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Acoustic Improvements for a New Generation of Variable Speed Compressors

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ABSTRACT

Variable speed reciprocating compressors (VSRC) present some advantages comparing to single speed compressors, mainly in terms of energy consumption and noise. Once the acoustic performance is analyzed, it is verified that the compressor running most of the time at frequencies lower than the ones of the electric net imply lower sound level for VSRC and, therefore, many home appliances manufacturers direct their use to the premium refrigerator models. Following a general market trend for comfort items, home appliances noise is becoming more and more important nowadays, implying even VSRCs demand acoustic improvements in order to reduce the refrigerators sound level. In such a scenario, this paper presents numerical and experimental studies carried out in an existing platform of variable speed compressors, identifying some sources and transmission paths associated to the overall sound level of the compressor itself and to a representative refrigerator model. Based on the foreseen opportunities, a design optimization focusing on sources and paths is presented and, as a result, the acoustic performance of the new generation is compared to the performance of the current version of this variable speed compressor platform.

1. INTRODUCTION

When a refrigeration hermetic compressor is developed, in the acoustic point of view basically two things must be evaluated: its airborne noise and its structure borne noise. The structure borne noise can be manifested, for example, by the gas pulsation that comes from the suction and the discharge connectors of the compressor and by its low frequency vibration. On the other hand, the airborne noise is the sound radiated by the compressor itself.

Analyzing the behavior of a typical household cabinet, it's known that the airborne noise of the compressor influences typically over 3 kHz, as it will be seen in the next chapter. In turn, the structure borne noise generated by the coupling compressor and appliance appears normally at frequencies below 3 kHz.

Knowing these facts, the VSRC are being used more and more nowadays not only due to the energy consumption reduction, but also because with this kind of compressor, during the regular cycles, the airborne and the structure borne noise decrease. Both the energy consumption and the noise are lower because the compressor in regular cycles runs at a frequency lower than the frequency of the alternating voltage of the electric power. The main consequence from the acoustic point of view is, obviously, a quieter cabinet and with better sound quality.

Even with all the advantages listed before, the general home appliances market and customers are demanding more and more improvements in order to reduce the sound level radiated by the refrigerators. This work aims to present a new generation of VSRC compressors, comparing it to the existing one.

An initial analysis of the opportunities will be presented, as well as the optimized components and, in the end, some comparisons from the acoustic point of view between the new and the current generations of this VSRC

2. VSRC CURRENT GENERATION AND THE OPPORTUNITIES TO IMPROVE THE NOISE

Before starting to optimize any component from the acoustic point of view, it's necessary to understand some characteristics of the current generation of VSRC.

The current generation of VSRC mentioned during this work is considered nowadays by being very good at low speeds in terms of airborne noise, but some drawbacks take place when the structure borne noise is analyzed. Besides this, the inverter for this compressor generation (device that permits the compressor to run at variable frequency) must avoid some ranges of speeds because there is some component's natural frequency close to 54Hz. Figure 1 shows a schema of the inverter jump principle which does not allow the compressor to work continuously at a region where resonance occurs.

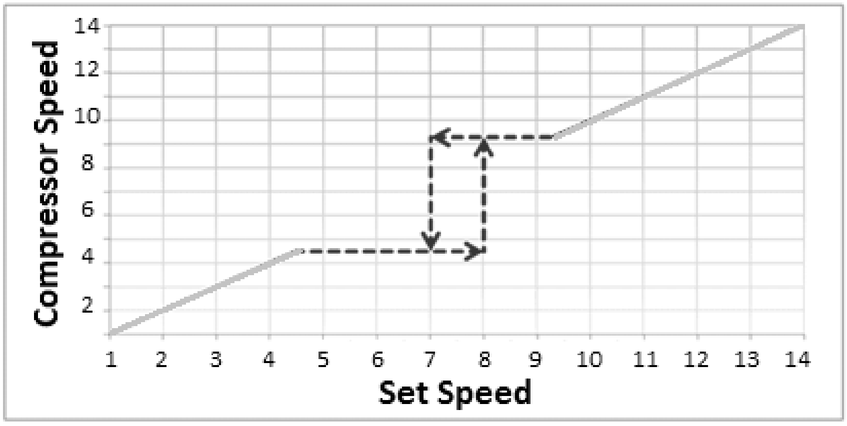

Figure 1: Inverter jump principle.

A second point that must be understood is the reason of the high structure borne noise. Some previous analysis showed that the most important part of this noise is the suction pulsation. This pulsation is considered here as the acoustic waves created by the suction valves movement which, after being attenuated by the suction filter, continues outside the compressor and could excite the cabinet suction tubing, causing an increase of the structure borne noise. Figure 2 shows the layout of the suction filter that the current version of VSRC uses and the proposal that has the potential to reduce the suction pulsation. Figure 3 presents a comparison in terms of sound power level of a representative cabinet with the current version of VSRC with the regular versus the proposed suction filter.

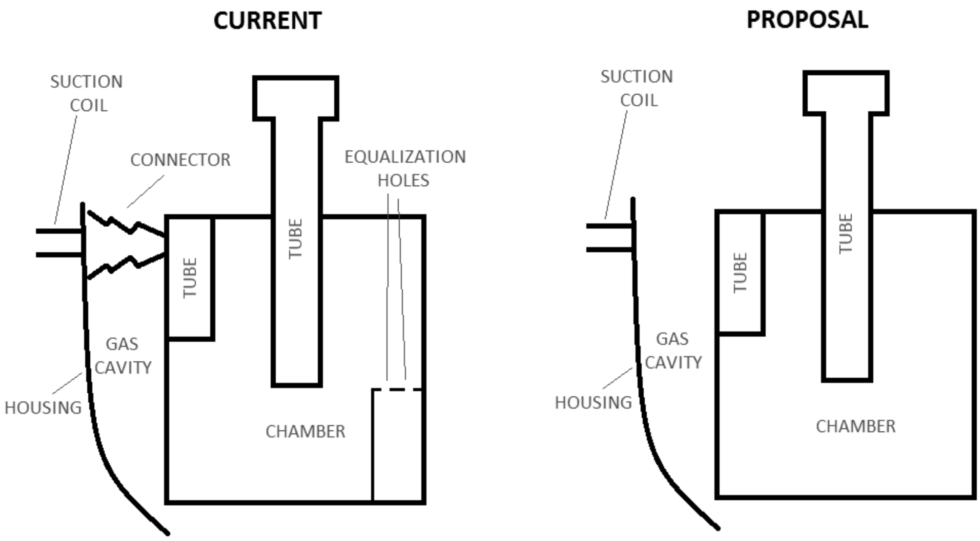

Figure 2: Current and proposed concept of suction filter layout

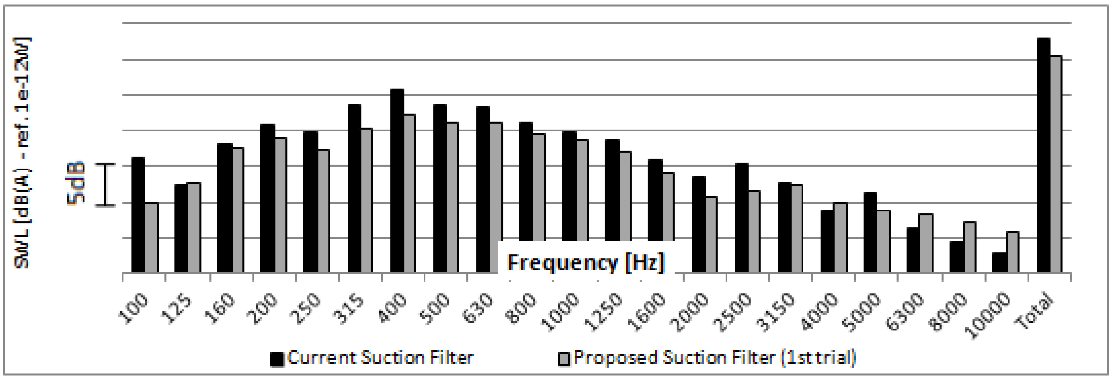

Figure 3: Sound power level of a representative appliance with current VSRC with regular and proposed (1st trial) suction filter.

Finally, even being very good in terms of airborne noise at low speeds, it was decided to further improve it and, so an optimization of the housing response was required.

Consequently, some points of optimization were evaluated:

- Redesign of the discharge tube: the goal of this modification would be design a tube without natural frequencies in the range that the compressor could run. As a consequence, it would be assured that the compressor could work at the full range of speeds.
- Modification of the concept and redesign of the suction muffler: as it has been seen, the suction pulsation could potentially excite the cabinet tubing, increasing the contribution of the structure borne noise for the current VSRC version. Therefore, a concept of semi-direct suction should be adopted, but a re-design of the suction muffler would be required in order to maintain or even to reduce the airborne noise of VSRC version.
- Optimization of the housing response: even being very good in terms of airborne noise at low speeds, the goal of this improvement would be the reduction of the noise not only at low speeds, but also at high speeds. The airborne noise has a huge contribution for the cabinet overall noise and for the sound quality at higher speeds.

3. OPTIMIZATION OF THE COMPONENTS

3.1 Discharge Tube

Basically, the discharge tube optimization aimed to obtain a tube with higher natural frequencies. In order to improve the shape, the first step consisted in designing various proposals and simulating some modal analysis using a commercial simulation code.
After setting the material properties and the mesh of the discharge tube (this mesh used quadratic quadrangular elements of maximum area of 1 mm^2, considering the geometry of the midsurface of the tube), the boundary conditions were defined, as exemplified in Figure 4.

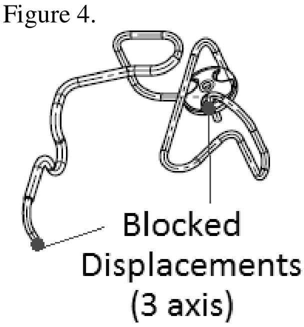

Figure 4: Modal Analysis – Boundary Conditions.

After solving the modal analysis, the best proposals were analyzed by means of a numeric harmonic analysis in order to choose two shapes that would be prototyped and validated. The excitation for the harmonic analysis was placed in one of the regions that the displacements were blocked in the numeric analysis (for the other region, the blocked displacement was maintained).
With the prototyped proposals, the validation of the modal analysis was done with an impact test, that used an impact hammer to excite the resonances of the tube and a laser vibrometer measures the response of the tube in one or more regions, as showed in Figure 5.

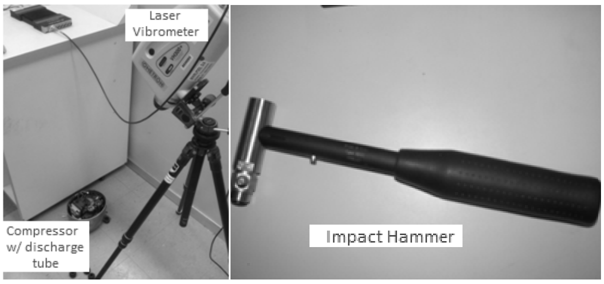

Figure 5: Impact test to determine experimentally the natural frequencies of the discharge tube.

Table 1 shows the simulated natural frequencies resulted similar to the tested ones.

Table 1: Comparison between simulated and tested natural frequencies of the discharge tube.

# of Natural Frequency	Reference Tube (Current VSRC)		Proposal 1		Proposal 2 (New VSRC)	
	Numeric	Experimental	Numeric	Experimental	Numeric	Experimental
1	55 Hz	54 Hz	86 Hz	84 Hz	83 Hz	84 Hz
2	81 Hz	80 Hz	109 Hz	109 Hz	109 Hz	110 Hz
3	133 Hz	132 Hz	155 Hz	151 Hz	155 Hz	153 Hz

Even having the first natural frequency lower than Proposal 1, Proposal 2 was chosen because it had lower transmissibility between the two ends of the tube.

3.2 Suction Muffler

The suction muffler shape optimization considered an acoustical harmonic analysis aiming to reduce the pressure radiated from the muffler.
The harmonic analysis considered the sound speed and the density of the refrigerant inside the muffler as the scope. The imposed boundary conditions are a vibrating panel at the region close to the suction valve and a radiation impedance at the nozzle placed close to the housing and / or at the exit of the little chamber that contains the equalization holes (as per the current muffler solution).
Figure 6 shows a typical mesh for the muffler of the current VSRC with the imposed boundary conditions.

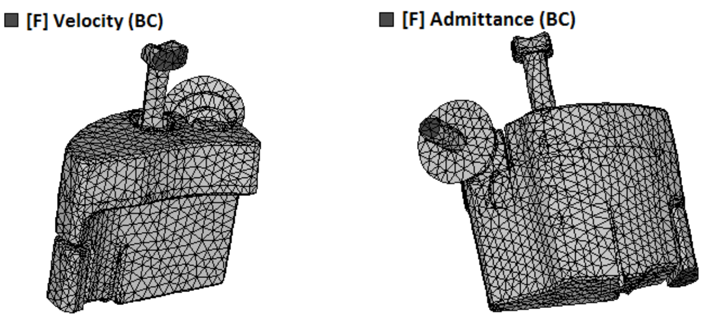

Figure 6: Suction Muffler – Harmonic Analysis.

For the radiation impedance of an unflanged pipe Z_r, a typical equation is the one presented by Levine and Schwinger (1948):

$$\frac{Z_r}{Z_c} = 0.25(ka)^2 + j0.6133(ka) \tag{1}$$

where $Z_c = \rho_0 c_0/S$ is the characteristic impedance with $\rho_0$refrigerant density, c_0 speed of sound and S surface area of the tube, $k = \omega/c_0$ is the wave number, w is the angular frequency and a is the tube radius.
After some steps of optimization, it was possible to generate three charts, comparing the current suction muffler response to the chosen proposal.

- Figure 7 compares the response measured at the suction pipe. This chart shows how the suction pulsation will be reduced in the new generation of suction muffler.
- Figure 8 compares the response measured at the opening of the little chamber close to the equalization holes (current muffler) to the response measured at the nozzle close to the housing (proposed muffler). This chart is used to understand how the muffler can excite the compressor gas cavity and, as a consequence, how the suction muffler can influence the compressor airborne noise.
- Figure 9 is similar to Figure 8, but focused on the range between 300 and 600 Hz; the frequency axis is expressed in narrow bands and not in third octave bands, as in Figures 7 and 8.

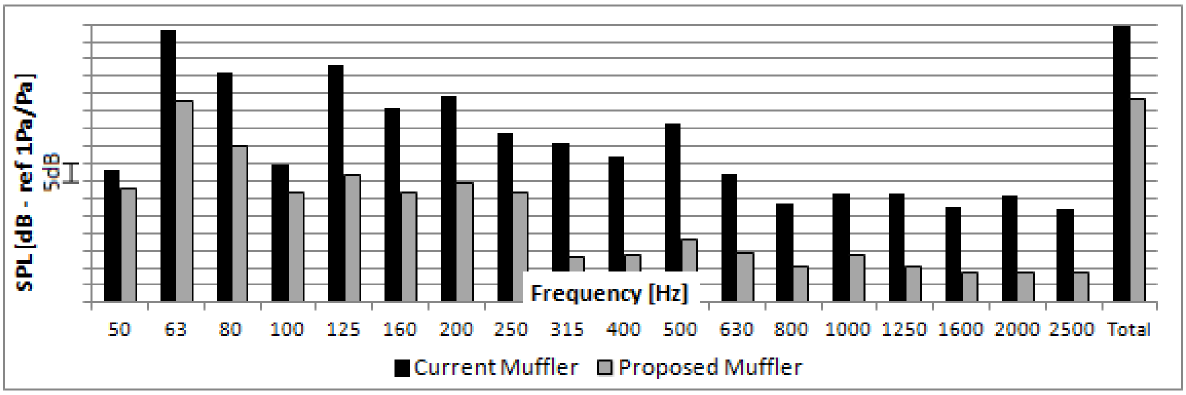

Figure 7: Frequency Response Comparison (old vs. new muffler) – influence on the suction pulsation.

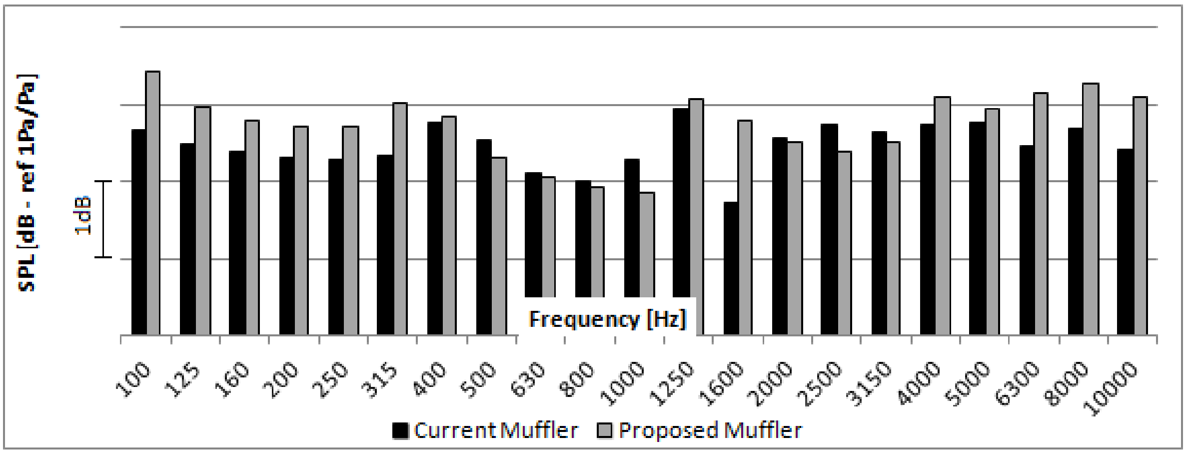

Figure 8: Frequency Response Comparison (old vs. new muffler) – influence on the airborne noise (1/3 octave band).

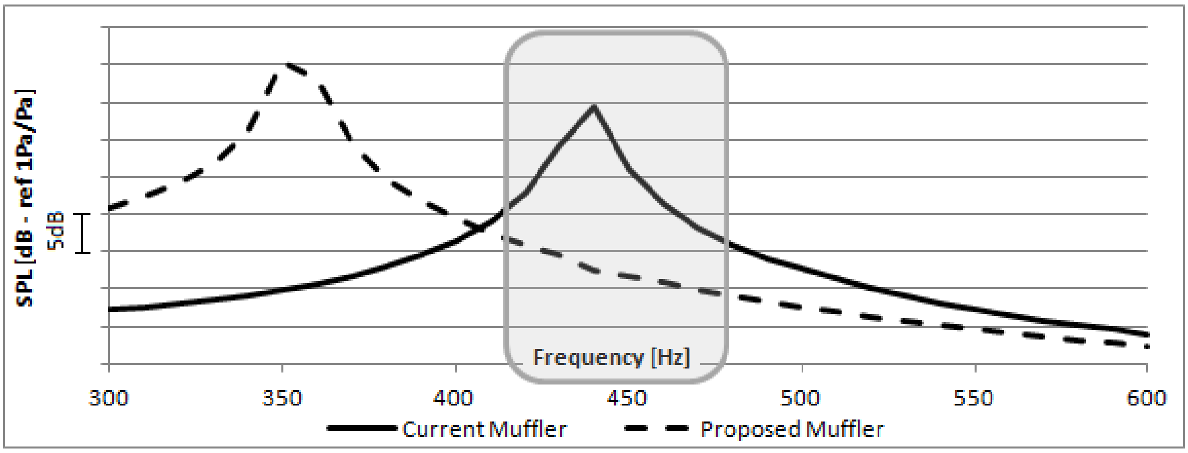

Figure 9: Frequency Response Comparison (old vs. new muffler) – influence on the airborne noise (narrow band).

Besides having lower suction pulsation levels, it's expected with the new muffler similar noise results, considering the suction muffler as a path. However, with the proposed suction muffler, 400 and 500Hz noise must be better because the region in gray in Figure 9 was optimized and for the current generation of VSRC this is the main contributor in terms of noise at 400 and 500 Hz third octave bands.

3.3 Housing

The VSRC housing was optimized by the modification of two different parameters of the housing: their thickness and the mass of the oil available in the bottom shell.

These two parameters were evaluated using two types of simulation: a coupled modal analysis and a harmonic analysis, both with a commercial finite elements code.

The coupled modal analysis considers a structural modal analysis to analyze the natural frequencies and the modes of the housing and an acoustical modal analysis to calculate the acoustical modes of the oil in the sump. For the housing, the material was the standard steel: density of 7850 kg/m^3, Young's module of 200 GPa and a Poisson coefficient of 0.33. The properties that characterize the oil were the sound speed and the density, both similar to the ones of the water at 25°C and 1 atm (1500 m/s and 1000 kg/m^3, respectively).

After performing the modal analysis of both housing and oil, a coupled modal analysis is done, considering a strong coupling between the housing and the oil, as it was presented by Cordioli (2010).

Table 2 shows the difference in terms of natural frequencies for the housing of the current VSRC and the new compressor. The thickness and the oil volume are normalized by the ones presented in the current VSRC.

Table 2: Natural frequencies for different housing configurations.

# of Natural Frequency	Thickness = 1 Oil Volume = 1 (Current VSRC)	Thickness = 1.2 Oil Volume = 1	Thickness = 1.2 Oil Volume = 0.8 (New VSRC)
1	2487 Hz	2729 Hz	2787 Hz
2	2495 Hz	2788 Hz	2990 Hz
3	2660 Hz	2881 Hz	3057 Hz
4	2815 Hz	3025 Hz	3177 Hz
5	2833 Hz	3065 Hz	3209 Hz
Average	2658 Hz	2898 Hz	3044 Hz

It's observed that the thickness increase contributes for a raise of the natural frequencies of the housing of 240 Hz, in average. If one compares these results to a typical plate equation, as mentioned by Blevins (1995), one sees in this case that the natural frequency is proportional to the thickness of the plate. However, Blevins (1995) also mentioned that, for a cylindrically curved panel (more similar to a housing than a flat plate):

$$\omega_{ij}\Big|_{\substack{curved\\panel}} = \left[\omega_{ij}^{2}\Big|_{\substack{flat\\plate}} + \frac{\alpha_{ij} E h}{R^2 (1-\upsilon^2)} \right]^{0.5} \quad (2)$$

where ω_{ij} are the circular natural frequency, i is the number of half-waves along the circumferential axis of the curved panel, j is the number of half-waves along the longitudinal axial of the curved panel, E is the modulus of elasticity, h is the plate thickness, R is the radius to midsurface of the curved panel, γ is the mass per unit area of panel (μh, where μ is the material density), α_{ij} is the dimensionless constant that is a function of the boundary conditions and v is the Poisson's ratio. That means for a housing the natural frequencies don't raise proportionally to the thickness because there is a constant that reduces the rate that the natural frequencies increases.
The way the natural frequencies reduces with the oil volume increase, especially the ones related to the bottom housing modes, in turn, is well explained by the simple equation of a system with 1 degree of freedom:

$$f_n = \frac{1}{2\pi} \left[\frac{k}{m} \right]^{0.5} \quad (3)$$

where f_n is the natural frequency, k is the term related to the stiffness of the housing (proportional to the thickness) and m is the oil mass.
Another important point is to verify that when the housing natural frequencies increase, the noise tends to decrease. So, a simulation of the radiated noise was prepared (harmonic analysis up to 10 kHz third octave band end), considering the three housing configurations of Table 2 and excitations of 1 N at each suspension spring position and where the discharge connector is placed, as showed by Figure 10.

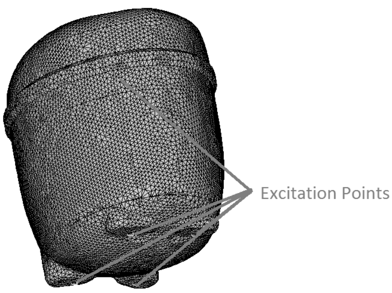

Figure 10: Mesh prepared to simulate the harmonic analysis of the housing.

As a result of these simulations, it was calculated the projected sound power level. Taking as the reference the sound power level and the housing of the current variable speed reciprocating compressor, it's possible to see the noise reduction projected for the new version of this compressor in Figure 11.

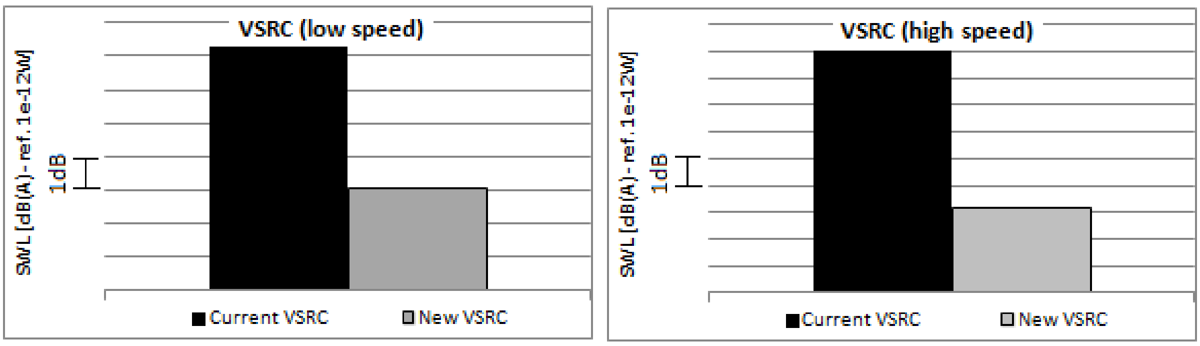

Figure 11: Projected noise reduction for new VSRC due to housing optimization.

As observed, at low speed it's expected a reduction higher than 4dBA of noise and at high speed more than 6dBA reduction for the new VSRC, if the kit is maintained.

4. COMPARISONS BETWEEN VSRC OLD AND NEW GENERATIONS

The following sections are intended to show the acoustic improvements of the new generation of VSRC when the compressor standalone (not assemble to the cabinet) is analyzed.

4.1 Airborne Noise

The airborne noise measurement analyzed here considers a standalone compressor placed in a reverberant chamber. Two hoses link the suction and the discharge connectors to a panel that imposes evaporating and condensing pressures to the compressor, simulating a condition similar to the one that a cabinet imposes.
The noise is measured by a microphone that rotates around their own axis, 1 meter far from the source.
Results in Figures 12 and 13 are presented in terms of sound power level (SWL), normalized by the curve A and comparing the new and the current generation of the VSRC for the lowest and the highest speeds that the compressors run, respectively, considering AHAM (Association of Home Appliance Manufacturers) condition: evaporating temperature = -23°C; condensing temperature = 40.5°C.

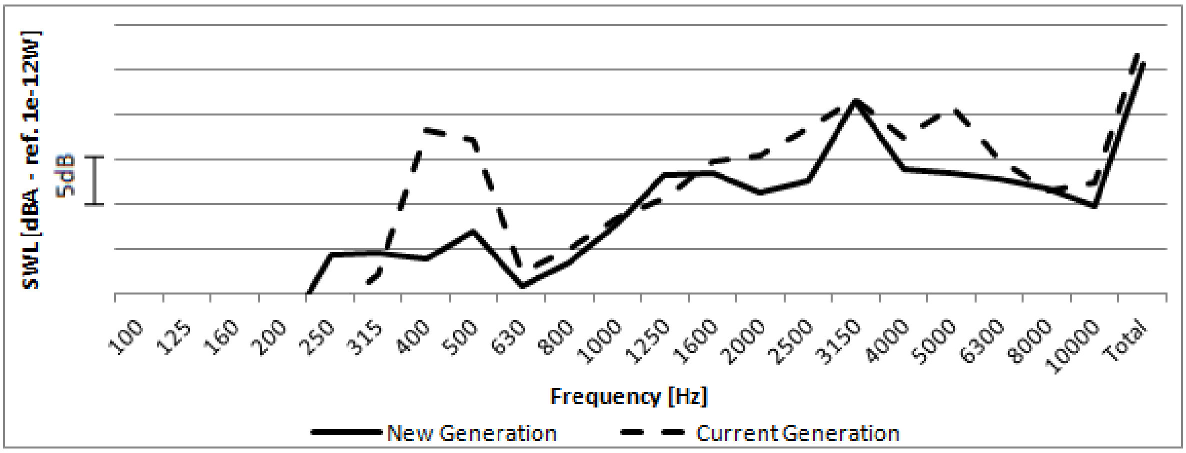

Figure 12: Noise comparison (low speed – AHAM condition).

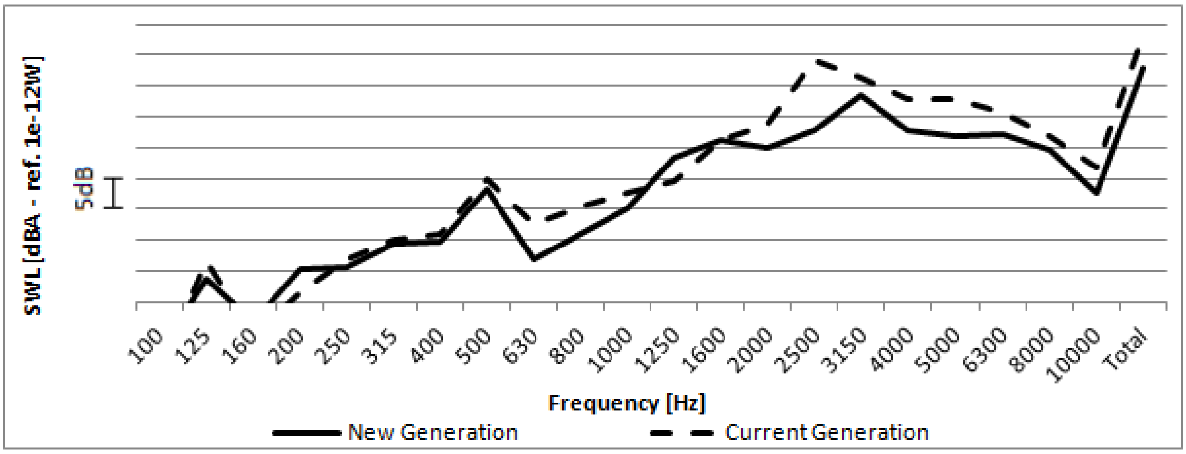

Figure 13: Noise comparison (high speed – AHAM condition).

First of all, it's clear, especially at low speeds, a huge improvement (around 10dB) for 400 and 500Hz bands for the new generation of VSRC, due to optimization of the suction muffler response at this region of the spectrum. At low speeds, the housing optimization also contributed to a better high frequency noise and as consequence, to an overall noise 3dB lower for the new generation.
At high speed, again the housing optimization can be remarked, contributing to 5dB noise reduction for the new generation of VSRC.

4.2 Suction Pulsation

The suction pulsation measurement is done at the same time that the compressor airborne noise is measured, but it's measured in terms of sound pressure level (SPL).
The equipment necessary for this measurement is very simple: a pressure transducer is placed in the suction hose 50 centimeters away from the compressor.
A comparison in terms of suction pulsation between the new and the current generation of VSRC with the compressor running at low speed (AHAM condition) is similar to the one showed in Figure 7, where is possible to see the advantage of using the compressor gas cavity as a chamber that attenuates the suction pulsation.

4.3 Low Frequency Vibration

The low frequency vibration is measured here in terms of velocity (mm/s) using accelerometers placed on the center of three faces. The direction that the velocity is evaluated by one of these accelerometers is orthogonal to the direction the other accelerometers: the longitudinal direction is the one parallel to the piston movement, the top direction is normal to the base where the compressor is placed and the transversal is orthogonal to the other directions.
The analyzed compressor is supported by typical rubber grommets, which are components that link the base plate of the compressor to the base plate of the cabinet. Besides, all this system is placed on a rigid base, as showed in Figure 14.

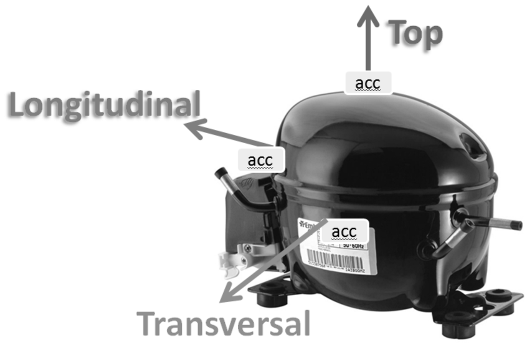

Figure 14: Schema showing the position of the accelerometers (acc) placed on the compressor.

The following results (Figure 15) compare the vibration, normalized by the highest velocity, in these three axes of the current and the new generation of this VSRC, for 27 different speeds.

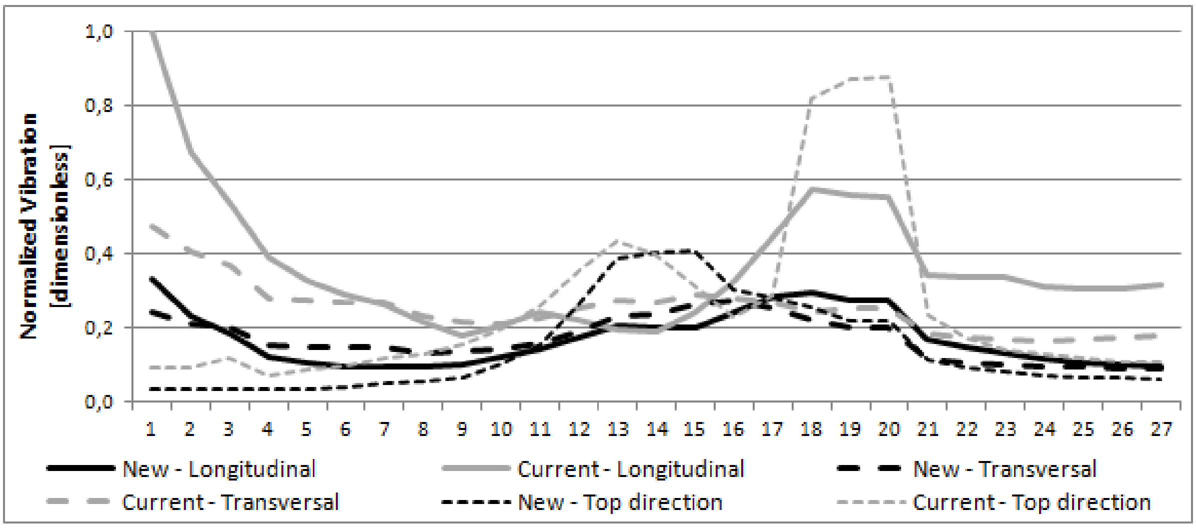

Figure 15: Vibration of the old and new generation of VSRC.

One can conclude that the new generation is better than the current one for almost all the tested speeds, mainly because of the new concept of the suction filter, that doesn't require a suction rubber connector. Besides, a huge improvement for the 18^{th}, 19^{th} and 20^{th} speeds were obtained because of the re-design of the discharge tube: in the current generation this speed range is "jumped" by the current inverter.

5. CONCLUSIONS

- This paper presented previous studies carried out in order to better understand the influence of the suction muffler layout and the discharge tube geometry for the current generation of VSRC and their consequences at the appliances noise that used this generation of compressor.

- For a discharge tube modal analysis, the set mesh, material properties and boundary conditions allowed results similar to the experiment, considering the first 3 natural frequencies of the tube.

- The optimization of the suction muffler response between 425 and 475 Hz had as a consequence a new generation of VSRC with significantly better 400 and 500 Hz noise results than the current generation of VSRC.

- The optimized housing allowed 3dB noise reduction at low speeds and 5dB at high speeds for the new generation of VSRC.

- As a result of some systems optimization, the new generation of VSRC presents better noise, suction pulsation and low frequency vibration levels. Besides this, with this generation it will be possible to run the compressor at the full range of speed that the inverter permits, without a jump that avoids the excitation of some internal resonance.

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