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ABSTRACT

A significant part of the noise radiated by a hermetic compressor uses the gaseous cavity and the compressor shell as a transmission path.

Encouraging results have been obtained in the compressor insulation putting together in the shell design some concepts coming from the theoretical study of the shell vibrations; as a consequence high stiffness shapes have been studied and realised to obtain the highest possible Noise Reduction.

The second basic concept used, besides this, has been the vibration de-coupling between the gaseous cavity and the structural shell modes: the natural frequencies of the shell and global modes of the gas cavity must be placed in different frequency ranges.

In this study all the parts of the shell have been considered, including the external compressor brackets whose characteristic frequencies were found to be in a critical range.

Numerical prediction tools such as FEM analysis and three dimensional acoustical modelling of the gaseous cavities were extensively used.

Through cross checking of the numerical modelling with the tested results we had the possibility to demonstrate a good level of predictivity of the used theoretical tools, these allowed an important reduction of costs and times for the product realisation to its final stage, avoiding time consuming re-iterations.

In the following presentation the techniques used and the results obtained are reported.

PRELIMINARY INVESTIGATIONS

Sound power level:

Measurements carried out, in the different working conditions of the compressor, in a reverberant room with an averaging time of 32 sec.

Fig. n. 1 shows a typical diagram that represents the average spectrum for a 9 cc compressor tested at -30°C evaporating temperature (55°C condensing . 32 °C liq. sub cooling and vapour overheating) with R 134 A as refrigerant.

Noise Reduction Index:

$$N.R. = 20 * \log \frac{P.\text{int}}{P.ext}$$

Where P.int and P.ext mean respectively internal and external Sound Pressure Levels.
The test was carried out on the old shell shape and on some other interesting available shells

The sound pressure levels were measured (see fig n. 2) inside the shell, where an acoustical mass generator was placed, and outside the shell, in near field, in perpendicular direction to the surface; for the low emission levels we were compelled to use a sound intensity probe.

The main reasons for this type of measurements were, first of all, the identification of the surfaces with lower Noise Reduction index, then the definition of a level to which it should have been possible to make a reference once got the new shell: In fig. n. 9 is reported the final comparison.

Energy flow identification:

This was carried out with the objective to evaluate the relative importance of the two possible ways of transmission of the noise: the structural and the gaseous ones.

For this reason a measurement of vibrations was carried out: accelerometers were put on the compressor block and, externally, on the shell (see fig. n. 3) while the compressor was working in standard conditions; then, through an electrodynamical shaker and a proper transmission, the compressor block was excited to have the same vibration spectrum previously measured: at this point the acceleration in different points of the shell was measured.

The comparison of acceleration spectra on the shell top (fig 4 shows the acceleration specrtum on the shell top), allowed to conclude that:

a. There was no possibility to reproduce via the external excitation the acceleration spectrum recorded during the compressor turning, therefore the difference, in frequency response should be evaluated as the contribution of the gaseous cavity.
b. In the acoustical interesting interval, the frequency range related to the thirds of octave from 1600 to 2500 Hz can be considered as mainly due to the structural way of transmission, while the range related to the third of octave from 3150 to 5000 Hz can mainly come from the gaseous side.

RESEARCH EVOLUTION

Shell Study

The development of the shell was based on the foundamental concepts of the Thin Shell Theory:

a. The coupling of the Bending stress and the "membrane" stress, only present on curve surfaces, gives to the shell itself a higher stiffness; for this reason a particular attention has been paid to reduce the use of flat surfaces.
b. For a curve surface, the natural frequencies are inversely proportional to the radius of curvature: in agreement with internal dimensions, the shape of the shell has been optimized to minimize the radius of curvature. (e.g., for a sphere we have the following relation:

$$\omega \propto \left(\frac{s}{r^2}\right)$$

Where: ω= Natural Frequency,s= Surface thickness and r= Radius of Curvature)

c. To have a symmetric section means to have infinite triads of principal axes; the section strain along the principal axis is maximum and then the resistance is minimised.
Therefore to have a non symmetric section means to reduce dramatically the probability of concurrence of the force resultant direction with one of the principal axis.
d. By introducing some changes in the section of external brackets an increasing of torsional stiffness has been obtained , with a simultaneous decreasing of vibrating surfaces

The FEM analysis predicts the behaviour of the new shell: the first natural frequencies of the shell body only are reported below in comparison with the old shell.

	OLD	NEW
1 st MODE	2690 Hz	3023 Hz
2 nd MODE	2714	3137
3 rd MODE	2794	3441
4 th MODE	3208	3640
5 th MODE	3330	4093

In the Fig. n.5 the full comparison of the two shells, including the external brackets, is reported.

Gaseous Cavity Study

This was carried out on the cavity resulting from the new shell internal room and the compressor body; an extensive use of a numerical three-dimensional acoustical code was made, able to solve, locally with the FEM technique, the Helmholtz's equation; fig. n. 6 shows a section of the cavity mesh.

As a result, natural frequencies of the gaseous cavity, at the different temperatures /pressures conditions were calculated and the "global modes" only were taken into consideration.

Note that "global modes" mean the pulsation modes related to the whole cavity volume, with variation of pressure on the whole boundary surface; therefore the natural frequencies related to local pulsation only are excluded.

The non symmetricity of the volume did not allow to create a computer model of only half cavity; therefore .with the requested mesh element dimensions, the number of elements was so high that we could only extend the frequency range to 4000 Hz which, at a first evaluation, seemed to be too limited; instead, after the first analysis, we realised that over 4 KHz global frequencies were very unlikely to exist.

Shell / Cavity Decoupling

During the study of the above two elements a special attention was paied to the decoupling of the natural frequencies of the gas and the shell structure to be sure that there was no reciprocal excitation of the two parts.

In the Fig. n 7, the comparison between the two frequency ranges is reported.

RESULTS ON PROTOTYPES

The prototypes of the shell were submitted to **modal analysis**. In fig. n.8.1 and 8.2 the comparison with the old one is reported; The reported Sum-Blocks represent the sum of the Frequency Response Functions measured in the different points of the shells.

The obtained improvement is significant : lower number of modes and higher normal frequencies.

Noise Reduction: In fig. n. 9 the already mentioned comparison is reported; this shows an improvement.

As a consequence the sound power measurements (Lw) on compressor show improvements and, as expected, on appliance as well (see fig. n. 10 and 11); the reported measurements are the average on at least 10 compressors.

It is important to outline that the comparison was done using the same compressor bodies replacing the shells only to be sure not to include the production spread in the effective evaluation.

CONCLUSIONS

The obtained results demonstrate that the increased stiffness of the shell structure, joined with non-symmetric shape and natural frequency ranges decoupled between shell and cavity give good results in terms of noise reduction: 6 dB(A) of sound power level (Lw)

Similar results have been obtained on the final appliance.

Also an important improvement was obtained in the time required for the development: the good predictivity of the numerical tecniques avoided the need for big time consuming re-iterations of sampling-and testing.

REFERENCES

W. Soedel	Vibrations of Shells and Plates	Marcel Dekker Inc.	New York 1981
S. Timoshenko	Vibration Problems in Engineering	Van Nostrand	New York 1955
O.C. Zienkiewicz	The Finite Element Method	McGraw Hill Book Company	1989
R.L. Taylor			
L.E. Kinsler	Fundamentals of Acoustics	Wiley	New York 1962
A.R. Frey			
SDRC	Finite Element Modelling and Model Solution User's Guide		

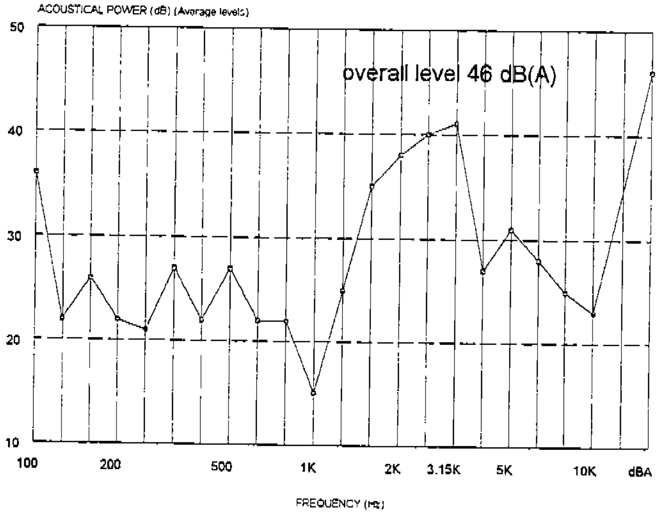

Fig. 1 Old Shell acoustical emission

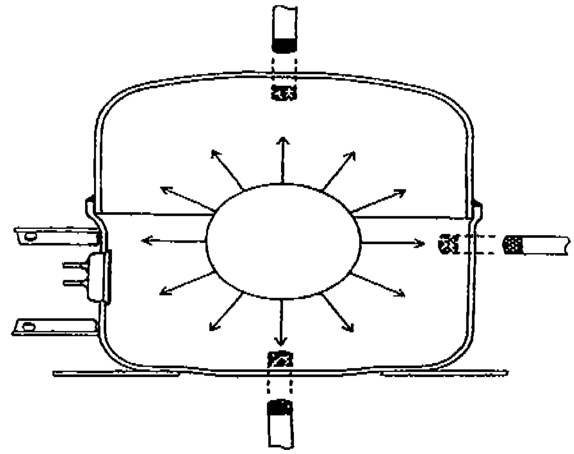

Fig. n. 2 Noise Reduction Meas. Scheme

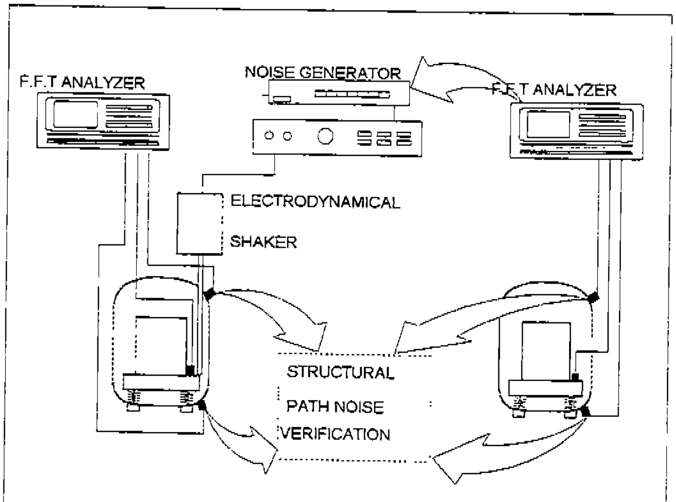

Fig. 3 Structural Path Noise Verification

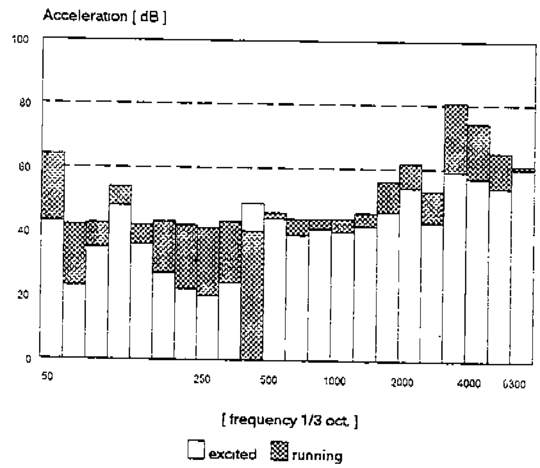

Fig. 4 run/exc acceleration comparison

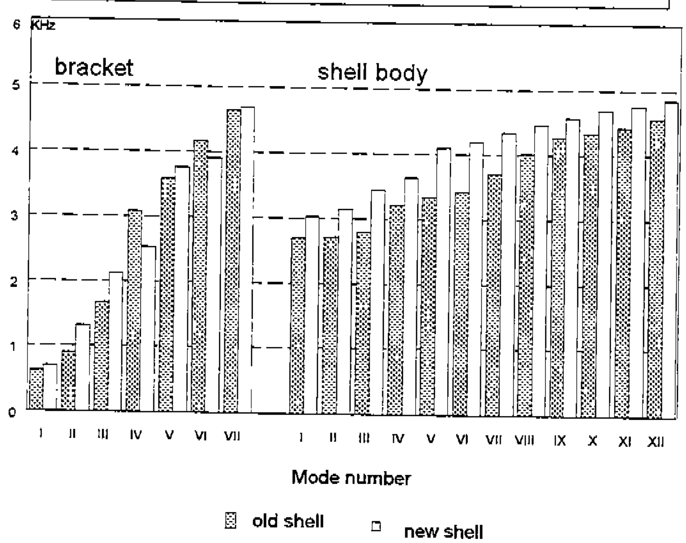

Fig. 5 F.E.M. Comparison

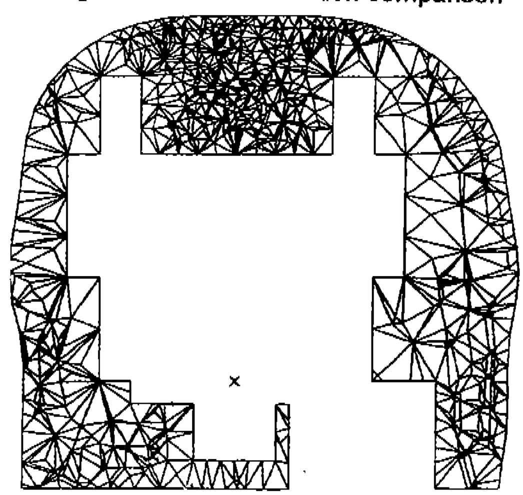

Fig N. 6 Meshed Cavity

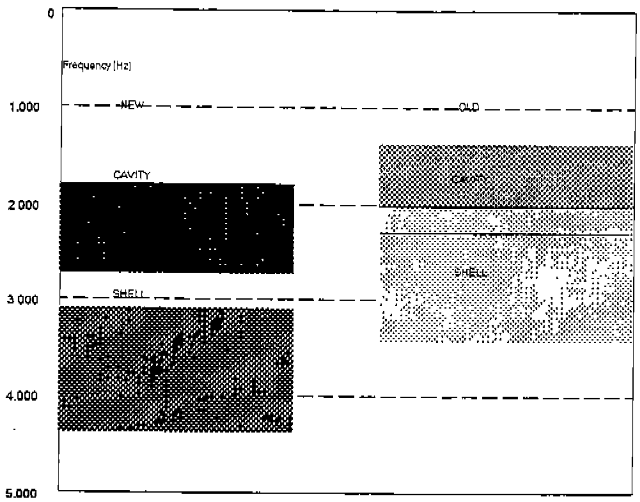

Fig. n. 7 Global Frequencies de-coupling

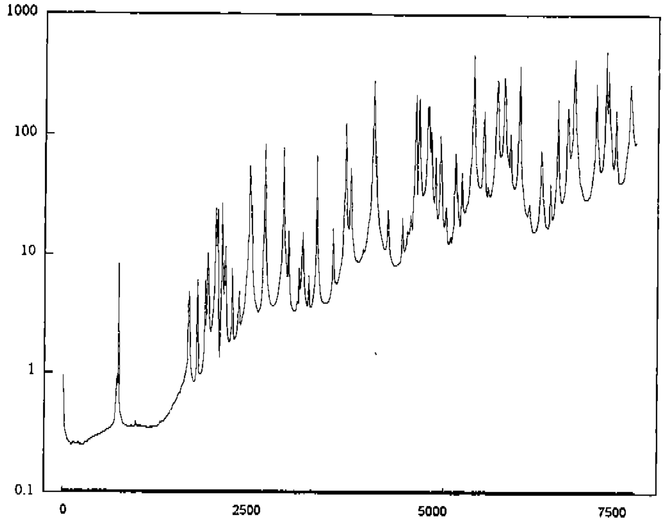

Fig. n. 8.1 Old Shell sum-Block

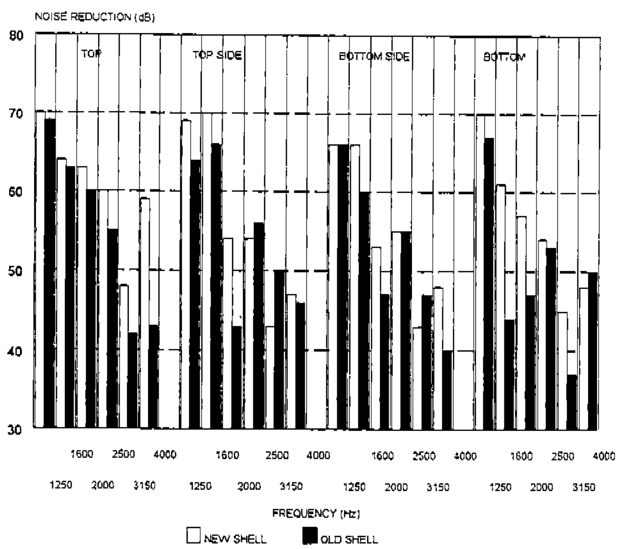

Fig. n. 9 Noise Reduction Comparison

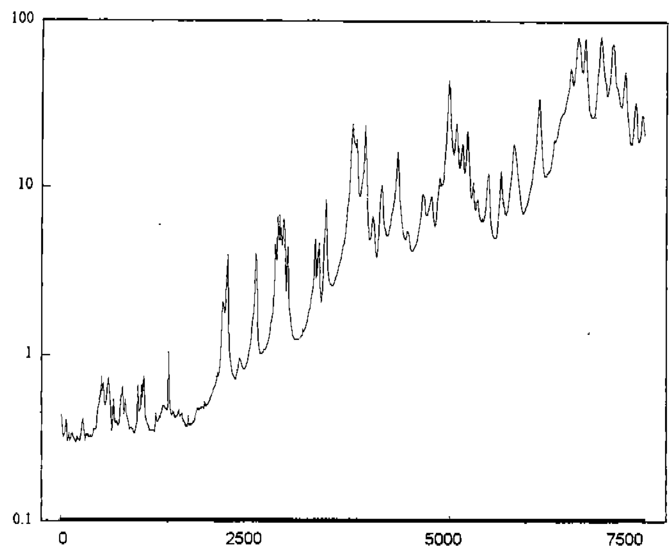

Fig. n. 8.2 New Shell sum-Block

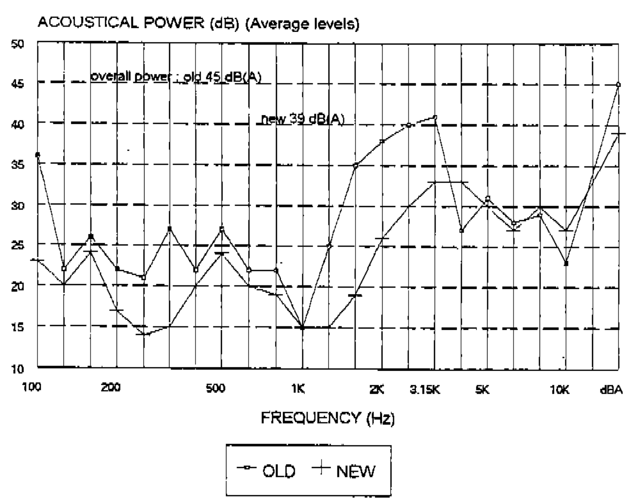

Fig. 10. Compressor Noise Comparison

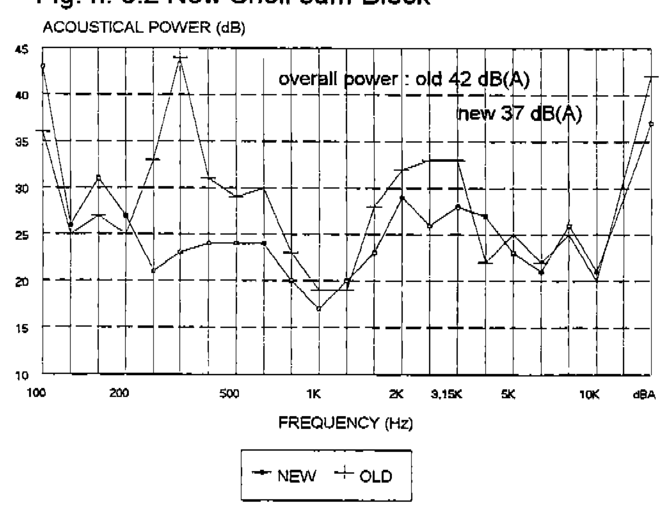

Fig n. 11 Noise Comparison on Appliance