1992

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MODAL ANALYSIS OF A COMPRESSOR SHELL AND CAVITY FOR EMITTED NOISE REDUCTION

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ABSTRACT

This work concerns the design of a domestic refrigerator compressor for noise and vibration control.
Sources of noise and vibrations and the mechanisms of their propagation to the environment are first reviewed in the introduction.
The following sections then focus on the vibrations of the gas in the cavity inside the compressor shell and on the vibrations of the shell itself.
An interpretative model of modes of vibration of the shell is also given, showing that modes can be classified in two groups: those involving side walls and those involving end-caps.

INTRODUCTION

The request for systems of refrigeration with better performance, both in terms of energy consumption, and in terms of the vibrations generated and the noise spread into the surroundings, has been acquiring more and more importance. As a result of such demanding requirements, the planning and design of compressors need put greater emphasis on noise control and this demands the use of more and more sophisticated techniques, both experimental and theoretical. Furthermore, considering the recent EEC regulations and instructions concerning the reduction of chlor-fluor-carbides in the atmosphere, a further effort in noise control is necessary since the thermodynamic cycles with the new fluids are less favourable from this point of view.

In a conventional compressor, noise is substantially caused mainly by the shell which (stimulated by the springs, by pressure fluctuations of the refrigerating fluid, by the discharge piping and by the lubricating oil) vibrates and spreads the noise into the surroundings.

Figure 1 shows a scheme of the mechanisms which generate vibrations, their mechanisms of transmission, and the parts of the compressor which, by transforming the vibrations into pressure waves, make up the sources of sound which radiate towards the immediate surroundings.

The vibrations originate substantially from the motion of the compressor body suspended elastically and from the pressure fluctuations of the refrigerating gas present in the intake and discharge.

The compressor is set in motion by the forces due to: the unbalancing of the rotating parts, the forces deriving from the masses with alternating (piston) or alternating and rotatory (connecting rod) motion, and, to a lesser degree, the impulse forces caused by the closure of the valves and by magnetic forces.

The pressure waves, which spread into the intake and discharge piping, are instead produced by the thermodynamic compression cycle and valve action. In fact the valves open and close at the same frequency as that of the rate of rotation, and since their motion is impulsive rather than harmonic, pressure fluctuations with consistent superior harmonic components are generated. Such fluctuations first excite certain acoustic modes of the cavity, and then the shell containing it.

Another important source of excitation is represented by the turbulence and by the release of eddies generated in the cavity and in the intake and discharge pipes.

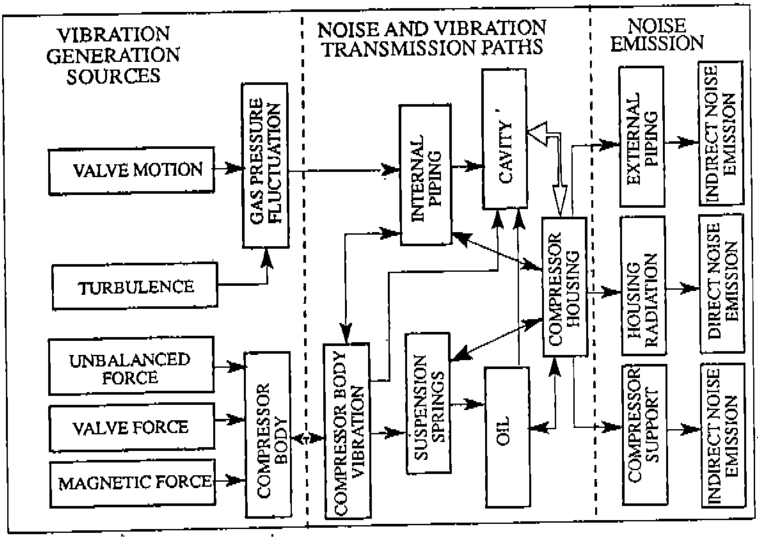

Fig. 1 - Scheme of vibration and noise generation and transmission.

Figure 2 shows a typical noise spectrum.

As can be observed from the figure and from what other authors [1,2,3] have already pointed out, three fields of frequency can be identified:

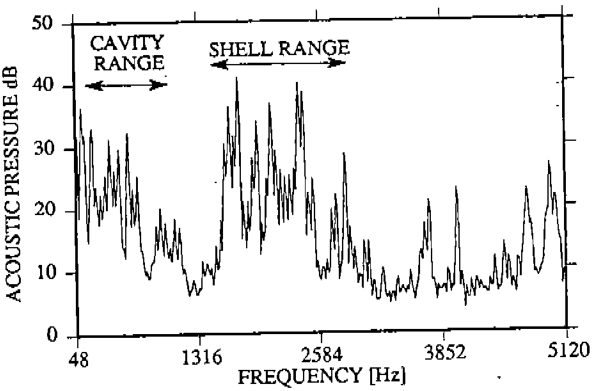

Fig. 2 - Typical noise spectrum of a conventional type of compressor.

- low frequencies (under 1000 Hz): low frequency noise is related to the rigid vibrations of the shell, caused by the fluid pulsing in the cavity in which it is contained and by the forces transmitted through the discharge pipe and the suspension springs of the compressor body;
- medium frequencies (1200-2600 Hz): This interval is dominated by the fluids (refrigerating and lubricating) contained in the cavity; this kind of excitation gives rise to flexure vibrations of the shell and they mainly come from intake gas pressure fluctuations. The unbalancing of the rotating shaft, whose basic critical speed is in this range produces stimuli which should not be neglected;
- high frequencies (above 2500 Hz): high frequency noise is caused by the vibrations originating from the impulsive action of the valves and from turbulence. Such vibrations are transmitted to the shell by the compressor suspension springs and by the discharge pipe of the re-

frigerating gas. Besides narrow band noise above 2500 Hz can be generated by magnetic-acoustic effects of the electric motor.

This work investigated the dynamic behaviour of the cavity and of the containing shell with an aim to evidencing the importance of their influence on the noise emitted. Both the cavity and the shell have been modeled with finite elements and then the relative eigenvalues and autovectors were determined. Modal analysis was performed by varying some parameters, such as cavity dimension, oil level, shell thickness and curvature radii, with an aim to pointing out their importance on the values of the natural frequencies and on the modal shapes.

MODAL ANALYSIS OF THE CAVITY

The cavity, delimited by the compressor body and by the containing shell, was modeled with 191 finite elements; the modal analysis, conducted with the ADINAT code, was performed on the assumption that the cavity contained freon at 60° C.
Figure 3 shows the first 28 natural frequencies of the cavity of a traditional compressor; the figure points out the modal density which is about 20 modes per 1000 Hz. Figure 4 schematically represents the first eight vibration modes. The broken lines indicate the positions of the nodal planes. In the figures relative to modes 5 and 6 the arrows indicate the shifting of the nodal lines starting from the bottom and going to the top of the compressor.

It can be noted that the first mode presents ventral zones (longitudinal extremities of the cavity) corresponding to the positions where the maximum pressure variations are verified. The basic frequency value depends on the distance between the points of maximum pressure (equal to about twice the wave- length) and so, it is hardly influenced by small variations in form carried out on the cavity.

In fact, there is no significant altering of the basic frequency when the cap is lowered, in such a way that the maximum distance between the compressor body and the shell is less than 10 mm. The most important effect is the decrease of the spectrum density at low frequencies (from 9 modes under 1000 Hz in the nominal shell to 5 in the case of the lowered cap). Raising the cap by 50 mm takes the spectrum density at the lower frequencies to 11 modes under 1000 Hz. Similar effects are produced by lowering or raising the oil level.

The analyses carried out have shown that the values of the natural frequencies are not substantially altered by small variations in form compatible with production requirements.
It should be noted that it is nevertheless advisable to reduce the gap between the compressor body and the shell in order to increase the modal damping values. The suppression of the undesired effect of some modes seem possible only with the use of suitable acoustic resonators [4].

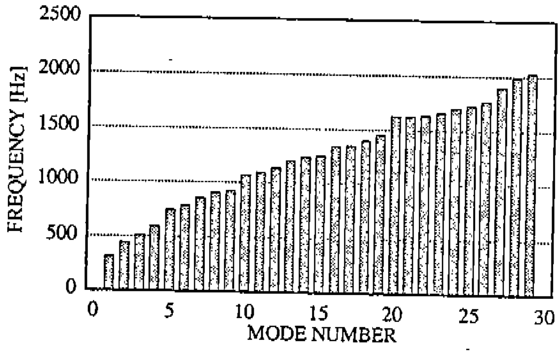

Fig. 3 - Natural frequency of the cavity of the conventional compressor.

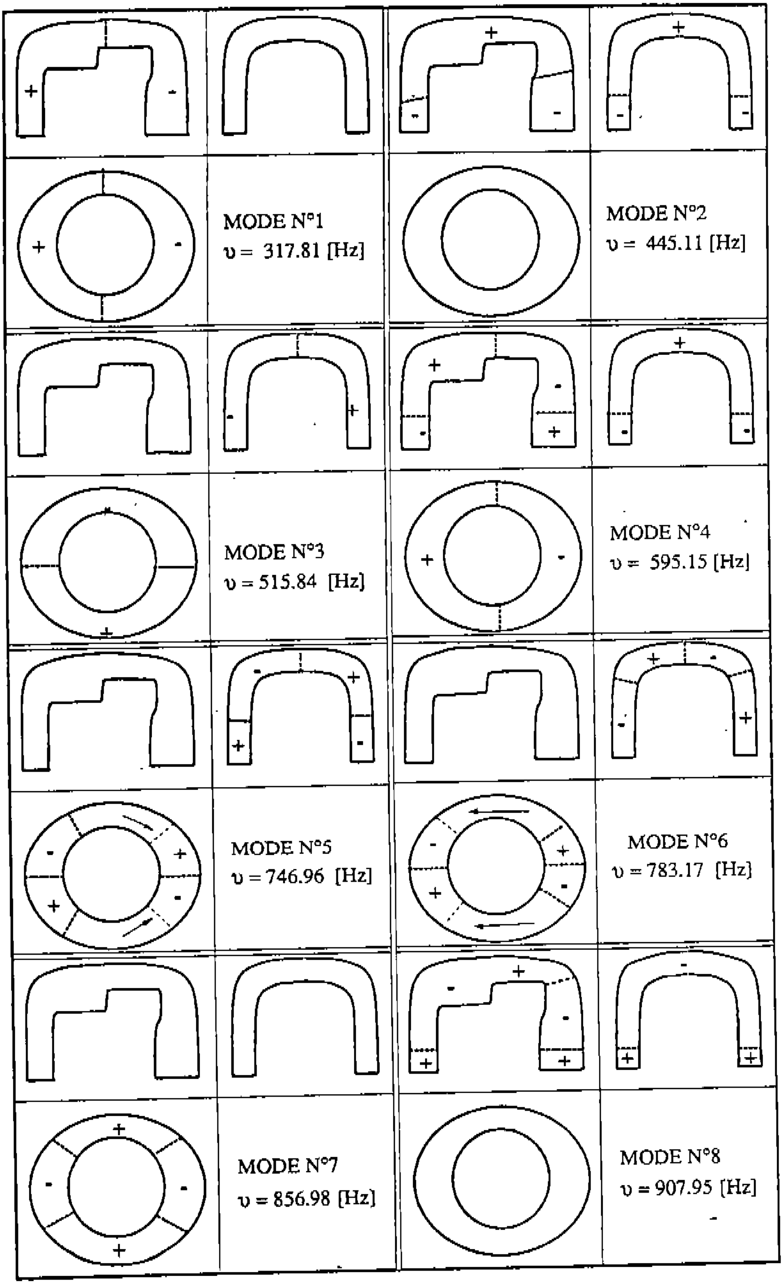

Fig. 4 - Scheme (not to scale) of the first eight vibration modes of the cavity.

CLASSIFICATION OF THE SHELL MODES

The review of the computed modal shapes of the shell (see next section), indicated that they belong to two classes (at least in the range of frequencies investigated here):
- Class 1 shows large displacements in the lateral pseudo-cylindrical portion of the housing, while the end-caps remain practically undeformed. This was called the *class of cylindrical modes* (Fig. 5).
- Class 2 shows large displacements at the domed end-plates, without involving the cylindrical body. This was called the *class of end-cap modes* (Fig.6).

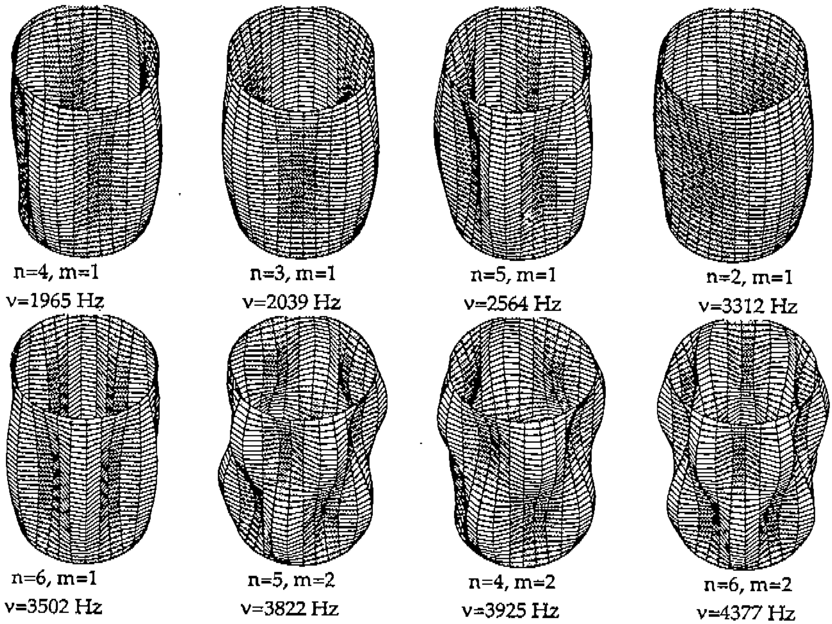

Fig.5 - Class of *cylindrical modes*.

The very existence of the two classes means that end-caps and cylindrical body are weakly coupled (coupling weakness is probably due to the high stiffness of the fillets linking the cylindrical body to the caps). The weakness of coupling allows us to explain the cylindrical modes in terms of the modes of an 'equivalent' tube (regardless of the presence of the caps) and, in their turn, the end-cap modes in terms of the modes of an 'equivalent' circular domed plate.

Closed-form solutions are available for tubes of circular cross-section, simply supported or clamped at both ends and for circular domed plates clamped at their boundary [5].

Fig.5 displays the first 16 modes of a tube simply supported at both ends. Height is equal to the height of the shell; radius is equal to the geometric mean of the two radiuses of the cylindrical part of the shell.

Because of the circularity of the cross-section of the tube, modes are in groups of two, sharing the same eigenvalue. They are characterized by two wave numbers: n, the number of half-waves in the circumferential direction, and m, the number of half-waves in the axial direction. Fig.7 shows eigenvalues as a function of n, for a given m. Frequency does not monotonically increase with n because the constraints applied at the ends of the tube (owing to its finite length) primarily affect the modes of vibration with low n. As Fig.5 shows, for $m = 1$, the minimum frequency is attained when $n = 4$; for $m = 2$ this is attained when $n = 5$.

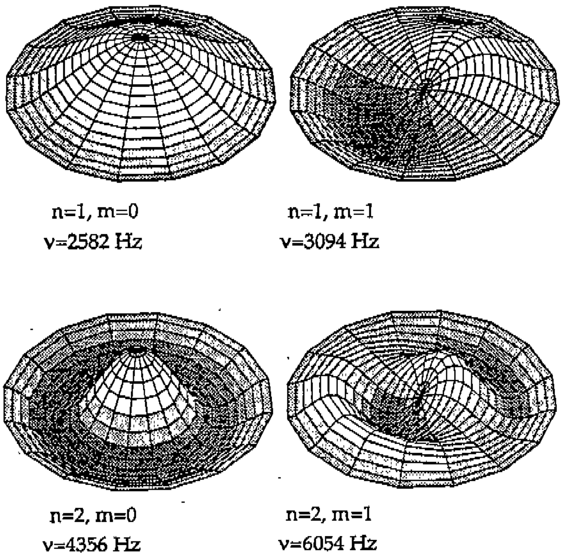

Fig. 6 - Class of *end-cap modes.*

Fig.6 shows the modes of vibration of a circular domed plate clamped at its boundary. The radius of the plate is the same as that of the tube above, and the radius of curvature of the dome is equal to the mean radius of curvature of the actual caps. As for the tube, the modes of vibration are characterized by two wave numbers (again called n and m).

The eigenvalues are also shown in Fig.7, together with the eigenvalues of the tube. Since the former increase more rapidly than the latter, we find fewer end-cap modes than cylindrical modes (up to the same frequency value).
The eigenvalues of a spherical shell with the same volume (and mean thickness) of the actual shell are also reported in Fig.7, to show the potential benefits of good shell shape design.

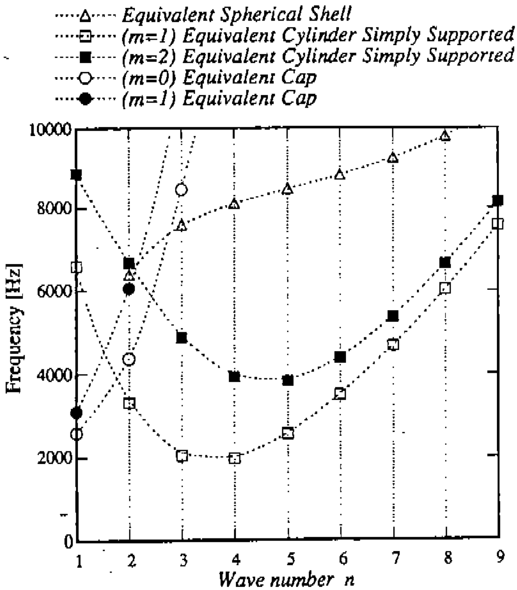

Fig. 7 - Eigenvalues of 'equivalent' spherical shell, domed end-plates and cylindrical body.

The seven figures 8A-B-C-D-E-F-G represent the natural frequency values of the first twenty modes of vibration of the shell according to different geometrical configurations.

In the figures the kind of vibration mode is also indicated with symbols: the first letter S indicates a symmetric mode, letter A an antimetric mode, the first number (*n*) shows the number of half-waves in the circumferential direction, and the second (*m*) shows the number of half-waves in the axial direction. The letters *T* and *B* indicate modes characterized by deformations at the top and at the bottom respectively.

As a reference the shell with a constant thickness of 2 mm was assumed (fig. 8A). Based on the results of the numeric analyses the following considerations can be made.

An increase of 25% in the shell thickness (fig. 8B) increases the natural frequency values of the first 20 vibration modes by an average of 10.7% and the fundamental frequency by 8.6%; a more substantial increase of 50% (from 2 to 3 mm) involves an 18.9% increase in the natural frequencies and a 14.6% increase in the fundamental frequency (fig.8C). These results agree with the values supplied by the approximate formula proposed in [6].

The stiffening weld ring, obtained by interference coupling of the cap with the lower part (10 mm x 5 mm), increases the value of the first 20 natural frequencies by 2.0% (fig. 8D) while the fundamental frequency increases by 2.6%.

A substantial increase in shell rigidity, and therefore of the natural frequency values was obtained by inserting double curvature surfaces (fig. 8E) which, for the same internal dimensions, involved a 5.5% increase in the shell overall length and a 6.6% increase in width. The natural frequencies of the first twenty vibration modes increased by 15.5% on the average and the fundamental frequency increased by 22.6%, compared to the reference (Fig. 8A) which shows simple curvature surfaces on its sides.

The double curvature must be maintained throughout the whole surface of the shell, in fact the introduction, on the bottom, of flattened areas used for facilitating the welding of the feet, caused, in the case in question, a 15% decrease of the natural frequency value.

The forming process for the shell causes variations in thickness, for example in the areas of greater shell curvature, thickness is reduced by about 20%. The natural frequency values of the shell with real thickenesses (variable from 2.5 mm to 1.8 mm) did not undergo variations of note (fig. 8F), whereas the vibration modes instead, because of the greater flexibility of the bottom, presented significant variations. For example, even the first vibration mode, type S_3-1, is already slightly coupled with the bottom.

In order to study the possibility of increasing the rigidity of the bottom, or of the cap, stiffening ribs were inserted, wherever the forming process permitted. The ribs inserted only on the cap (visible in fig. 6D) raised the natural frequency of the first mode, which only interests the cap, by about 300 Hz (fig. 8G).

Figures 9A-B-C show some typical modes of vibration.

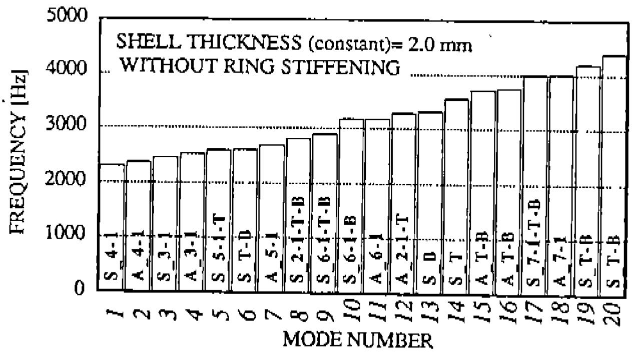

Fig. 8A - Natural frequencies and type of modes of a shell having constant shell thickness and without ring stiffening due to welding seam.

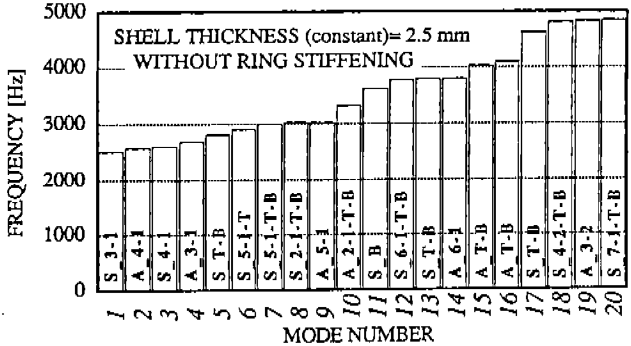

Fig. 8B - As the previous figure with shell thickness constant and equal to 2.5 mm.

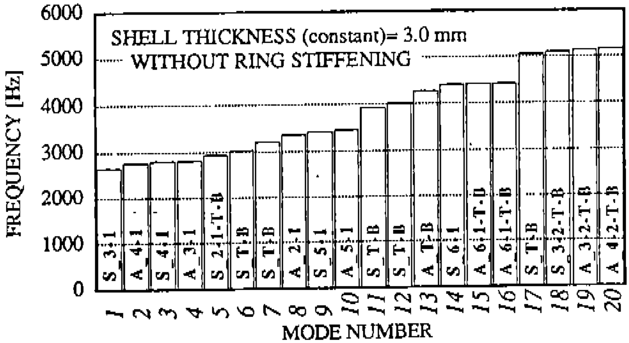

Fig. 8C - As the previous figure with shell thickness constant and equal to 3.0 mm..

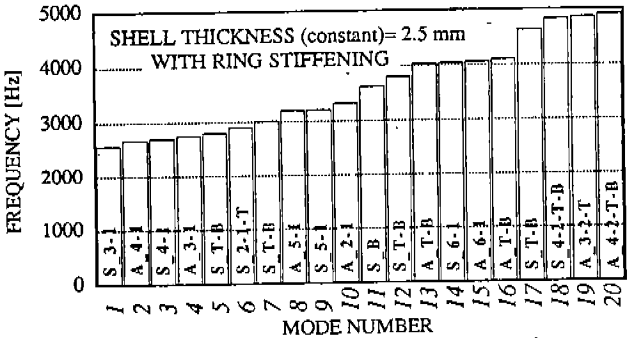

Fig. 8D - Natural frequencies and type of modes of a shell having a ring stiffening due to the welding seam (thickness=5.0 mm, width= 10 mm)

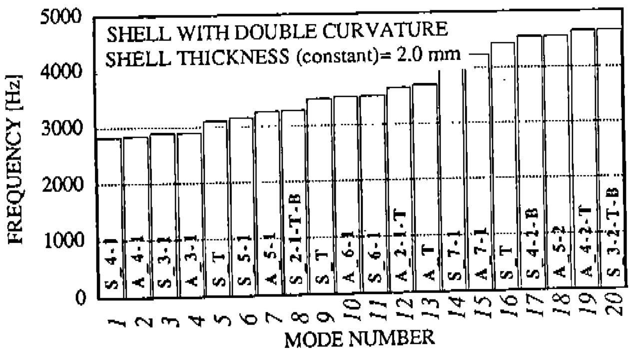

Fig. 8E - Natural frequencies and type of modes of a shell having surfaces with double curvature ranging from 30 mm to 300 mm.

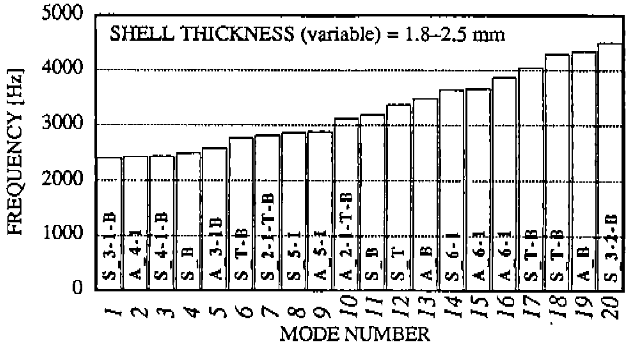

Fig. 8F - Natural frequencies and type of modes of a shell having variable shell thickness.

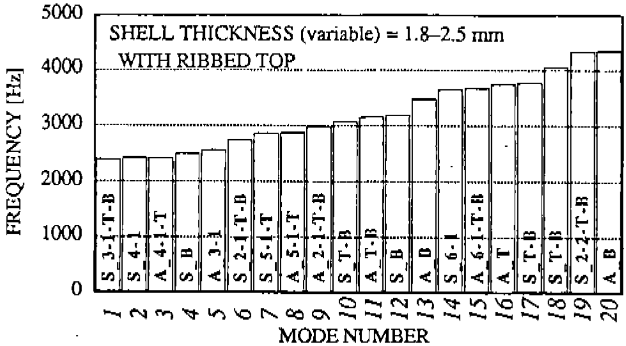

Fig. 8G Natural frequency and type of modes of a shell having a rib applied on the top.

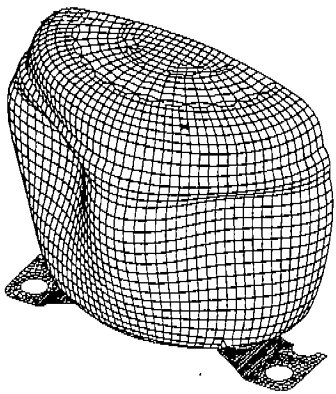

Fig. 9A - First symmetric eigenmode shape of case 8F (type of mode S_3_1-B).

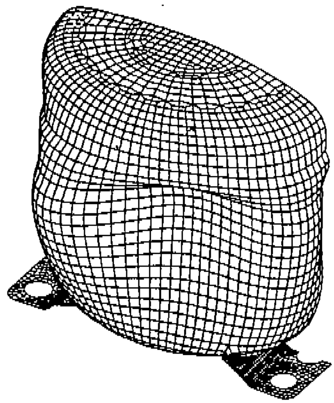

Fig. 9B - First antimetric eigenmode shape of case 8F (type of mode T_4_1).

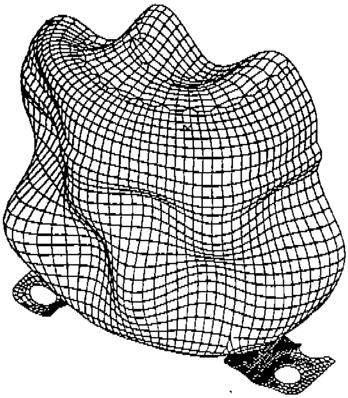

Fig. 9C - 20th symmetric eigenmode shape of case 8G (type of mode S_3-2-B).

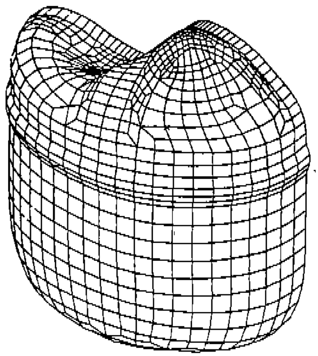

Fig. 9D - 10th symmetric eigenmode shape of case 8G (type of mode S_T-B).

CONCLUSIONS

In this work the authors have presented some results on the numerical studies performed on the cavity and shell with an aim to optimizing their acoustic behaviour.

The study on the influence of the form of the cavity on the vibration modes has shown that small variations of form do not substantially alter the values of the natural frequencies.

The study carried out on the shell has pointed out the possibilities of increasing its rigidity by means of operations that can be easily carried out and that are compatible with production requirements.

A final remark concerns the importance of the interpretative scheme for improved product quality. Knowledge of the existence of the two classes of modes and, above all, knowledge of how much natural frequencies can be increased by moving toward spherical-like shapes (Fig. 7) states an interesting and important guide-line for development.

REFERENCES

1 N. TSUJIUCHI, T. KOIZUMI: "Vibration and Noise Reduction of Household Refrigerator Using Modal Component Synthesis Technique". Proceedings of the 1990 International Purdue Compressor Conference; 1990, pp. 917-924.

2 K. TOJO, S. SAEGUSA: "Noise Reduction of Refrigerator Compressor". Proceedings of the 1980 International Purdue Compressor Conference; pp. 235-242, 1980.

3 D. C. LOWERY: "An Improved Shape for Hermetic Compressor Housing". Proceedings of the 1984 International Purdue Compressor Conference; pp. 285-290, 1984.

4. F. J. FAHY AND C. SCHOFIELD "A note on the interaction between a Helmholtz resonator and an acoustic mode of an enclosure". Journal of Sound and Vibration, n. 72, pp. 365-378, 1980.

5 W. SOEDEL, "Vibrations of Shells and Plates". Marcel Dekker Inc., New York, 1981.

6 MAU-PIN HSU W. SOEDEL "How Do Compressor Shells Vibrate and How Should They Be Designed". Report N. 0852 Ray W. Herrick Laboratories, 1986.