VIBRATION AND NOISE REDUCTION OF RECIPROCAL COMPRESSOR BY STRUCTURAL DAMPING APPLICATION

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Abstract. *Recently, noise and vibration characteristics have become important factors for choosing electric home appliances. Especially, low noise and vibration are an essential requirement for the quality of refrigerators. The acoustic radiation of the compressor installed in the household appliances can be a significant contributor to the overall noise level. A major portion of the measured noise is originated from structural vibration of the compressor housing, which is the most important component between the noise sources and the receiving human ear. This paper presents studies carried out on reciprocating commercial compressor in the attempt to reduce the vibration of the compressor housing increasing the structural damping with the use of high damping materials (viscoelastic materials) applied on the external surface of the compressor. By means of experimental procedures (vibration and noise measurements, vibration mapping and modal analysis), the mainly locations for application of the damping material were defined and the final issue of this work presents low vibration and noise levels due to the application of such damping material.*

Keywords*: Vibration Reduction; Noise Reduction; Reciprocal Compressor; Structural Damping*

1. INTRODUCTION

Recently, noise and vibration characteristics have become important factors for choosing electric home appliances. Especially, low noise and vibration have become an essential requirement for the quality of refrigerators (Hwang et al, 2006).

The acoustic radiation of the compressor installed in the household appliances can be a significant contributor to the overall noise level. A major portion of the measured noise is originated from structural vibration of the compressor housing, which is the most important component between the noise sources and the receiving human ear.

The noise of a small hermetic reciprocating compressor used for household refrigerators is mostly radiated by the vibration of the compressor shell. Therefore, an effective approach to the noise problem is to provide a compressor shell structurally less responsive to the vibration source (Saito et al, 1980).

Noise generated by a compression mechanism may propagate through the refrigeration gas in the shell and radiated externally through the shell. Vibration from the compression mechanism may resonate with compressor components, thus transmitting resonant noise externally via the shell. The vibration may also propagate to the shell via the compressor discharge and suction system and support system and resonate with the shell, thus transmitting resonant noise from shell (Kawai et al, 1996).

How can be seen, hermetic compressor's housing is the main radiator of the noise and vibration inner to the compressor. All compressors' transmission paths reach a final frontier in the compressor housing. For that reason all resulting noise is radiated by the housing surface. Structural damping increasing is an important ally for surface radiated noise reduction. One of the main techniques to increase structural damping is the use of high damping materials (Oliveira, 2006).

Therefore, in this work, a study was carried out on reciprocating commercial compressor in the attempt to reduce the noise and vibration of a hermetic compressor using a high damping material, which was applied on the external surface of the compressor. This paper has as purpose the study of damping material application effect on compressor housing to minimize structural vibration and consequently, radiated noise.

2. METHODOLOGY

First, a vibration mapping was carried on the compressor's surface. In the Fig. 1a and 1b is shown the mesh used for the vibration mapping. Vibration signals were acquired using accelerometers' at 113 points of compressor housing, during 10 seconds each one. After analyze the vibration results, a high damping material coat was applied on the meshed surface specifically in the areas of higher vibration levels.

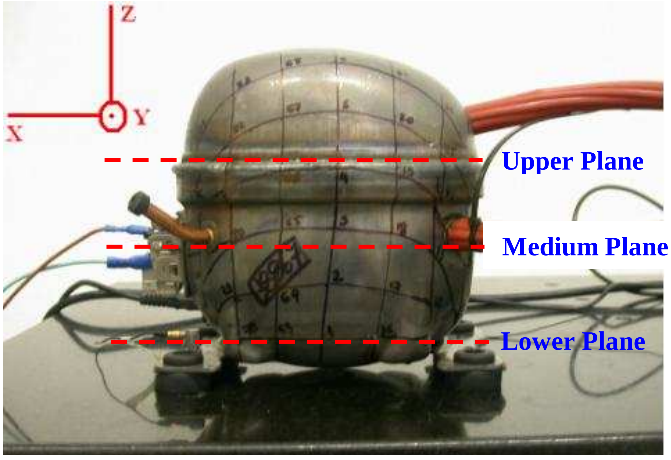

Figure 1a. Compressor front view

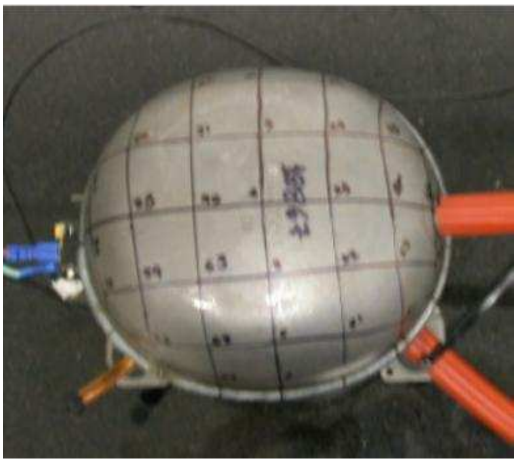

Figure 1b. Compressor top view

The damping material had 1 mm thickness, which is the value specified by the supplier to obtain good results of damping.

Frequency response function (FRF) of higher vibration areas was achieved with the purpose of to verify if higher acceleration levels agree with resonances of the FRF. In order to confirm the relation between the spectrum achieved in the points of high vibration levels on the housing and structural resonance (punctual FRF obtained in the same points), the Fig. 2 is shown below where the red straight line indicates the common frequency between both.

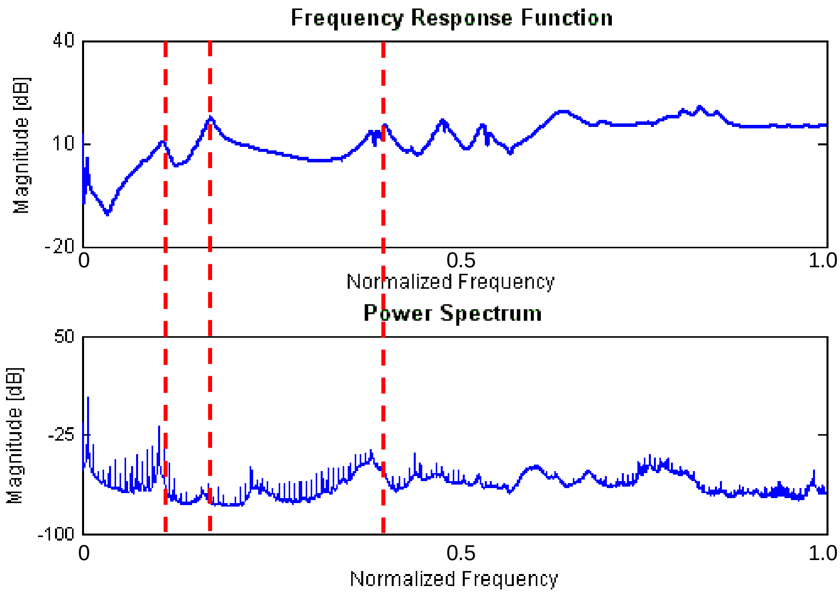

Figure 2. Frequency Response Function and Power Spectrum of acquired signal at one point located on compressor support bracket

In accordance with Fig. 2, the highest vibration energy levels occur in resonance regions, if the structural damping increases in this area, it will occur vibration reduction mainly in these regions. Due this fact, the material damping was applied only in higher vibration areas.

In normal conditions of operation, there are temperature changes on the compressor housing. In order to verify the damping material performance in function of temperature changes, which occurs in practice, the vibration signal were acquired on a compressor housing point at 25°C, 35°C, 45°C, 55°C and 60°C. The compressor operated under temperatures varying since 25°C until 60°C (steady temperature), where a flexible thermo coupler was used.

Finally, the areas with damping material were evaluated in respect to the noise radiation when compared with the surface without damping material. Then, the directivity pattern of the compressor was determined inside a semi-anechoic chamber before and after application of damping material on the surface. Ten seconds of sound pressure signal were acquired with a sample rate of 32768 Hz, at three planes like showed in Fig. 1a (lower plane, medium plane and upper plane). Two microphones were used, varying yours positions between 0 to 360 degrees with angle increment of 10 degrees (in turning of the z axes) in each plane.

4. EXPERIMENTAL RESULTS AND DISCUSSION

4.1. Vibration levels before and after viscoelastic material application

In Fig. 3, 4, 5, 6 and 7 are presented the normalized vibration amplitude in respect to maximum vibration values (m/s^2) before material damping application. The figures correspond, respectively, to the top view, left side view, front view, right side view and back view of the compressor. The sketch showed in the left side of Fig. 3 can help us to understand the position of the meshes in the compressor.

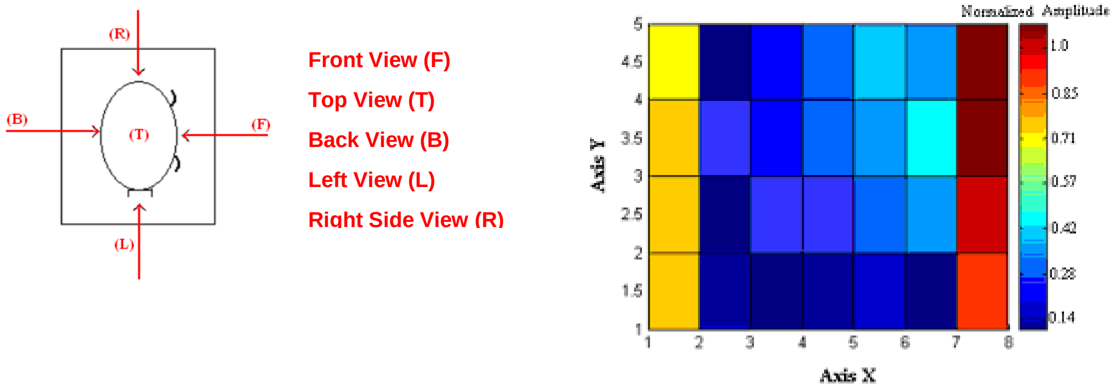

Figure 3. Top view

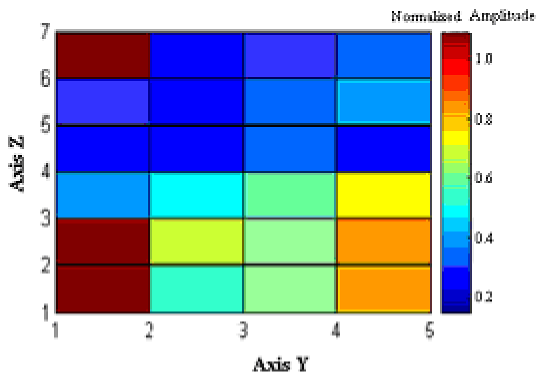

Figure 4. Left side view

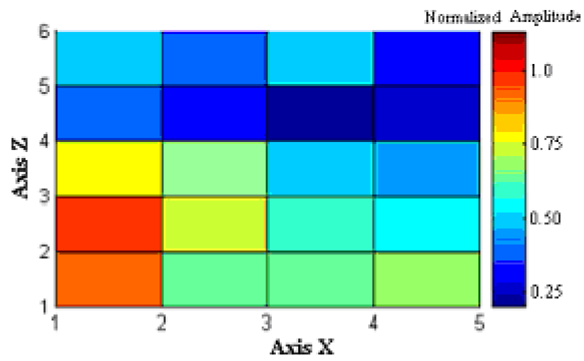

Figure 5. Front view

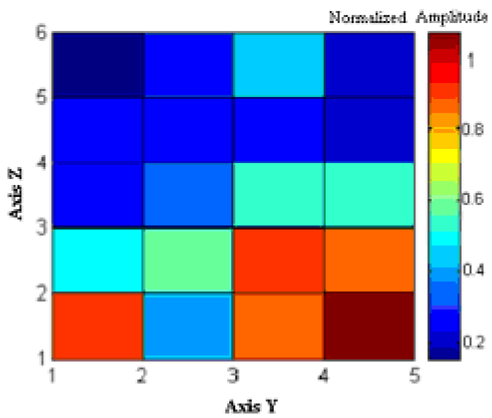

Figure 6. Right side view

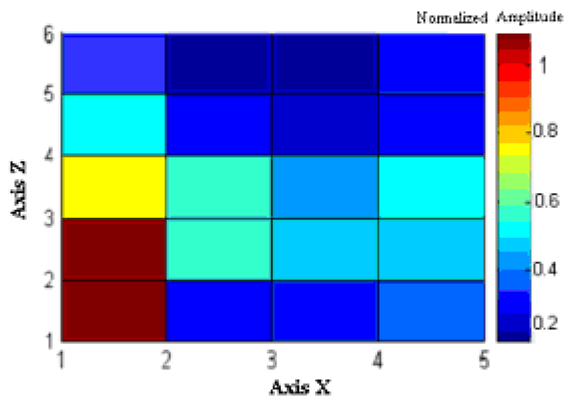

Figure 7. Back view

Analyzing Fig. 3, 4, 5, 6 and 7 we can see that there are some areas where the vibration level is high, which is the areas where the viscoelastic material was applied. We can see that doesn't have isolated points with higher vibration, always there are assemblages of points forming an area.

In accord to the supplier, it is necessary apply 1 mm thickness of viscoelastic material to begin achieve good results of vibration and noise. The performance of the viscoelastic material in function of thickness wasn't contemplated in this work, but it can be verified in next works. In Fig. 8 we can see the surface where the viscoelastic material was applied and the points were the vibration sensor was installed (measurement points).

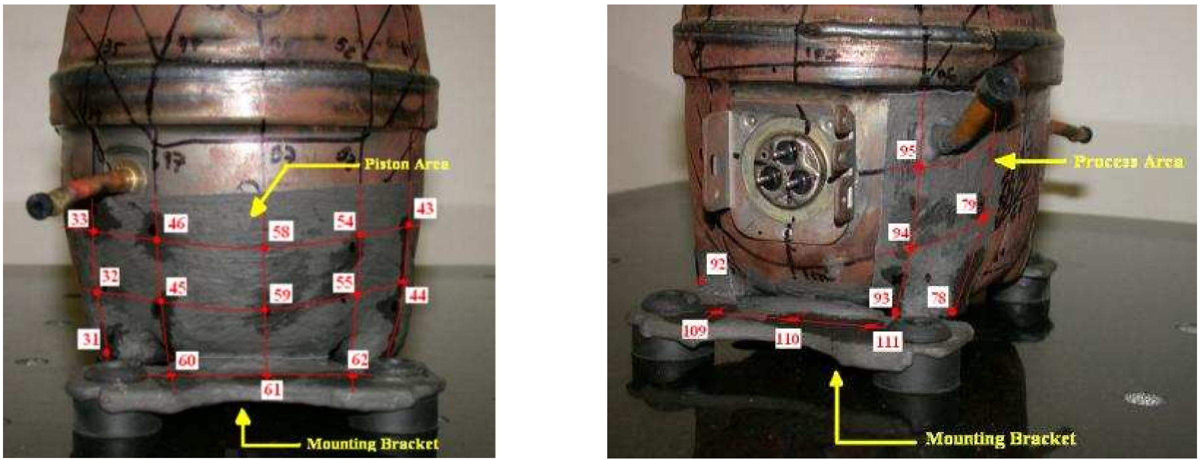

Figure 8. Piston, process and mounting bracket areas with high damping material

We can observe in this figure that the surfaces with high vibration levels is very closer to same process of the interior of the compressor, including the surfaces with direct contact with suspension springs (near mounting brackets) and discharge tubes.

In figure 9 are presented the vibration attenuation level (dB) on the points, which had the higher vibration level before apply the viscoelastic material. We can see the points listed in Fig. 8. The attenuation levels were estimated using the *rms* values in the points before and after applying of the damping material.

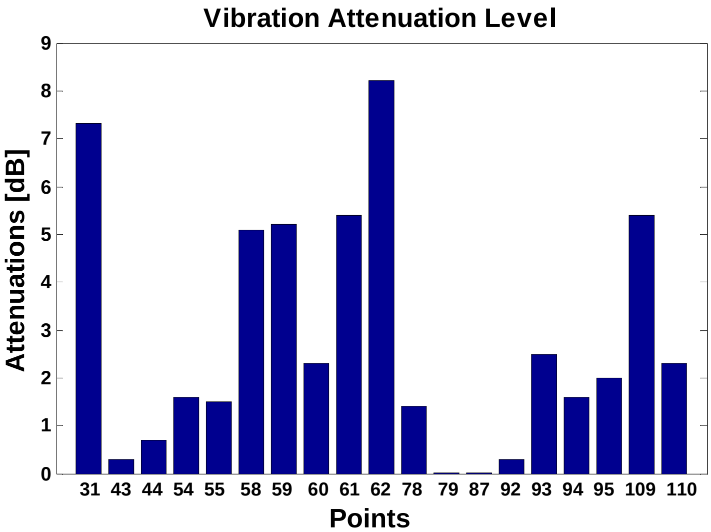

Figure 9. Piston, process and mounting bracket areas with high damping material

It's observed in Fig. 9 that the highest attenuations occurs on points located on the support bracket, on the central region of the piston area and on areas close to suspension spring regions.

The results obtained with the application of the viscoelastic material, which are shown in Fig. 9, emphasize the good performance of this kind of material coat for damping vibrations, mainly at transmission vibration areas.

4.2. Temperature Influence

In Table 2 are presented the RMS levels from vibration signal at different temperatures achieved on the point located in the central region of the piston area (Fig. 8). In a normal operating condition of the compressor, the housing has higher temperature value than the mounting brackets, then, we choose the point specified to monitoring the temperatures changes.

Table 2. Vibration level temperature dependency

Temperature (°C)	25	35	45	55	60
RMS value (dB)	-5.8	-5.5	-4.2	-3.5	-3.4

It's observed in Tab. 2 that damping material lost approximately 3 dB of efficiency in vibration reduction, what must be considered when viscoelastic material will be applied on compressor housing.

4.3. Compressor Directivity before and after high damping material application

In the Fig. 10 until 19 are presented polar graph of normalized sound pressure level (SPL) in 1/3 octave-bands (50 - 10000Hz). Theses SPL were normalized in respect to maximum pressures level in each band.

Polar graphs were presented here only in the 1/3 octave-bands showed below because the other ones didn't present significant differences between sound pressure levels before and after damping material application.

The sound directivities were estimated for all three planes (lower, medium and higher), but here is showed only the results obtained in the lower plane due to the areas with higher vibration level occur on lower regions of the compressor, like the mounting bracket.

The sketch showed in the left corner in Fig. 10 (top view of the compressor) can help us to viewer the position of the compressor in relation to the angles of the polar plot.

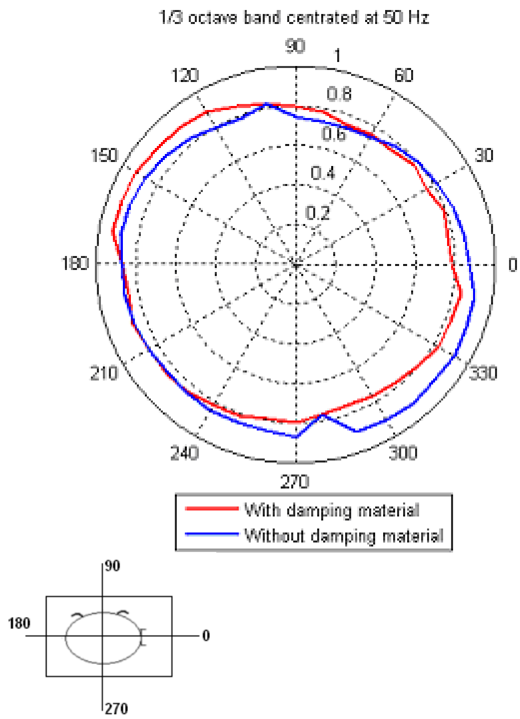

Figure 10. Polar plot of normalized sound pressure level at 50 Hz 1/3 octave band.

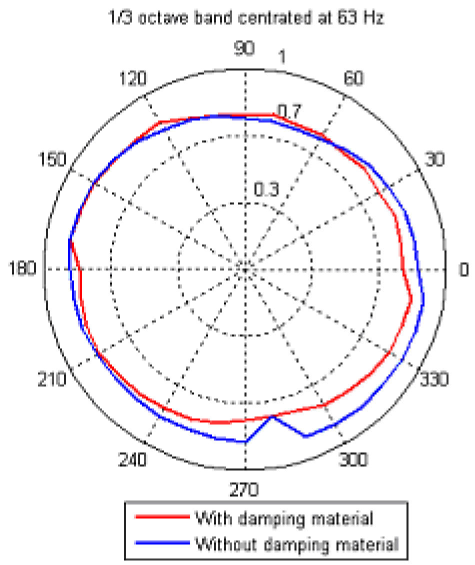

Figure 11. Polar plot of normalized sound pressure level at 63 Hz 1/3 octave band.

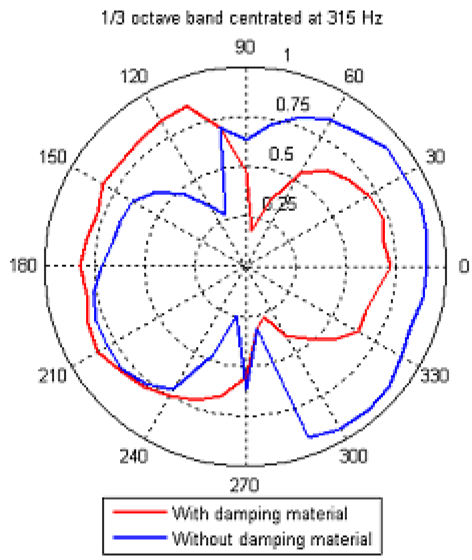

Figure 12. Polar plot of normalized sound pressure level at 315 Hz 1/3 octave band.

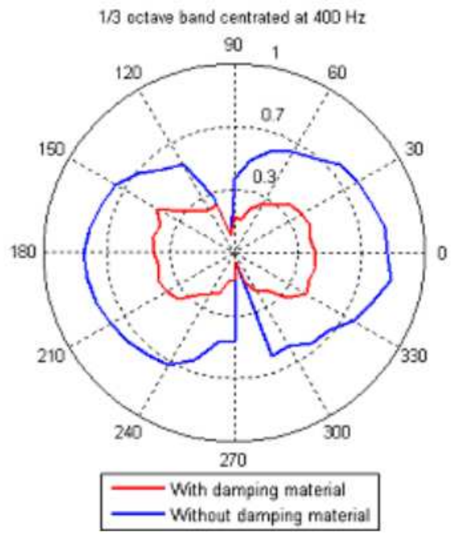

Figure 13. Polar plot of normalized sound pressure level at 400 Hz 1/3 octave band.

Figure 10, 11 e 12 shows attenuations of 5 dB, 8 dB e 4 dB in the process regions (300^o until 330^o), where was applied structural damping material. At 400 Hz band, an attenuation of 10 dB occurs in both lobes of Fig. 13, due to influence of material at all areas with damping material.

It is possible to note in Fig. 10 and 11 that the compressor has the behavior of an omnidirectional source, although, when the frequency increase, it is possible to observe the appearance of *lobulus* indicating directionality on determinate region.

For 500 Hz and 1000 Hz (Fig. 14 and 15) the attenuation is the order of 12 dB and 4 dB, respectively, is achieved in preferential regions of radiation noise.

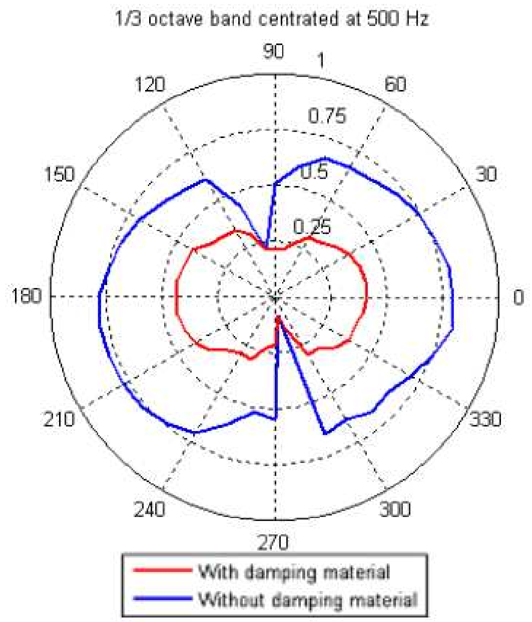

Figure 14. Polar plot of normalized sound pressure level at 500 Hz 1/3 octave band.

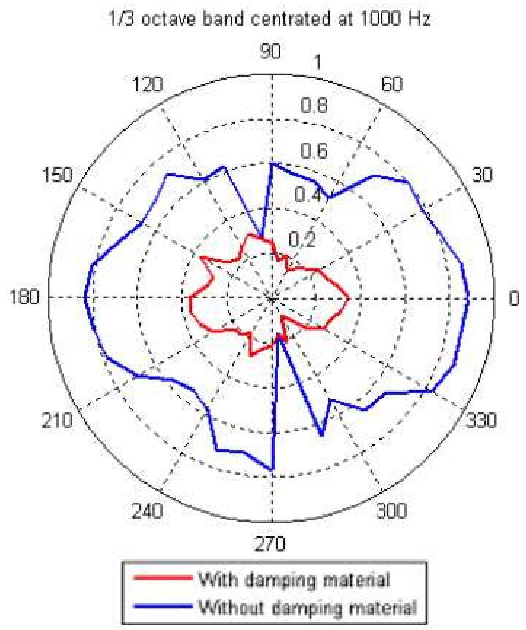

Figure 15. Polar plot of normalized sound pressure level at 1000 Hz 1/3 octave band.

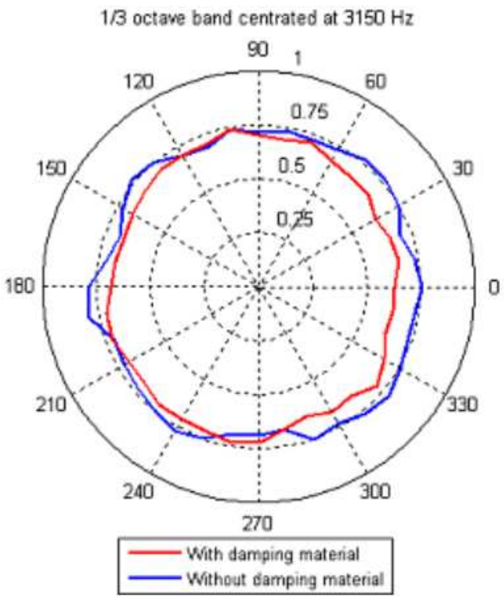

Figure 16. Polar plot of normalized sound pressure level at 3150 Hz 1/3 octave band.

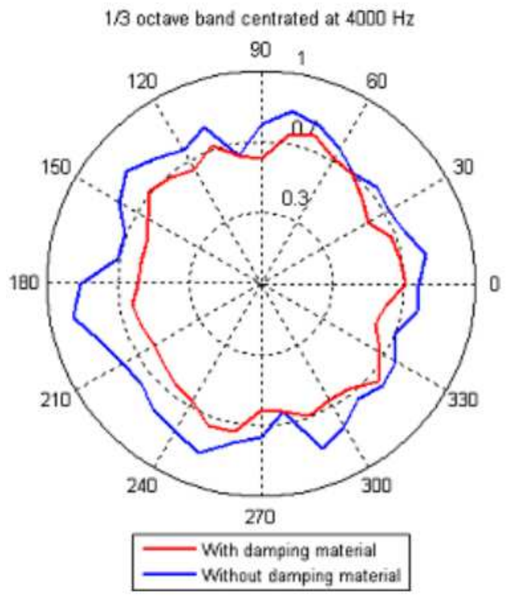

Figure 17. Polar plot of normalized sound pressure level at 4000 Hz 1/3 octave band.

Figure 16 shows that the attenuation obtained at 3150 Hz 1/3 octave band attenuation is close to 5 dB. The attenuation obtained at 4000 Hz (Fig. 17) and 10000 Hz (Fig. 19) 1/3 octave bands was almost 4 dB and at 8000 Hz 1/3 octave band the attenuation is equal 7 dB.

Upper the frequency band of 3150 Hz we can note that the attenuation achieved was along all the 360 degrees in turn of the z axes, so, the compressor starts having behavior of monopole source again, like in lower frequencies.
It is important to have knowledge about noise directivity of the compressor, since it is mounting close to the fridge wall which is like a barrier, and depending on the directivity is possible to take the compressor in a better position aiming the noise mitigate to the receiver ear.

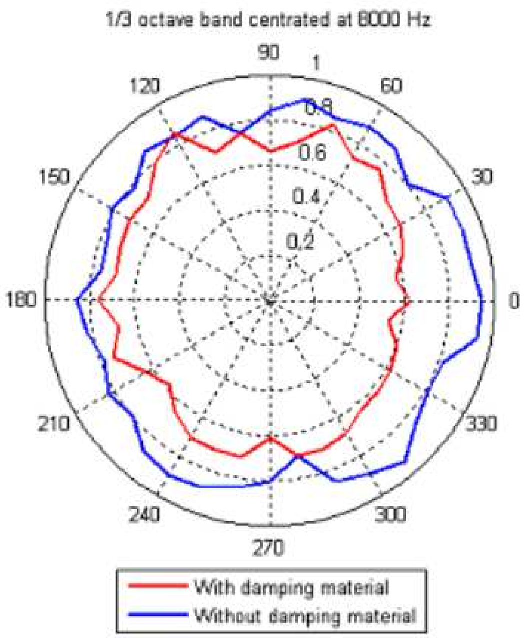

Figure 18. Polar plot of normalized sound pressure level at 8000 Hz 1/3 octave band.

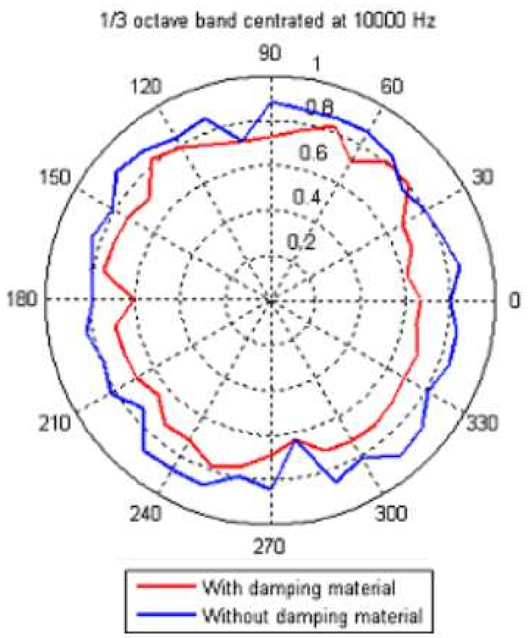

Figure 19. Polar plot of normalized sound pressure level at 10000 Hz 1/3 octave band.

5. CONCLUSIONS

The main conclusions of this work are:

- Low levels of vibration were obtained on the compressor housing when using a viscoelastic material applied in areas of the compressor surface with high vibration.
- The highest vibration attenuations occur on points located on the support bracket, on the central region of the piston area and on areas close to suspension spring regions.
- The damping material lost about 3 dB of your attenuation efficiency when the temperature increases.
- About noise directivity of the hermetic compressor, the results shows that lower 63 Hz 1/3 octave band the compressor is similar a monopole source, upper 315 Hz until 3150Hz 1/3 octave band we can see the contours of the directivity plot like a dipole source, and upper 4000Hz 1/3 octave band the directivity comeback to approximate of a monopole source.
- The noise attenuation gotten with the viscoelastic material depends of the frequency band and the position of the microphone in the plane sub evaluation. So, to defined positions, an attenuation of almost 10 dB was achieved.

6. ACKNOWLEDGEMENTS

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7. REFERENCES

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