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SHELL OPTIMIZATION THROUGH VIBRO-ACOUSTIC ANALYSIS

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ABSTRACT

The paper describes a model that can predict the noise emission of a compressor, taking into account both structure borne and air borne noise. Such a kind of model can be used to design shell and to uncouple sound sources and structural and cavity modes. The model was tested and validated on a production compressor. Each subsystem of the model has been validated by means of experimental modal analysis (shell and cavity). Finally the whole system was validated through measurement of the emitted sound power.

NOMENCLATURE

p: Acoustic pressure
k: wave number

INTRODUCTION

Using BEM analysis techniques it is now possible to predict noise emission at the first stage of development of a new compressor and to try to uncouple the modes of the shell, the cavity and the muffler.

The developed method has been applied to reduce noise emission from new shells. The knowledge of sources can be reached by experimental testing. As first characterization of the compressor, two sources were considered and measured: the frequency dependent structural loads and the pressure at the muffler's outlet as in first approximation, these factors do not depend on the considered shell. Then the frequency response of the system due to structural loads and pressure was calculated and the shell was redesigned in order to uncouple resonances from sources. Measurements were also performed to validate the noise emissions' calculation. The compressor considered and used for the tests has a refrigerant capacity of 78 Kcal/h in R600a fluid.

NOISE SOURCES

In a compressor there are three fundamental sources:

- Structural borne: due to impacts, unbalance, magnetic field, friction forces;
- Air borne: due to suction line;
- Fluid borne: due to gas pulsation at outlet pipe.

The first two sources are more important for uncoupling the shell’s resonances and have been characterized through direct measurement of frequency loads:

- Load on supporting spring;
- Load pressure at muffler’s outlet.

Structural borne load is transferred to the shell mainly through supporting springs: this was characterized measuring directly these forces.

The most important air borne load is due to the suction line: this was characterized measuring the pressure at the outlet of the suction muffler.

An experimental modal analysis on existing shell gave an estimation of the material’s modal damping factor; the used values were later confirmed by an experimental modal analysis performed on the redesigned shell.

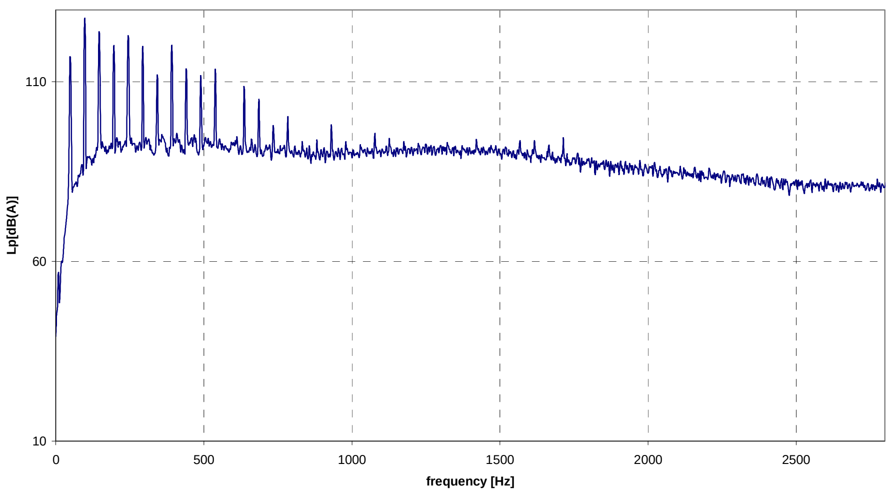

Fig. 1 Acoustic pressure level at muffler’s outlet

NUMERICAL ANALYSIS

STRUCTURAL MODAL ANALYSIS

The first analysis performed is a modal analysis of the shell, taking into account the effective shell thickness due to forming process and the presence of lubrication oil, which has been considered as a distributed mass on the shell’s bottom. The presence of compressor pump has been considered through a lumped mass connected to the supporting system through springs and discharge pipe, which were modeled as beam elements.

Numerical modes were validated through experimental modal analysis.

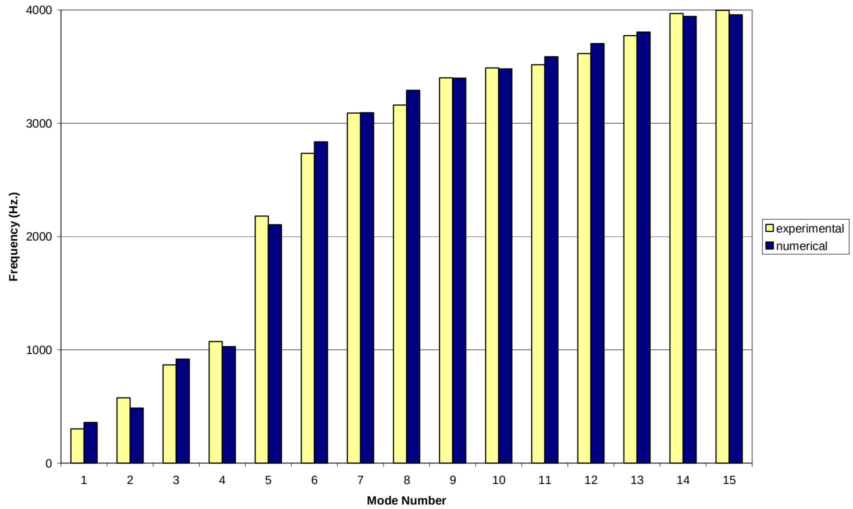

Fig. 2 Structural modes

FREQUENCY RESPONSE FUNCTION OF THE SHELL

The second step to evaluate structural noise emission was a numerical evaluation of the transfer functions of the shell respect to structural loads: unitary forces were applied in x,y,z directions to the shell's structural model.
They were applied at five points corresponding to the four welded points were the springs are in contact with the shell and to the connection between the discharge tube and the shell.
A constant damping factor was used for the sound power calculation.
From these forces the frequency dependent displacements of the shell's point were calculated; they were later used in a BEM model for the calculation of the sound power emitted by the compressor due just to the structure.

CAVITY RESPONSE ANALYSIS

A FE model of the cavity was used, in which the measured pressure at the outlet of the suction muffler was applied.
The first step to evaluate the cavity's response was a modal analysis of the cavity solving Helmolz equation:

$$\nabla^2 p + k^2 p = 0$$

through a FE mesh with tetrahedral elements. Up to 4 kHz octave band there are about 130 modes for the optimized shell. Then with modal superposition analysis the response was calculated from 49.2 Hz to 4684 Hz with a 49.2 Hz integration step.
This value corresponds to the rotating frequency of the compressor's crankshaft.
The frequency dependent pressure value, calculated at the cavity's points in contact with the shell, were later used as boundary conditions in another FE model for the calculation of the shell's nodes displacement due just to air borne sources.

The displacements were then applied, as for the previous structural model, in another BEM model for the determination of the sound power emitted by the compressor due just to the air borne phenomena.

ACOUSTIC RADIATION ANALYSIS

The experimental setup used for the measurements in the semi-anechoic chamber was reproduced in a BEM model, considering a symmetry plane on which the compressor is supported to simulate the floor, and a 1 meter radius semisphere to simulate the microphones array.
Both BE models, the structural and the air borne ones, were solved separately to obtain the two contributions to the sound power emitted.

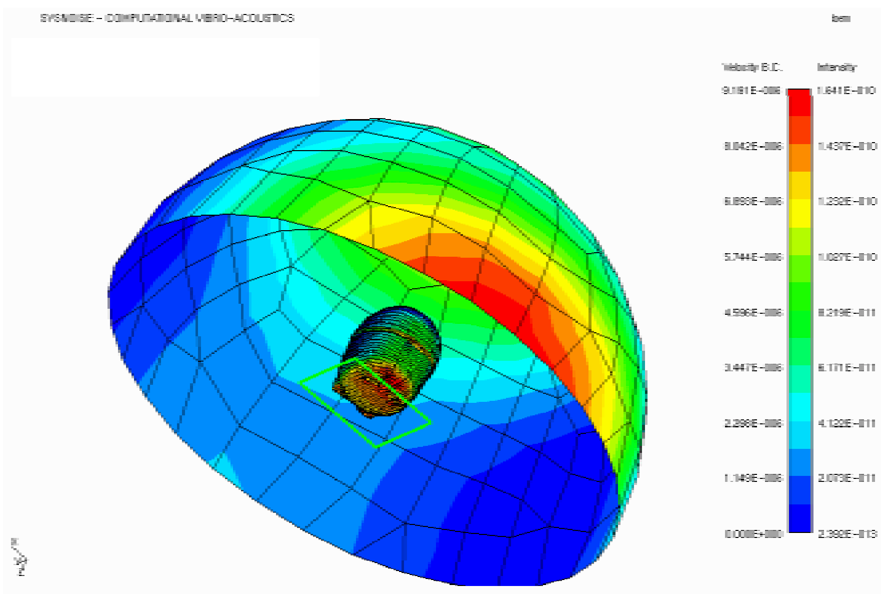

Fig. 3 Acoustic radiation

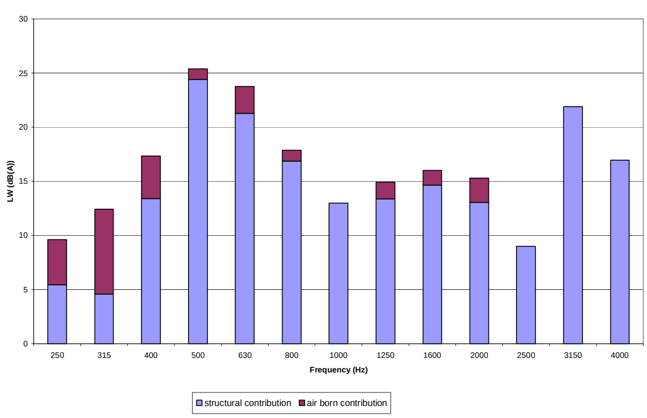

Fig. 4 Structural and air borne noise

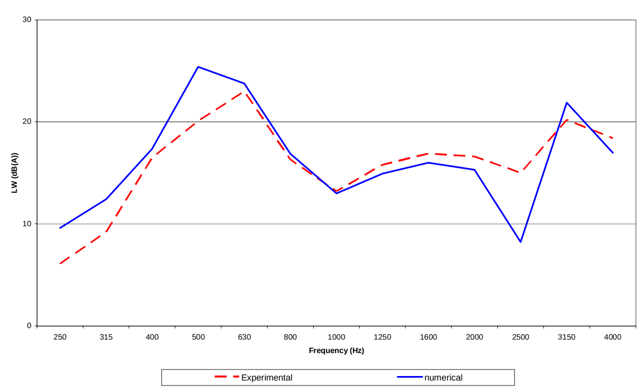

Fig. 5 Numerical and experimental results

Fig. 6 Old and new shell acoustic power emission

CONCLUSIONS

A technique for the optimization of a hermetic compressor's shell based on FE and BE methods has been developed.
Contributions to sound power due to air borne and structural phenomena have been calculated separately and then summed.
A good correlation with experimental data was found. Main differences are found in the 500Hz third octave band.
The reasons of this mismatch are not yet completely known, so further investigations are necessary.
Using this technique a reduction of the emitted sound power was obtained, as you can see in figure 6.

ACKNOWLEDGEMENTS

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