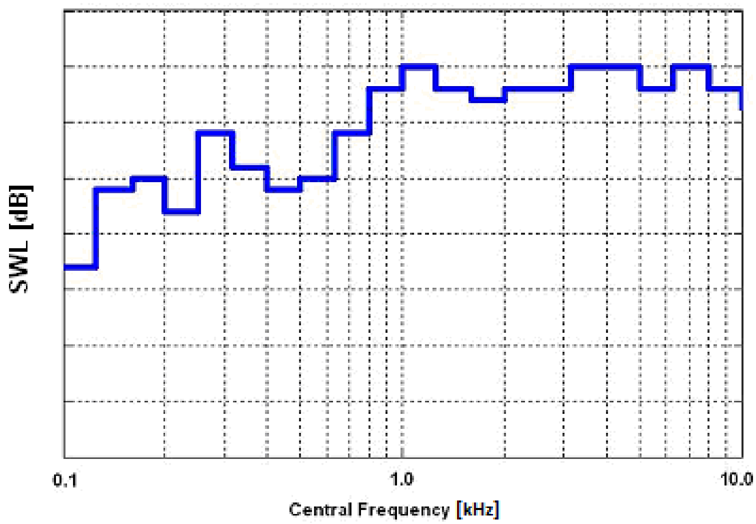


Figure 1 - Typical sound power levels spectrum of rotary compressors.

Figure 2 shows the finite element model (ANSYS ®) of the studied compressor. The structural part of the compressor was modeled using element type PLANE42 and the acoustic element was FLUID129. The element size used for the acoustical fluid ranged from 5 mm (close to compressor structure) to 20 mm (external boundary condition) as observed in Figure 2. These values were chosen to ensure a minimum number of 6 nodes per wavelength at the maximum analyzed frequency (2.8 kHz).

The boundary condition was modeled with tetrahedral element type FLUID 129 with present structure and the impedance of air (MU = 1). The length of acoustical field, from the boundary condition and the structure close to compressor was 1.22 m, which insures at least 2 wavelengths between the structural elements and the elements of external acoustic field (boundary) at 562 Hz (lower limit of 630 Hz band). The command FSI by Ansys® was used for the fluid structure interface.

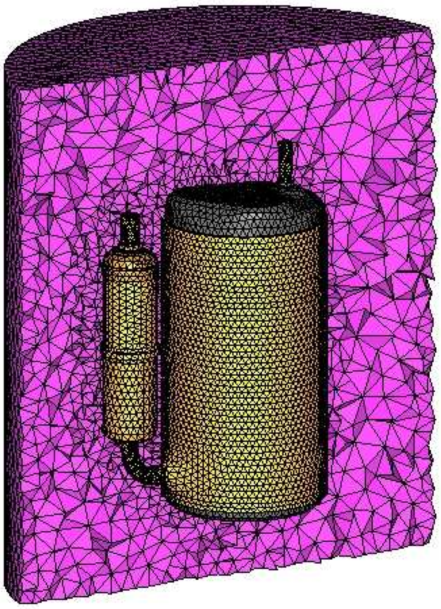

Figure 2 - Acoustic finite element model of the rotary compressor.

For sensitivity analysis purposes, the compressor housing was divided into four regions as shown in Fig. 3. The sensitivity analysis consisted of:

- Simulation of the acoustic field according to the vibration levels measured in 1/3 of octave bands from 630 Hz to 2500 Hz;
- In accordance to the vibration profile of compressor, a reduction of 10 dB in vibration levels of the studied area is imposed;
- Simulation of the modified vibration profile;
- Subtracting the resulting spectra.

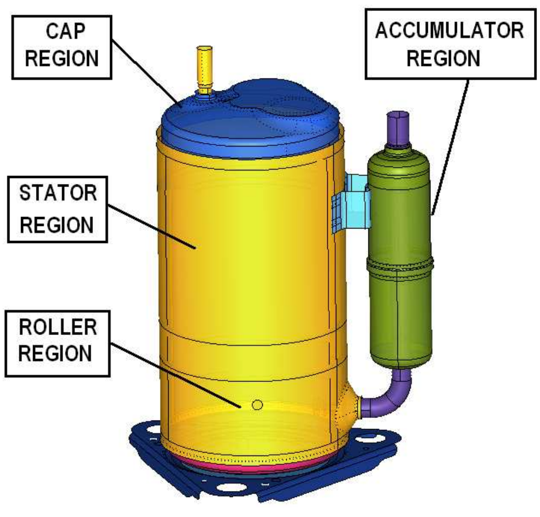


Figure 3 - Analyzed regions of rotary compressor.

## 3. EXPERIMENTAL PROCEDURE

The compressor housing was mapped using 124 measured points (Fig. 4), in order to represent several vibration modes of housing for analyzed frequency bandwidth. The instrumentation used on experimental set up were:

- 3 accelerometers B&K model 4371 and 1 accelerometer PCB model 320C33;
- 4 microphones PCB piezotronics, 377B02;
- 1 signal conditioner and amplifier B&K NEXUS model;
- 1 A/D board by National Instruments NI 9233 four channels;
- 1 notebook to record the data.

Data acquisition was realized with compressor operating after some minutes to ensure stable conditions of operation and the pressures of suction and discharge were about 3 and 21 bars, respectively. Ten seconds of signal were acquired with a sampling rate of 33KHz.

To validate the simulations, sound directivity of compressor was measured on its horizontal medium plane. Four microphones were used to acquire the sound pressures during 10 seconds with a sampling rate of 33KHz. Thirty-five measurements were realized in a semi-anechoic chamber with the microphones spaced 10 degrees from each other around the compressor until to reach a full circle.

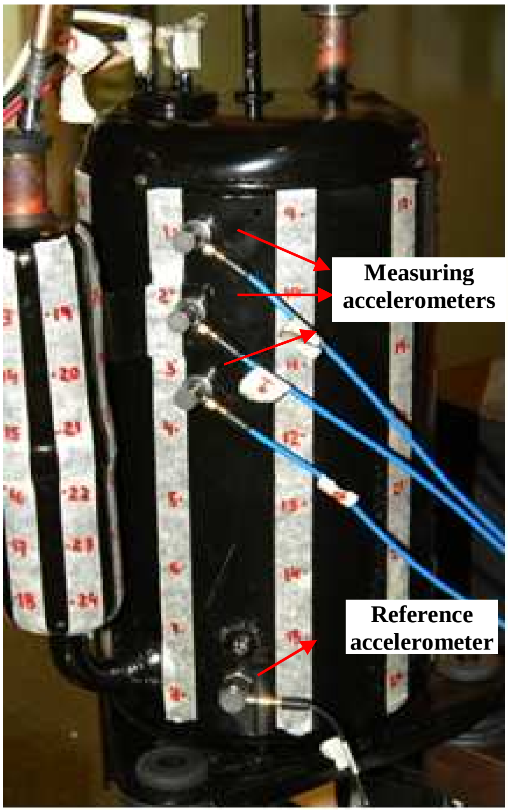


Figure 4 - Measurement and reference accelerometers used in the experiment.

## 4. RESULTS

Due to the analysis of numerical results are comparative, and for reasons of industrial sensitive information, the results of pressure or sound power levels were presented without the values.

The FEM model validation was carried out in the 1/3 of octave band centered in 2.5 kHz. Figure 5 shows the simulated acoustic field for the frequency of 2435 Hz, which is included in the 2.5 kHz band.

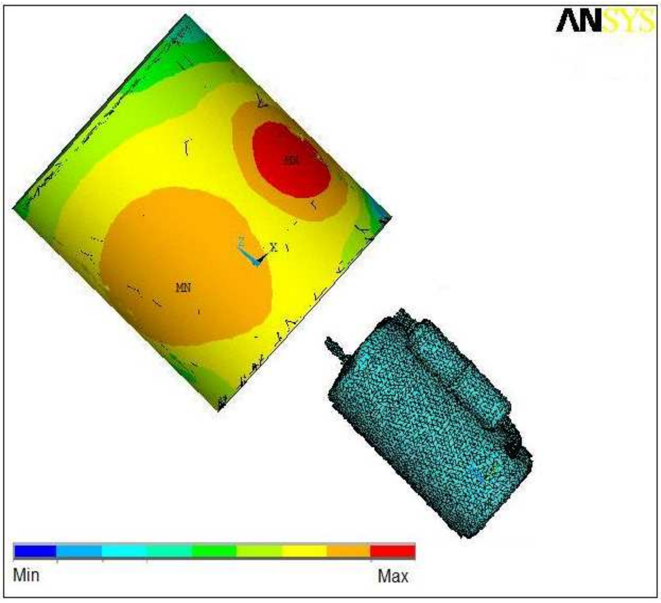


Figure 5 - Sound pressure field around the compressor at 2.5 kHz band.

Figure 6 shows the directivity curve in the horizontal medium plane of the compressor for the same frequency band of 2.5 kHz and for overall sound pressure level. Comparing the Fig. 5 and Fig. 6 the maximum levels of sound pressure occur at accumulator region. Numerically a difference lower than 2 dB between the simulated and measured overall values at 2.5 kHz band was observed.

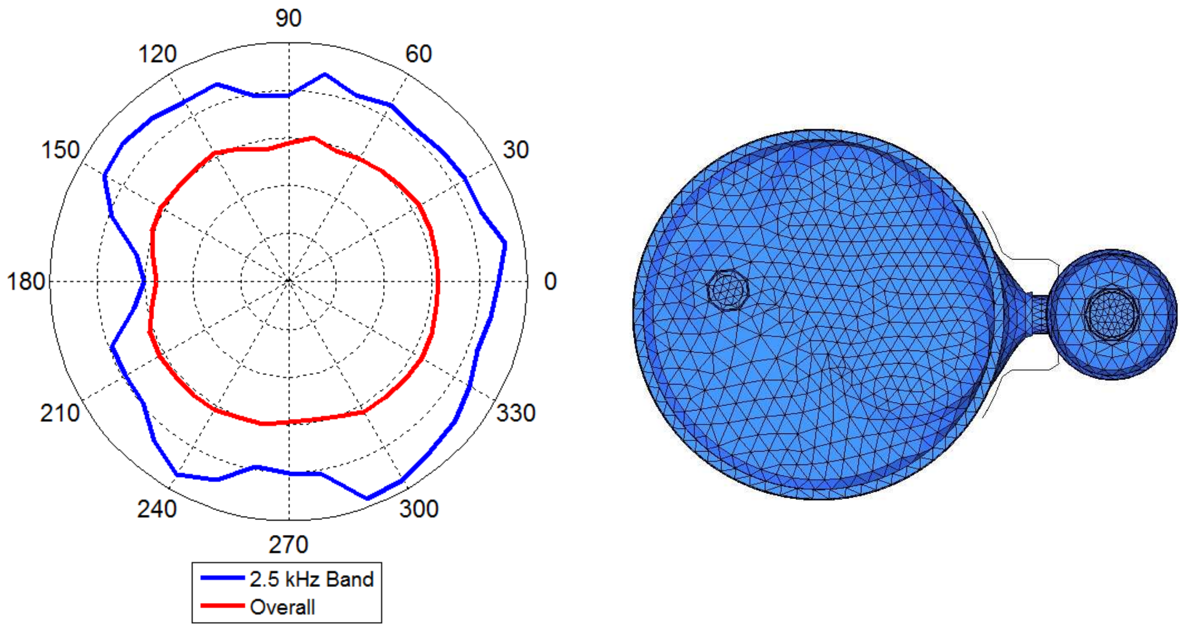


Figure 6 - Measured sound pressure around the compressor at 2.5 kHz band.

Figure 7 shows the Sound Power Level SWL histogram simulated for the compressor at 1/3 of octave bands from 630 Hz to 2.5 kHz.

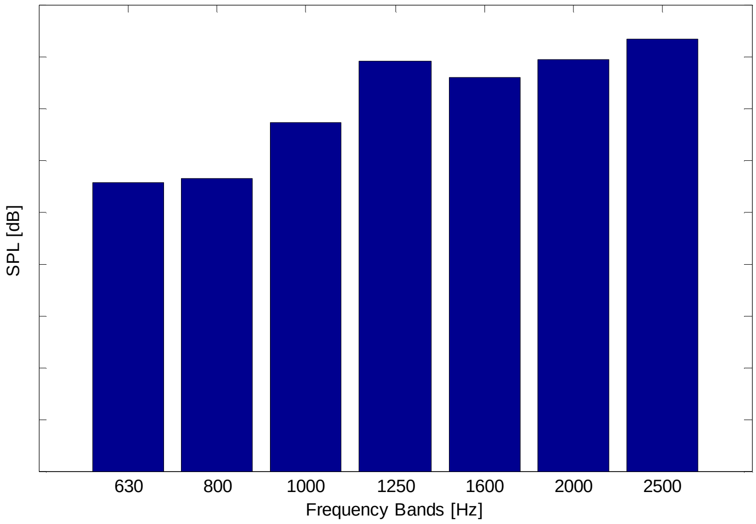


Figure 7- Simulated SWL with the original measured displacements.

Table 1 shows the Insertion Loss in SWL due to the reduction of 10 dB in vibration for the compressor regions at the 1/3 of octave bands from 630 Hz to 2.5 kHz.

Table 1 – Results of the sensitivity analysis to verify the vibration influence in the estimated SWL.

| **Regions** | **Insertion Loss [dB]** |
|---|---|
| Roller Region | 0.7 |
| Accumulator Region | 4.5 |
| Upper Housing Region | 5.4 |
| Stator Region. | 1.6 |

The small attenuation observed for roller and stator regions can be explained by their high stiffness, so the measured vibration levels in these regions at the frequency bands analyzed are low.

The results of vibration reduction of 10 dB in roller region are shown in Fig. 8, where a low insertion loss in all analyzed frequency bands is observed.

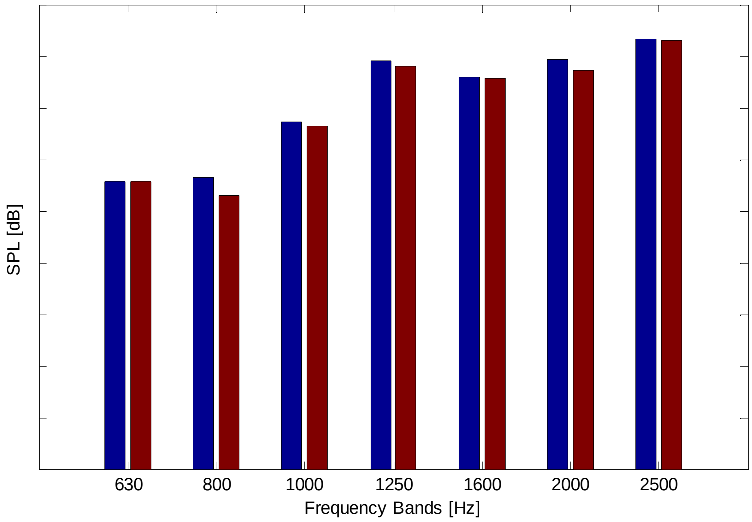


Figure 8 - Simulated SWL with the original measured displacements (blue) and with 10 dB of reduction in vibration levels for the Roller Region (red).

With the reduction of 10 dB in vibration levels of accumulator region a decrease in noise levels at 1250 Hz, 2000 Hz and 2500 Hz bands was observed, as shown in Fig. 9.

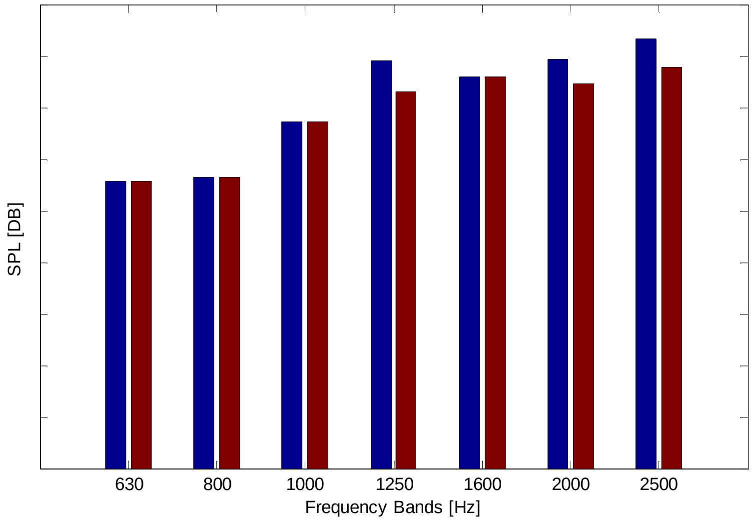


Figure 9 - Simulated SWL with the original measured displacements (blue) and with 10 dB of reduction in vibration levels for accumulator Region (red).

For the upper housing region of compressor there is attenuation of noise levels at bands from 1250 Hz to 2500 Hz, except at bands below 1000 Hz (Fig. 10).

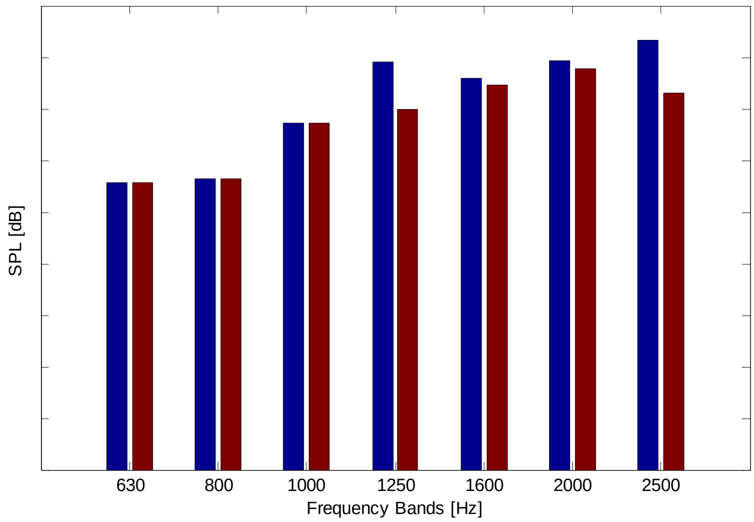


Figure 10 - Simulated SWL with the original measured displacements (blue) and with 10 dB of reduction in vibration level for Upper Housing Region (red).

The reduction of 10 dB in vibration levels for stator region resulted in a little reduction of noise below 1600 Hz band, and a significant reduction only at 2000 Hz band (Fig. 11).

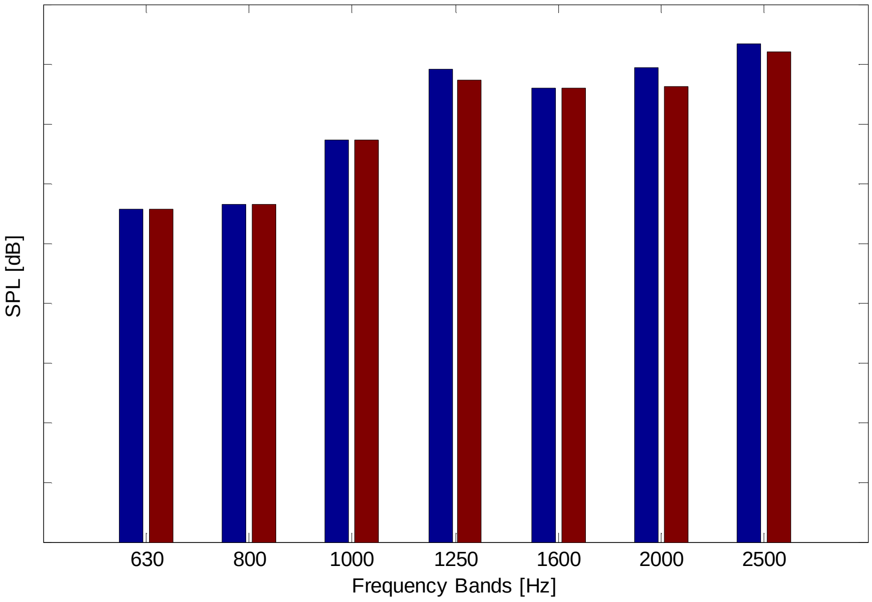


Figure 11 Simulated SWL with the original measured displacements (blue) and with 10 dB of reduction in vibration level for Stator Region (red).

A resonance at the 2500 Hz band, which has a peak in 2435 Hz, was observed. In order to discover the reason of this resonance the acoustic field around the structure of the compressor at 2435 Hz (Fig. 12) and the structural forces that generate the vibrations on compressor housing at this frequency were simulated by comand Ansys® structural forces. The analysis of these figures indicates that the vibrations that generate the sound field at the frequency of 2435 Hz are due to the reaction loads on the bearings.

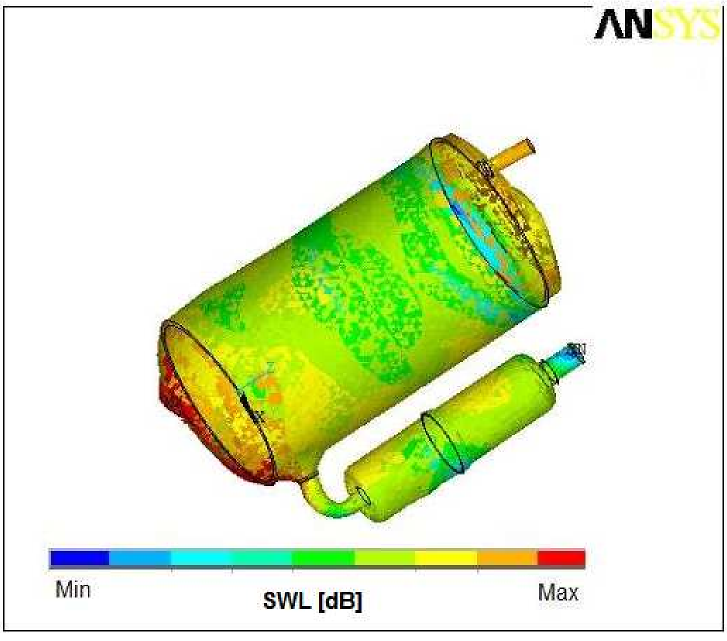


Figure 12 -Sound pressure field around the structure of compressor at 2435 Hz.

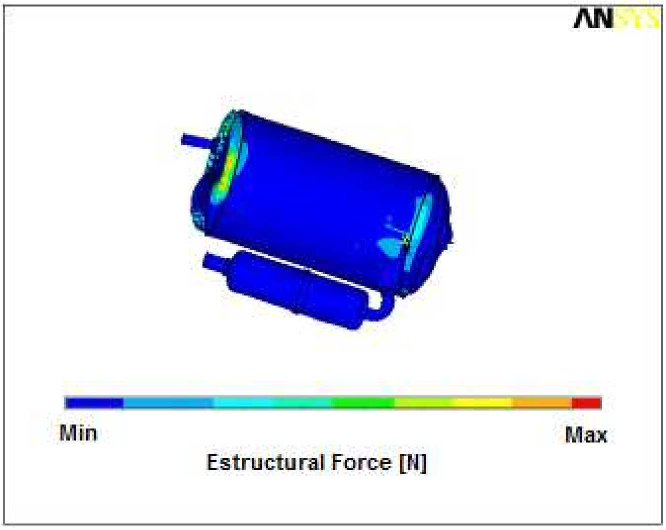


Figure 13 –Forces acting on the compressor structure at 2435 Hz

## 5. CONCLUSION

The conclusions of this paper are:

- The purposed methodology can be considered an efficient tool to identify the main noise source of a rotary compressor.
- In the validation of this methodology the maximum error observed between the simulated acoustical field and the measured one was about 2 dB.
- A insertion loss of 0.7, 4.5, 5.4 and 1.6 dB in the SWL (1/3 of octave bands from 630 Hz to 2.5 kHz) due to the reduction of 10 dB in vibration levels of Roller, Accumulator, Upper housing, and Stator regions, respectively were observed. The small influence of Roller and Stator regions can be explained by their high stiffness at studied frequency bands.
- The presence of a pick at 2435 Hz in the spectrum of SWL is due to the reaction forces into the bearings. A natural frequency whit this value was observed in a modal analysis of the shaft.

## 6. ACKNOWLEDGEMENTS

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