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EXPERIMENTAL MODAL ANALYSIS OF A NEW SHELL FOR
DOMESTIC REFRIGERATION COMPRESSORS.

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Abstract

This paper reports about experimental investigations conducted on shells
for domestic refrigeration compressors: the main goal was to state the
benefits in noise reduction obtainable with shells having surfaces of high
curvature.

Measurement data was collected and analysed employing experimental
modal analysis techniques which are briefly described.

The results obtained from the experimental modal analysis have been useful
to understand the noise emission mechanism and to tune a finite element
model of a new shell for hermetic compressors to be employed in appliances
utilizing ozone friendly refrigerants.

NOMENCLATURE

Y Input signal
Y Output signal
H Frequency response function
G\text{xx} Positive frequency autospectrum
G\text{xy} Positive frequency cross spectrum
Y\text{yy} Coherence function

Fig. 1 - A typical reciprocating compressor.
Introduction

Frequency response function measurements were taken on a compressor shell consisting of a hermetic container made of two metal parts joined under static pressure and welded together along a closed curve. These two parts which the shell is made up of, are two circular or oval cylindrical bodies with domed end plates of rather complicated curvature.

Usually the zones of the shell surface are called, in a self-explanatory way, Bottom, Top and Side Walls: in the following text they will be referred to by these names. On the Side wall surface four holes are made: three of small diameter for welding Bundy or Copper tubes, the fourth, slightly bigger one, is used to weld the electrically insulated plug of the power supply of the compressors electrical motor (the terminal block of figure 1). Furthermore, on the Bottom of the shell two thick metal sheets of rather simple geometry are welded, usually at suitable points to fasten the compressor to the refrigerator cabinet.

Once the compressor is closed inside its shell and in a steady running condition, it is the shell surface that emits most of the sound (called Noise) and that gives it the characteristic signature, i.e.: the shape of the Noise Spectrum. For this important reason a large amount of the development work for a new compressor shell is devoted to the analysis of the acoustic behaviour of the shell by means of experimental and theoretical Modal Analysis.

For the development of the new compressor shell a step by step method was followed: first of all a Finite Element Model was made from a shell having all the requirements of good geometry for noise reduction purposes, once a reasonable shape was found, Experimental Modal Analysis was carried out on a prototype shell and results were compared in the effort to get a more accurate model and to achieve better results.

Fig. 2 - Experimental apparatus.

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Experimental modal analysis was carried out on an empty closed shell with a built-in electric plug but without connection tubes. The experimental apparatus is shown in figure 2: soft rubber strings were used to suspend the shell; a piezo-electric 1-gram accelerometer (with charge amplifier) measured accelerations; and an impact hammer with built-in force transducer excited the structure. An alternative method of excitation, based on an electrodynamic shaker, was also tried, but the impact hammer gave better performance and was ultimately preferred.

The choice of measurement and excitation points was made by exploiting the a priori knowledge of the existence of two classes of modes: the former primarily involving the side walls, the latter the bottom and top. Measurement points were placed on the three major cross-sections of the shell: the first cutting the cylindrical part of the housing, the second placed in the plane of symmetry (cutting top and bottom caps), and the third normal to the other two.

Excitation points were placed at the intersections of the three cross-sections and on the "fillets" joining top and bottom to side walls. The transducers were calibrated one respect to the other and International Standard Units were used. Tests were carried out to identify the frequencies of the rigid body modes of the system (shell - rubber spring - heavy beam).

By employing the impact technique, no overlapping of the rigid body modes frequency with the shell modes frequency was detected, the former being far apart from the latter. Modal testing was carried out on different cross sections of the structure and Frequency Response Functions (FRFs) were collected for an amount of 200 points on the shell. Coherence functions were carefully examined to control the input signal when it had too low an energy and when the frequency resolution was too poor, in the attempt to improve the quality of the measurements.

Another fact we kept in mind was the following: the whole set of mode shapes of the shell could not be fully excited by an input signal acting upon those points which allowed an exciting spectrum of constant and rather flat energy density all over the frequency bandwidth of interest. On the other hand, input points which allowed the excitation of the low frequency modes of the shell, yielded a set of FRFs that over-estimated the resonances located in the high frequency region of the spectrum of interest.

For this reason a compromise was made: sets of FRFs were collected consisting of four subsets, each belonging respectively to four
different reference points: two lying on a flat surface and two on the high curvature "fillets".

Acquired data, usually inertance, was stored in the following format:

\[ R_{xy} = \frac{G_{xy}}{G_{xx}} \] for the Frequency Response function, assuming noise at output and maintaining \( G_{xx} \) as constant as possible;

\[ Y_{xy}^2 = \left| \frac{G_{xy}}{G_{xx}} \right|^2 \] for the Coherence function.

A smoothing window was selected and applied to output signals to fulfill the requirements of a correct FFT application for the response time frame length.

Several tests were effected with different frequency resolutions and bandwidth setups. Finally, it was found satisfactory to employ an eight Hz resolution for a 4096 Hz frequency baseband. This setup allows a better storage of data blocks in the Computer System when the number of actual FFT lines is taken into account. Figure 3 shows a FRF and Coherence function for an 8 Hz resolution. By examining the whole set of stored FRFs a preliminary selection of the frequency range and of the candidate resonances was accomplished. It seemed the best choice to limit the frequency interval of our analysis between 2000 and 3500 Hz and to focus our efforts on the evaluation of modal parameters for peaks located around the frequencies that are listed in table 1. With this approach, the most important range of sound emission of the compressor was taken into account, being absolutely certain that no resonances were present below 2000 Hz in the FRFs spectra. On the other hand, input signals, acquired with the impact technique, showed a decreasing slope in energy density beyond 3500 Hz, preventing enough excitation through the structure.

Fig. 3 - An example of frequency response function and corresponding coherence function.
When all the collected sets of FRFs were found to be reasonably self-consistent all the data was transferred to the computer system for the application of the subsequent modal parameters and shapes extraction method. Various modal parameter extraction techniques for Multi Degrees of Freedom Systems were in turn applied. Surprisingly, no method showed itself to be thoroughly decisive, neither for its ability to resolve all the resonances that were present in the Modal Indicator Function, nor (in many cases) for the evaluation of the residues, so that the analytical function was in close resemblance with the measured FRFs.

![Fig. 4 - Typical FRFs for side walls and domed end caps.](image)

What seemed critical was the difference between the frequency response functions measured on the cylindrical (barrel-shaped) part of the shell and those measured on the top and bottom (spherical) parts. As can be seen from figure 4 the FRFs are varied; the first shows well separated peaks as is typical of the barrel-shaped side walls; the second shows tight coupled peaks in a narrow frequency band. At last we were confident with the results that a typical single degree of freedom approach gave us. The Circle Fit method, implemented in our computer, was then used. This method, as is well-known, is based on the fact that a plot of the out-
of-phase components versus the in-phase components traces a circular arc in an Argand plane (imaginary versus real parts plane) for each resonance.

A circle is therefore fitted to the data in a narrow frequency range around the resonance frequency. Resonance frequency and damping can be estimated by determining the greatest angle between two data points along the circle. Estimation of residue is based on the diameter of the circle (amplitude) and its location in the Argand plane (phase).

Table 1 reports frequencies, maximum amplitudes and dampings of the first six modal parameters which have been extracted.

Figures 6.a and 6.b show the first four mode shapes involving the whole shell. Shapes are shown for the three typical cross-sections plotted in figure 5.

<table>
<thead>
<tr>
<th>MODE NUMBER</th>
<th>FREQUENCY [Hz]</th>
<th>DAMPING</th>
<th>TYPE</th>
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</thead>
<tbody>
<tr>
<td>1</td>
<td>2336</td>
<td>.05</td>
<td>CYLINDRICAL</td>
</tr>
<tr>
<td>2</td>
<td>2857</td>
<td>.14</td>
<td>CYL./SPH</td>
</tr>
<tr>
<td>3</td>
<td>2837</td>
<td>.08</td>
<td>SPHERICAL</td>
</tr>
<tr>
<td>4</td>
<td>2877</td>
<td>.04</td>
<td>CYL./SPH</td>
</tr>
<tr>
<td>5</td>
<td>3109</td>
<td>.03</td>
<td>CYL./SPH</td>
</tr>
<tr>
<td>6</td>
<td>3182</td>
<td>.68</td>
<td>SPHERICAL</td>
</tr>
</tbody>
</table>

Table 1 - Natural frequencies and dampings of the first 6 modes.

Fig.5 - The three cross-sections of the shell.
Fig. 6.a - The first mode shape of the shell; only two cross-section are involved.

Fig. 6.b - The three other mode shapes involving the whole shell.
CONCLUSIONS

Initial experimental analyses did not use any form of a priori knowledge about modes. This lead to less effective placing of measurement and excitation points. They were placed all over the shell. The three-dimensional structure of the data thus collected required more measurement locations, was more difficult to interpret, and gave less information. Numerical simulation, and above all the availability of an interpretative scheme, greatly benefited experimental analysis. This lead to the definition of the three meaningful cross-sections and of the most useful excitation locations. A point to be stressed is the difficulty encountered in correctly identifying the splitted modes of the oval shell. It seems that the second and the fourth mode of fig. 6.b are the first two splitted modes of the oval cross section of the shell.

A final remark concerns the importance of the attained results. As can be inferred from the plot at figure 7, our new developed shell has improved the acoustic performance of Sanussi Elettromeccanica SpA compressors. This characteristic is especially useful for those compressors to be employed with new refrigerants which are supposed to be ozone friendly. Such a plot was obtained by comparing the sound power levels of the same set of compressors (tested by standard acoustic measurements) with the old shell and subsequently, with the new one. As the graph shows, the best results have been obtained between the 1600 and the 2500 Hz range represented in third octave bands.
range the new shell appears to be particularly effective. Overall, the accomplished improvement has permitted a reduction in sound power level by 2 dBA, with the same compressors inside the new shell. Such a result seems promising for the future use of our new compressor shell.

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