Purdue University
Purdue e-Pubs

International Compressor Engineering Conference | School of Mechanical Engineering

1998

Numerical Prediction of the Radiated Noise of Hermetic Compressors Under the Simultaneous Presence of Different Noise Sources

E. Biscaldi
Americold - Electrolux Compressors

A. Faraon
Z. E. M. - Electrolux Compressors

S. Sarti
Engin Soft Trading

NUMERICAL PREDICTION OF THE RADIATED NOISE OF HERMETIC COMPRESSORS UNDER THE SIMULTANEOUS PRESENCE OF DIFFERENT NOISE SOURCES

E. Biscaldi (1) A. Faraon (2) S. Sarti (3)

(1) Americold – Electrolux Compressors / R&D, 2nd Avenue N.W. Cullman, AL 35058, USA, Ph (205) 736-1345, FAX 739-0217
(2) Z.E.M. – Electrolux Compressors / R&D, Via Consorziale 13, Loc. Comina 33170 (PN) Italy, Ph 011-39-434-393489, FAX 393313
(3) Engin Soft Trading – Via 5°Alpini 6/a, 24124 Bergamo, Italy, Ph 011-39-35-362988, FAX 362970

ABSTRACT

The prediction of the acoustic radiation of the compressor shell is resulting from a numerical vibro-acoustic model, involving the description of the compressor pump structure, compressor gas cavity and compressor shell structure, through the use of the FEM and BEM techniques. The presence of refrigerant and lubricating oil in the compressor cavity were taken into account, as well as the presence of external mounting grommets, as stiffness and damping, was considered.
The model response was predicted under the simultaneous presence of different noise sources. Structural path: Unbalance forces and forces generated by compression cycle. Gas cavity path: Suction noise and cylinder head radiation.
Noise sources were previously experimentally determined and then introduced in the model, so that phase relationships between different sources were taken into account in the prediction of the radiated noise. Due to the interaction between compressor body → cavity → shell structure, the model response was studied, considering a coupling between cavity and structure.
Finally, the vibration velocity field on the shell, was the boundary condition in a BEM model for the study of the external radiation. Pressure fields in free field condition, due to the simultaneous presence of the internal noise sources, as well as the contribution of each noise source, were the output of the model.
Correlations of the model with measurements is quite good, especially for the frequency range controlled by structural noise sources. A model refinement was necessary, to take into account the structural damping physically present in the compressor. In this case, a modal damping factor was introduced based on experimental determination.
Considering the frequency range of the radiated sound pressure, controlled by the gas cavity noise sources, the correlation with experimental measurements is not as good as expected. Generally, in this case, the prediction is higher than measurements, and was not feasible to find a model for the gas cavity damping factor as a function of frequency. Further investigations in this direction are still in course.
The model was used for the prediction of the radiated noise of a new shell shape design, as well as of new compressor parts, having impact on the noise source modification. Correlation with the available data is shown in the paper.

INTRODUCTION

In the design phase of a compressor, many design criteria such as the expected noise levels must be met before the desired product can be cleared for production . In a modern design environment intensive use is made of numerical simulation tools to assist in this task. With the state of the art in acoustical simulation tools, is nowadays possible to study the acoustical field inside a complex enclosure as well as sound radiation problems with vibro-acoustic interaction. For accurate modeling, input data from experimental tests can be incorporated with structural and acoustical finite element calculations.

Due to the complexity in describing in exhausted form the noise radiated from an hermetic compressor in normal operation, the main target of this work was to predict the noise behavior of a current compressor, in free field condition, having enough experimental data available to verify the model consistency. Based on experimental investigations, the following noise sources were considered mainly contributing to the noise emission:

Exciting forces from compressor unbalance
Exciting forces from compression cycle
Suction noise

The assumptions above are acceptable however, considering that the final usage of this numerical model is to predict noise improvements based on a comparison method between existing and new solutions.

METHODOLOGICAL APPROACH AND RESULTS

Numerical Simulation Techniques

In low and medium frequency range, techniques for numerical simulation of vibro-acoustic problems are based on the discrete solution of the Helmholtz equation $\nabla^2 p + k^2 p = 0$ (were k is the wave number) (1).

F.E.M. techniques are used for the discrete set of the domain under investigation, B.E.M. techniques for the discrete set of the boundary of the domain under investigation. B.E.M. technique is suited to solve both interior and infinite domain problems, a typical application being the radiated noise study. Facing the problem of compressor vibro-acoustic simulation it is furthermore necessary to consider if there is a mutual interaction between structural vibrations and sound waves. In that case it becomes necessary to solve vibro-acoustic coupled problems. In the case of study the following procedure was adopted:

- Coupled vibro-acoustic response analysis of compressor body and compressor cavity, to describe the compressor shell vibration-induced field
- Uncoupled acoustic response analysis on the compressor exterior domain, under the effect of shell structural vibrations

A more detailed procedure is summarized in the following table.

	ANALYSIS	RESULT
1	Compressor structural modal analysis (F.E.M. – Structural)	Compressor resonance frequencies and mode shapes
2	Compressor cavity – acoustic modal analysis (F.E.M. – Acoustic)	Compressor cavity resonance frequencies and mode shapes
3	Compressor shell – coupled frequency response analysis , under exciting forces (F.E.M. – Structural + F.E.M. – Acoustic)	Vibration velocity on compressor shell surface. Frequency range 50-3500Hz
4	Compressor acoustic radiation analysis (B.E.M. – Acoustic)	Compressor Sound Power Level, Sound Pressure Level, for each exciting source, at each harmonic order in the frequency range 50-3500Hz

Structural Modal Analysis

A compressor shell model with shell elements was prepared for the analysis, taking into consideration a variable thickness distribution due to the stamping process. Although dynamically the compressor pump can be considered a rigid body with its own inertia properties, it constitutes also a boundary for the compressor cavity. For that reason, a specific model with shell elements was adopted, with element thickness adjusted to reproduce the inertia characteristics of the compressor pump. Suspension systems and elastic connections were taken in consideration. Compressor discharge tube was described with beam elements, compressor springs were modelled as mono-dimensional elastic elements and the external grommets were modelled distributing the stiffness around the external bracket hole. Finally the presence of the lubricating oil was considered also either as distributed mass on the lower shell, or part of the compressor cavity for the acoustical study of the cavity. The global model consists of 12000 elements and 12000 nodes

In order to verify the impact of each above described variable, F.E.M. modal analysis with different configurations were then carried out. Finally Experimental Modal Analysis on the whole compressor was necessary for model validation. The following table shows results of the F.E.M. modal analysis and experimental modal analysis for the most critical compressor frequency ranges.

1/3 octave	Shell constant thickness		Shell variable thickness		Compressor w/o oil		Compressor w/oil		Experimental modal analysis	
	Mode	Freq.	Mode	Freq.	Mode	Freq.	Mode	Freq.	Mode	Freq.
400					20	377	20	376		
					21	403	21	397		
					22	420	22	417		
					23	437	23	432		
500					24	532	24	532		
					25	555	25	555		
630					26	637	26	637		656
					27	679	27	679		
2000	5	2169	5	2083	40	2121	40	1862	14	1879
	6	2205	6	2115	41	2161	41	2116	18	2085
					42	2237	42	2149	19	2188
							43	2237	20	2208
2500	7	2350	7	2251	43	2265	44	2278	21	2252
	8	2493	8	2398	44	2301	45	2301	22	2321
	9	2620	9	2474	45	2405	46	2400	23	2371
	10	2807	10	2626	46	2523	47	2487	24	2446
			11	2744	47	2705	48	2556	25	2530
					48	2739	49	2705	27	2689
					49	2789	50	2739	28	2742
					50	2803	51	2784		
							52	2803	29	2816

Looking at the table, it becomes evident how the variable thickness and the presence of the oil effect the compressor structural behavior, introducing a shift in frequency and increasing the modal density.

Compressor Cavity - Numerical Modal Analysis

The next step of the procedure is to provide the acoustic modal base of the compressor cavity. An F.E.M. model with tetrahedron solid elements was provided, considering the cavity partition due to the presence of oil and refrigerant. The cavity mesh on the boundary was built in a way that, nodes of the cavity match nodes of the shell structure for the next coupled analysis. The model consists of about 18000 nodes and 70000 elements.

Results of this analysis are the pressure standing waves distribution inside the cavity and the related eigenvalues. This information gives indications that those frequencies can be excited by compressor noise sources, particularly those that are close to the structural eigenvalues. Up to 200 cavity modes were found from 0Hz to 4000Hz

Experimental Noise Sources Determination

One important part of this study consists of the experimental determination of the main excitations, caused by electric motor and pressure cycle, that through the structural path or the airborne path generate the compressor shell vibrations

Forces released from the spring, on the compressor bottom support system, as well as forces released from the discharge tube on the shell, were considered from the structural excitation point of view.

In order to be able to measure forces coming from the compressor pump only, the assumption that the excitations are discharged on a body of infinite stiffness was done .

The airborne noise sources, exciting the cavity and the shell structure, were considered mainly due to the suction inlet pressure fluctuation and the radiated noise from the compressor cylinder head.

The force measurements on the inner bottom supports were carried out using a special bolted shell (Figure 1) equipped with four three-axial force transducers. The base plate, where the bottom pins were located, was designed such that the intrinsic stiffness ensured the hypothesis of "non compliance" of the structure .

Even for the excitations transmitted by the discharge tube to the shell, a three-axial force transducer was used.

The transducer was rigidly connected to the discharge tube by means of a complex mechanical apparatus able to ensure good accuracy of the force measurement.

The major problem of this measurement was to ensure the non compliance of the structure, avoiding to underestimate force amplitude values due to the bolted shell structural modes. To solve this problem, an accurate dynamic calibration of the force transducer was carried out and a correction curve was then applied to the measured data.

The measurement of the excitation due to the suction process was realized with a probe microphone introduced in the inlet duct of the muffler. The noise radiated from the cylinder head, measured trough accelerations and acoustic measurements, was found to be a very low contribution to the compressor noise and was not considered in the model. For these kinds of tests a special bolted shell with no change in the shape of the cavity was used.

In all the measurements the phase relationship of each component was considered.

The frequency analysis of forces and pressures was carried out on the range of validity of the numerical model (both structural and cavity) and until the non compliance effect for the forces on the discharge tube was satisfied.

All the data were post-process to provide data to the numerical simulation for the harmonic analysis. The data correction to take account of windowing effect was also considered.

The experimental results were necessary to understand the dynamic behavior of the compressor. Tests on the discharge tube and suspension supports showed critical components in the frequency range where that compressor is more critical. The spectrum of the sound pressure level at the muffler inlet duct showed a good behavior of the muffler in the whole frequency range (Figure 2)

For a better accuracy of the numerical model a dynamic characterization of the compressor's external grommets was necessary. A specific test bench was realized to measure the dynamic stiffness of the grommets, both axial and transverse.

It is important to underline that the damping is not viscous, but seems rather a structural damping, according with the model $K = \dfrac{f}{x} = (k - \omega^2 m) + i(h)$ where K is the dynamic stiffness, $h = \eta\omega_n^2 m$, with η structural damping loss factor (2)

Cavity - Structure Coupled Analysis and Acoustic Radiation Analysis

Following the structure and cavity modal base extraction described in the above paragraph, the next step was to study the frequency response function of the coupled system cavity → structure, under the excitation of the sources described above. This study produces two sets of nodal results: acoustic quantities, pressure and velocity for the fluids into the cavity, and displacements on the shell structure. The last one will be used as boundary conditions for the radiated noise outside the compressor.
As the dimension of the model is particularly heavy becomes convenient to calculate the harmonic response of the system through the modal superimposition technique, using the modal bases of structure and cavity previous calculated.

For each exciting condition and component at each frequency, the resulting vibration field on the compressor shell becomes a boundary condition for a B.E.M. model, used to study the noise radiation on the assumption of free field condition. An Indirect Boundary Element Method approach with a solution based on a variational formulation of the Green's function was used. Sound Pressure Levels outside the compressor shell as well as Sound Power Levels are the results of the analysis.
From the results of the radiated sound power by each single source (Figure 3), it is possible to underline that power levels due to suction noise are too high, especially in the frequency range from 2400Hz to 3500Hz. One of the reasons can be attributable to the absence of damping in the cavity. Further investigations are in process. Moreover, numerical noise spectra show that forces transmitted from the compressor inner pins are responsible for more of the noise radiated in the low frequency range, while forces transmitted from the discharge tube are responsible for more of the high frequency range.
As far as the global contribution of all the noise sources to the radiated sound power (Figure 4), the correlation between the numerical model and experimental measurements is quite good until 2400Hz, above which frequency the overall level is strongly affected by the suction noise source.

APPLICATION TO A PRATICAL PROBLEM

New Shell Design

The results of the experimental test and the noise radiation analysis allowed the design of a new shell shape. The guidelines for the shell design were based on the change of the shell stiffness in the areas of the suspension supports and the discharge tube joint, which the numerical model showed to be critical surfaces. The design validation was based on the reduced acoustic power emitted as predicted by the numerical model.
A comparison of the radiated sound power from the old and new shell design is shown in figure 5. Final test on the compressor as reported in figure 6, shows a noise reduction in the range of 2500Hz with an overall reduction of 2 dB, despite the shell thickness reduction of 0.5mm.

CONCLUSIONS

A detailed description of the methodological approach used in the model was done. Although there remains some open items related to the noise radiated by the noise sources exciting the cavity, it was demonstrated that the numerical prediction of the compressor radiated noise is quite close to what was experimentally measured for the compressor under investigation. The existence of a detailed calculation procedure makes the model suitable to be used as a development tool. Further works mainly focused on the possibility of finding a suitable model for the gas cavity damping factor are in progress.

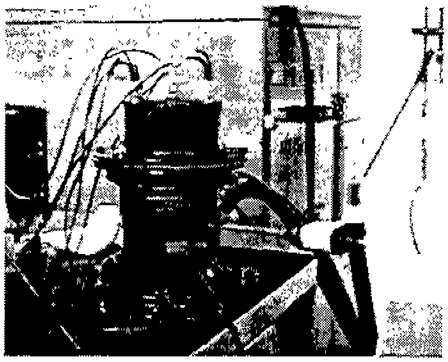

Figure-1. Compressor Bolted Shell

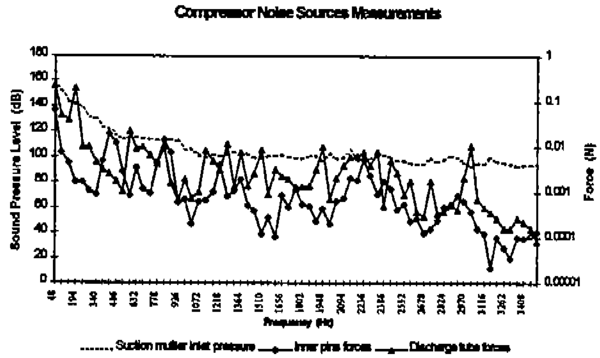

Figure-2. Compressor Noise Sources Measurements

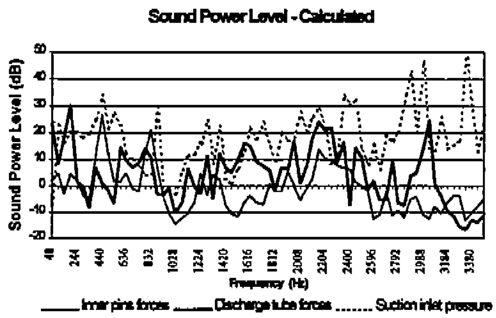

Figure-3. Calculated Sound Power – Sources Contribution

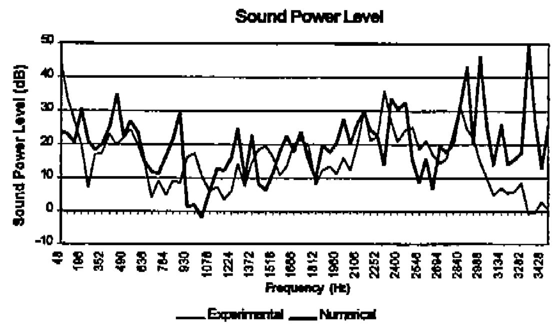

Figure-4. Numerical vs Experimental Lw Correlation

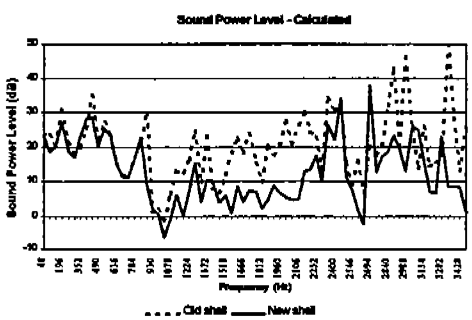

Figure-5. Calculated Sound Power – Old shell vs New Shell

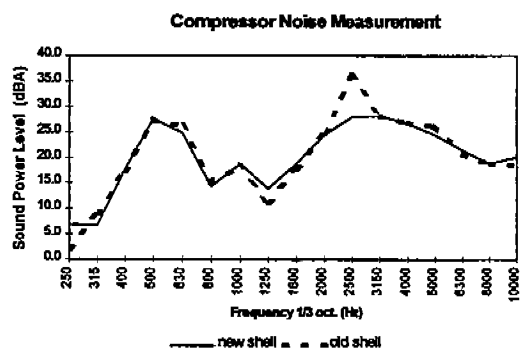

Figure-6. Sound Power Measurements – Old shell vs New Shell

REFERENCES

1) P. Sas, "Advanced Techniques in Applied and Numerical Acoustics" Proceedings ISAAC5, KU Leuven, 1994, Belgium
2) D. J. Ewins, "Modal Testing: Theory and Practice" Research Studies Press 1985
3) SYSNOISE "Theoretical Manual version 5.3A" LMS Numerical Technologies N.V. Leuven, Belgium