2004

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SENSITIVE DEPENDENCE FROM THE CYLINDER HEAD POSITION ON THE COMPRESSOR'S NOISE EMISSION – A NUMERICAL ANALYSIS

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ABSTRACT

The acoustic power radiated by an hermetic household compressor is studied, via a numerical approach, in relation to the distance between the cylinder head and the compressor shell. The boundary conditions have been introduced by direct measurements of the real accelerations on the cylinder head.
The vibro-acoustic model has been improved by considering a double fluid-structure interaction.
The mathematical model has been refined by studying the radiation efficiency of the compressor versus its structural eigenvalues.
The results have been compared with experimental measurements of the radiated acoustic power.

1. INTRODUCTION

During the development of a new product, which showed an unpredicted problem in the 3150 Hz third octave band, an intense campaign of measurements was performed, in order to identify the noise source.
This source was particular, and showed an unique peak at an unique frequency (2950 Hz) (Fig.1).
A direct relation between the acceleration values measured on the cylinder head and the emitted noise was detected.
In order to find a theoretical justification of this phenomena and to study its dependence from the cylinder head-shell distance, a numerical model of the compressor cavity was created and then forced with the experimentally-measured accelerations.
An interesting and useful noise reduction was obtained through simulation and then confirmed through direct sound power level measurement.

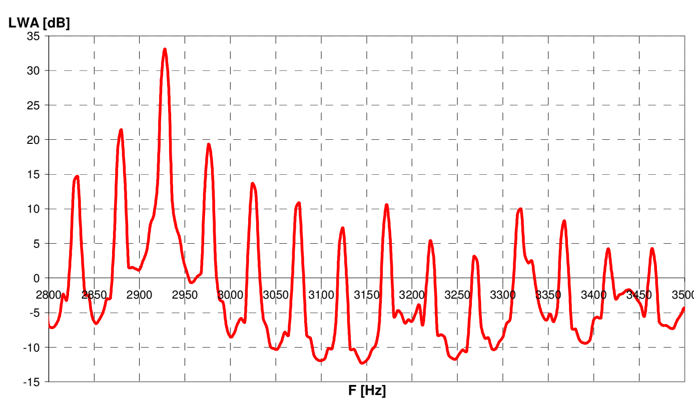

Figure 1: Narrow band acquisition at 3150 third octave band

2. EXPERIMENTAL MEASUREMENTS

2.1 Acceleration Measurements

A set of acceleration measurements was performed, in order to identify the noise source through the evaluation of possible anomalous values for some parameters.
The accelerations were measured in different points on the compressor pump (Fig.2), including some points on the crankcase, on the cylinder head, on the discharge tube and on the brackets.
The signals were acquired and then processed with Lms Cada-X software, in order to get their frequency content.
A peak value at the same frequency of the overall compressor emitted noise was found only in two positions, corresponding to:

1. the cylinder head "front" (point 4 in Fig.2)
2. the cylinder head "rear" (point 2 in Fig. 2)

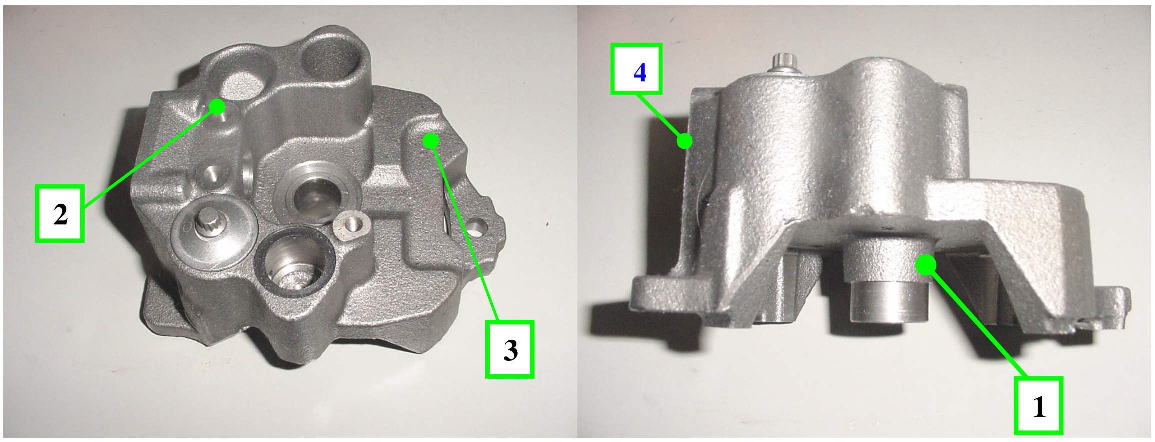

Figure 2: Some positions on the crankcase for accelerations measurement

2.2 Upper shell modal analysis

Experimental and numerical modal analyses of the upper shell were performed, showing good correlation between the results.
A mode in the 3150 Hz third octave band was detected, corresponding to the sound power level peak and to the previously measured accelerations zone (Fig.3).

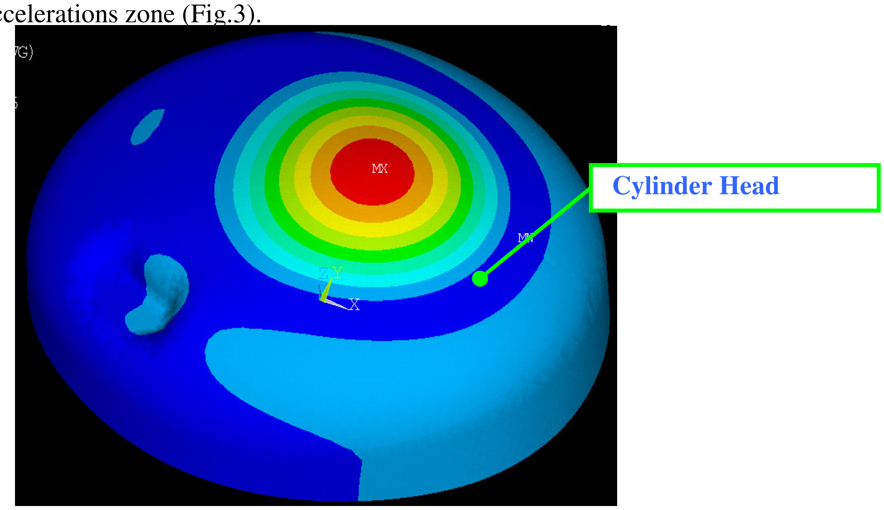

Figure 3: Upper shell's mode in the 3150 Hz third octave band

As there was no possibility to change the upper shell's geometry and to uncouple this mode from the cylinder head's vibrations, another way to operate was followed: we decided to take away, as far as possible, the cylinder head from the upper shell.

3. NUMERICAL SIMULATIONS

3.1 Compressor's cavity modal analysis

The compressor's internal cavity was built through volumes subtraction, meshed and imported in Lms Sysnoise software.
A numerical modal analysis, with gas properties set as in working conditions, was performed and modes were extracted (Fig.4).

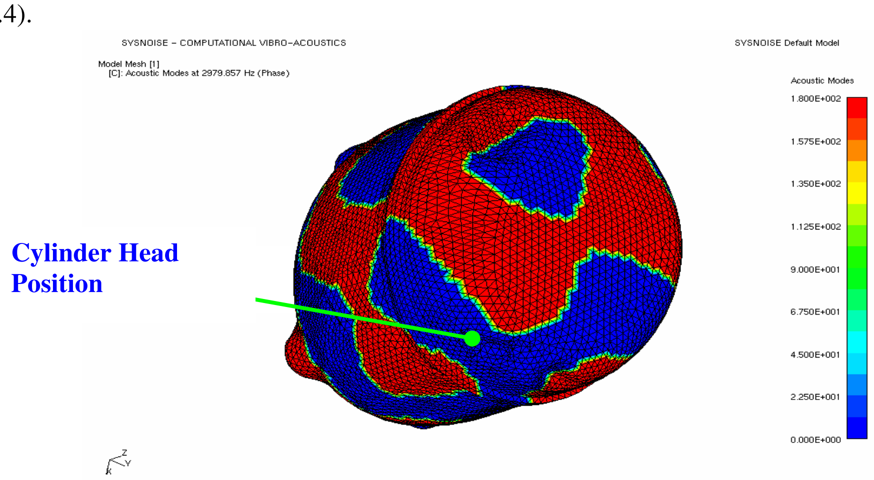

Figure 4: Cavity's pressure distribution at 2979 Hz (closest step to 2950 Hz)

3.2 Compressor's cavity forced analysis

The compressor's cavity was then forced with the experimentally-measured accelerations (previously turned into velocities). (Fig.5)

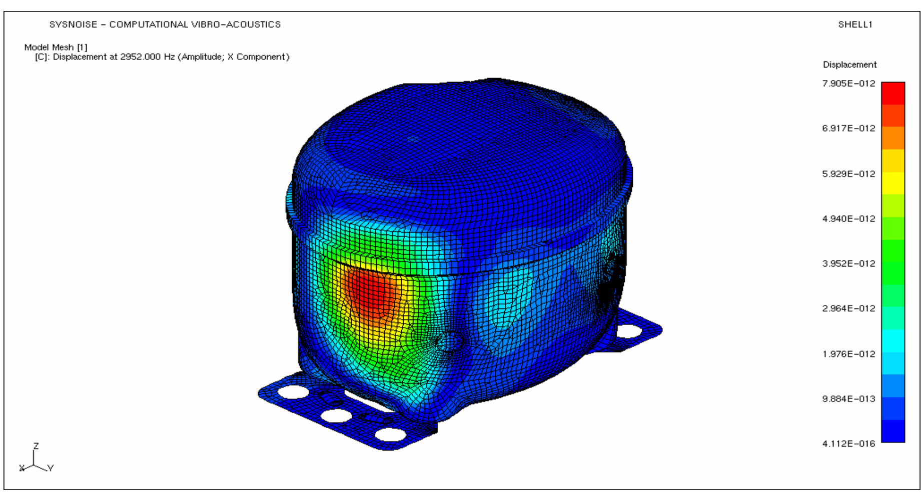

Figure 5: Forced cavity's pressure distribution at 2950 Hz

The accelerations were applied on the cylinder head's external surface, and were set in every node equal to the maximum experimentally-measured value but only in the 3150 Hz third octave band. This way the displacements on the external gas' surface (the one in contact with the compressor's shell) were determined.
This process was repeated twice, one for the compressor's original configuration, and one for the modified assembly, with the pump group 6 mm lower (corresponding to a situation with 6 mm lower suspension springs).

3.3 Vibro-acoustics analysis

A numerical modal analysis of the compressor's shell was run and successfully compared with experimental data. This model was imported in Lms Sysnoise, and a B.E.M. model (Fig. 6) with the shell forced with the overdescribed displacement was created and solved , both for the normal configuration and the modified (+6 mm) one.

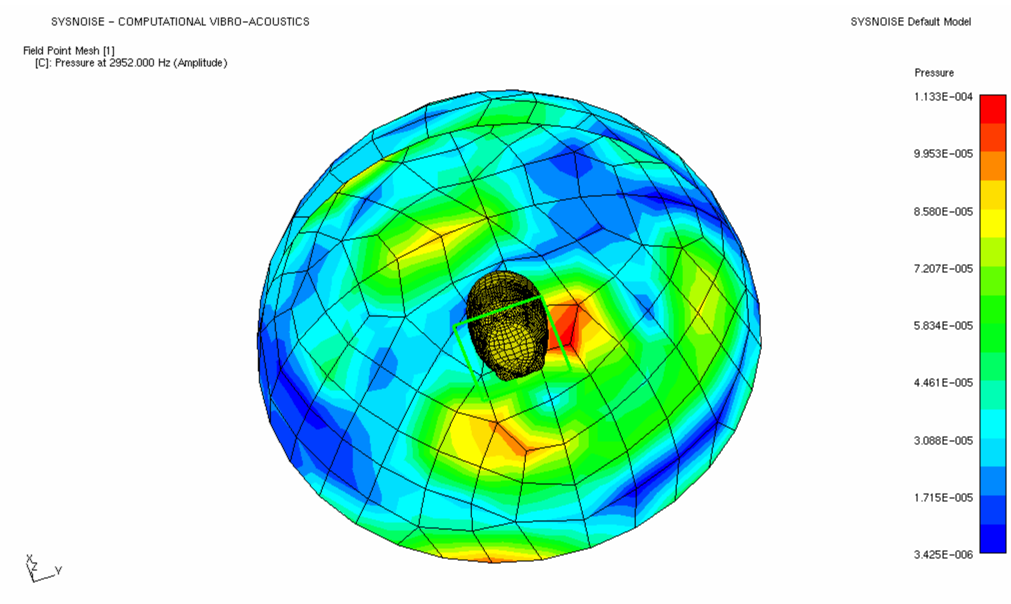

Figure 6: External pressure distribution from the B.E.M. calculation at 2950 Hz

From the B.E.M. model the pressure distribution on a 1-meter-ray hemisphere was obtained, and then the sound power level for the 3150 Hz third octave band was calculated.
The results showed (Tab.1) a 3dBA sound power level reduction passing from the original configuration to the modified one.

	Original Config.	Modified (-6mm) Config.
LWA at 3150 Hz [dBA]	35	32

Tab.1: Numerical results comparison

Validation of this result has been obtained directly measuring noise emitted from the compressors assembled in two different hermetic shells which reproduce the numerical configuration.
Sound power level at 3150 Hz third octave band emitted from the compressor assembled in the modified shell was 2.5 dBA lower than in the standard shell.

4. CONCLUSIONS

An initially-undetermined acoustic problem was solved combining experimental measurements and numerical simulations.

First acceleration measurements on the compressor pump and a upper shell's modal analysis gave evidence of the conditioning factors, then a numerical forced analysis of the internal cavity and a BEM calculation solved for the initial configuration and for a modified one, provided a good design solution with noise emission reduction.
This procedure gave also good indication on other factors, as internal distances and important points to-be-measured, affecting the compressor's noise emission.

5. REFERENCES

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6. ACKNOWLEDGEMENTS

Autors fully acknowledge Mr. Massimo Muzzin for his precious help.