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CONTROL OF ACOUSTIC VIBRATIONS INSIDE REFRIGERATOR COMPRESSORS BY MEANS OF RESONATORS.
COMPARISON BETWEEN ANALYTICAL AND EXPERIMENTAL RESULTS

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ABSTRACT

This work deals with the effect of a Helmholtz resonator on the acoustic vibration inside the cavity of a domestic refrigerator compressor. A multi-mode analytical model is developed and some analyses are carried out showing the influence of resonator parameters on the acoustic behaviour of the cavity. Then the results of the tests which were performed on an industrial compressor during its actual working are discussed and compared with theoretical results.

INTRODUCTION

In most domestic refrigerators the compressor is hermetically contained inside a shell and the room between the compressor and the shell is used as a refrigerating fluid and lubricating oil tank (figure 1). The refrigerating fluid occupies the acoustic cavity which is over oil level and is stimulated by many pressure fluctuations, they are caused by the impulsive motion of the suction valve, by the motion of the refrigerating fluid which enters and leaves the cavity and by the vibration of the compressor body.

These phenomena cause the forced vibration of the fluid, the vibration amplitude becomes large when one of the acoustic modes of the cavity is stimulated in resonance condition; since pressure fluctuations have many spectral components the possibility of stimulating an acoustic mode in resonance condition is high. Large acoustic vibrations of the fluid inside the cavity have to be avoided because they increase noise emission and disturb compressor operation.

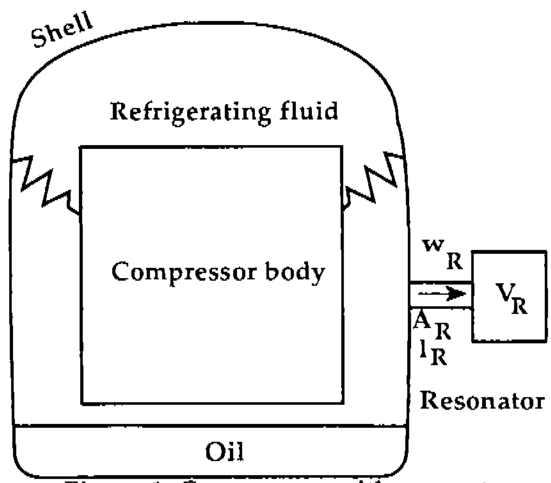

Figure 1: Compressor with resonator.

A Helmholtz resonator can reduce the amplitude of the most excited acoustic mode of a compressor cavity.
It is composed of a chamber of volume V_R which is connected to the cavity by an orifice of area A_R and effective length l_R. In a frequency range where the resonator dimensions satisfy the following conditions

$$V_R^{1/3} << \lambda, \qquad A_R^{1/2} << \lambda, \qquad l_R << \lambda, \tag{1}$$

the resonator has only one natural frequency which is $\omega_R = \sqrt{c^2 A_R / l_R V_R}$ (2)

where c and λ are sound speed and sound wavelength respectively [1].

If the resonator is properly coupled only with the i^{th} mode of an acoustic cavity, it strongly affects the behaviour of the cavity in a frequency range near ω_i, where ω_i is the frequency of the i^{th} mode. In particular the original mode disappears, two new modes arise, the first with lower frequency $\omega_{ia} < \omega_i$, the second with higher frequency $\omega_{ib} > \omega_i$, and their resonance peaks at ω_{ia} and ω_{ib} are lower than the original peak at ω_i [2]. Therefore if the i^{th} mode is the most excited mode of the compressor cavity, the Helmholtz resonator cuts down the most dangerous resonance peak. The proper coupling of the resonator with the i^{th} cavity mode is achieved only if some conditions are fulfilled [2].
First the resonator has to be tuned in order to make ω_R equal to ω_i, then the resonator/cavity volume ratio and the resonator critical damping ratio have to be properly chosen, finally resonator orifice has to be located close to an anti-node of mode i. The effect of the other cavity modes is negligible if their frequencies are far from ω_i and if resonator orifice is located close to their nodes. If these conditions are not completely satisfied the resonator affects cavity oscillations again [3], but it can be less effective and, owing to the coupling with many cavity modes, its influence on resonance peaks is difficult to foresee.

Several particular features have to be considered when a Helmholtz resonator is coupled to a compressor cavity. First the natural frequencies of the cavity are influenced by temperature and by oil level (which may not be the same in compressors of the same series), therefore it is difficult to achieve the exact tuning of the resonator to one cavity mode. Then the mode shapes of the cavity are rather complicated and it is difficult to find a position which is an anti-node of the most excited mode and a node of the other modes. Moreover constructive constraints may limit the possibility of choosing resonator location. Therefore the resonator may be coupled with many modes of the compressor cavity.

In the first part of this paper an analytical model of the system composed of a compressor cavity and a Helmholtz resonator is developed and the influence of the coupling of the resonator with several modes of the cavity and of tuning errors is analyzed. Then the experimental tests, which were performed to assess the effectiveness of the resonator are described. The last part of the paper deals with the comparison between experimental and theoretical results.

MATHEMATICAL MODEL

If conditions (1) are satisfied the resonator represented in figure 1 can be considered a lumped element device. Its dynamic equation is derived by considering the equilibrium of the gas contained into the duct (whose displacement is named w_R) under the action of external pressure p, of pressure caused by the compression of gas inside the chamber and of friction force, thus

$$\rho l_R A_R \ddot{w}_R + R_R A_R^2 \dot{w}_R + \frac{\rho c^2 A_R^2}{V_R} w_R = A_R p \tag{3}$$

where ρ is gas density and R_R is the acoustic resistance of the resonator duct excluding radiation resistance [2][4]. If both sides of equation (3) are divided by $\rho l_R A_R$, the following equation is obtained

$$\ddot{w}_R + 2\zeta_R \omega_R \dot{w}_R + \omega_R^2 w_R = p/\rho l_R \tag{4}$$

where ζ_R is the critical damping ratio of the resonator. Pressure inside the cavity can be expressed as [5]

$$p(x,y,z,t) = \sum_i P_i(t)\Psi_i(x,y,z) \tag{5}$$

where $\Psi_i(x,y,z)$ are the undamped acoustic modes of the cavity and $P_i(t)$ are time-varying modal coefficients. If the modes are lightly damped they are an orthogonal set of functions such that

$$\int_V \Psi_i \Psi_j dV = \begin{cases} 0 & i \neq j \\ V_i & i = j \end{cases} \tag{6}$$

This property allows the separation of cavity modes; each modal coefficient has to satisfy the wave equation and initial conditions. The following equation holds [5]

$$\ddot{P}_i + 2\delta_i \dot{P}_i + \omega_i^2 P_i = \frac{\rho c^2}{V_i}\int_V \frac{\dot{Q}}{V}\Psi_i(x,y,z)dV - \frac{\rho c^2}{V_i}\int_S \ddot{w}\Psi_i(x,y,z)dS, \tag{7}$$

where ω_i is the i^{th} natural frequency, V is cavity volume, S is boundary surface. In general δ_i is a complex quantity $\delta_i=\delta_{iR}+i\delta_{iI}$, the real part accounts for wall resistivity, while the imaginary part accounts for wall reactance [5]. Since the compressor cavity has rigid walls and the oil at the bottom of the cavity behaves like a rigid medium, only the real part (modal damping) is significant and it accounts for dissipation due to thermal and viscous phenomena. Modal damping δ_{iR} is related to the critical damping ratio ζ_i by the expression

$$\delta_{iR} = \zeta_i \omega_i \tag{8}$$

The right hand side of equation (7) represents sources of excitation of the cavity. $\dot{Q}$ is the time derivative of volume velocity of the monopole sources assumed to be distributed inside the cavity volume and $\ddot{w}$ is the vibration acceleration of the boundary surface, which is taken as positive if is directed outwards from the cavity. The second integral is not zero only over the vibrating sources and over resonator orifice where $\ddot{w} = \ddot{w}_R$, therefore equation (7) becomes

$$\ddot{P}_i + 2\zeta_i \omega_i \dot{P}_i + \omega_i^2 P_i = \frac{\rho c^2}{V_i}\int_V \frac{\dot{Q}}{V}\Psi_i(x,y,z)dV - \frac{\rho c^2}{V_i}\int_{S'} \ddot{w}\Psi_i(x,y,z)dS - \frac{\rho c^2}{V_i}\int_{A_R} \ddot{w}_R \Psi_i(x,y,z)dS \tag{9}$$

where S' is the area of vibrating sources excluding resonator orifice. Since the area A_R of resonator orifice is small, $\Psi_i(x,y,z)$ is assumed constant over this surface and equal to $\Psi_i(R)$, thus the equation of a cavity mode coupled to the Helmholtz resonator is

$$\ddot{P}_i + 2\zeta_i\omega_i\dot{P}_i + \omega_i^2 P_i = \frac{\rho c^2}{V_i}\int_V \frac{\dot{Q}}{V}\Psi_i(x,y,z)dV - \frac{\rho c^2}{V_i}\int_{S'} \ddot{w}\Psi_i(x,y,z)dS - \frac{\rho c^2}{V_i}A_R\Psi_i(R)\ddot{w}_R \tag{10}$$

In order to couple resonator equation with the cavity, pressure p in equation (4) is expressed as a summation of the pressures caused by the modes of the enclosure. The following set of second order differential equations is obtained

$$\begin{aligned} &\ddot{P}_i + 2\zeta_i\omega_i\dot{P}_i + \omega_i^2 P_i = \frac{\rho c^2}{V_i}\int_V \frac{\dot{Q}}{V}\Psi_i(x,y,z)dV - \frac{\rho c^2}{V_i}\int_{S'} \ddot{w}\Psi_i(x,y,z)dS - \frac{\rho c^2}{V_i}A_R\Psi_i(R)\ddot{w}_R \\ &\ddot{w}_R + 2\zeta_R\omega_R\dot{w}_R + \omega_R^2 w_R = \frac{1}{\rho l_R}\sum_i P_i\Psi_i(x,y,z) \qquad i = 1,n \end{aligned} \tag{11}$$

where n is the number of modes which are taken into account in the dynamic model. If the source of excitation is the vibration of a small part A_s of the boundary surface the surface integral becomes

$$-\frac{\rho c^2}{V_i}\int_{S'} \ddot{w}\Psi_i(x,y,z)ds = -\frac{\rho c^2}{V_i}A_S\Psi_i(S)\ddot{w}_S \tag{12}$$

where $\ddot{w}_S$ is the vibration acceleration and $\Psi_i(S)$ is the constant value of $\Psi_i(x,y,z)$ over the small area A_S.

INTERACTION OF THE RESONATOR WITH COMPRESSOR CAVITY MODES

A typical compressor cavity was considered, it had a volume V of 0.001359m^2 and it was filled with refrigerating fluid R134A, having density ρ=3.73 Kg/m^3 and sound speed c=172.3 m/s at 70 c^0. The first six modes of the cavity were taken into account in the mathematical model; their frequencies, shapes and dampings are summarized in the table.

Mode	f [Hz]	Type	V_i	ζ_i
1	394.4	axial	V/2	0.01
2	534.6	axial	V/2	0.01
3	644.4	axial	V/2	0.01
4	856.6	tangential	V/4	0.01
5	873.2	≈ axial	V/2	0.01
6	893.3	≈ axial	V/2	0.01

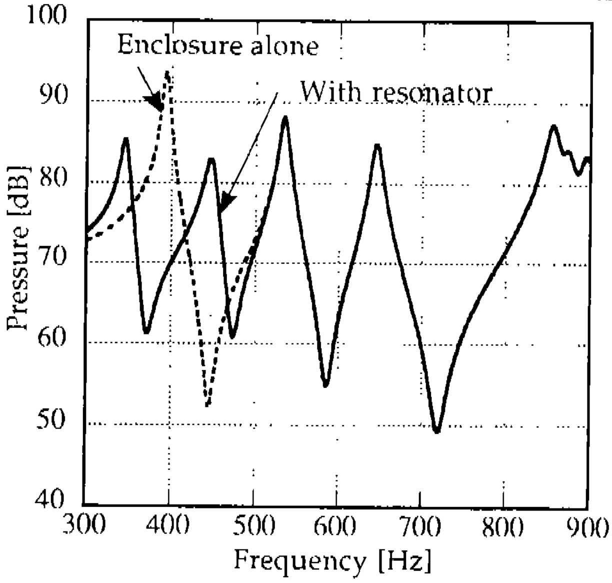

Figure 2: Response spectra ($\Psi_1(R)$=1, $\Psi_i(R)$=0 i=2,6)

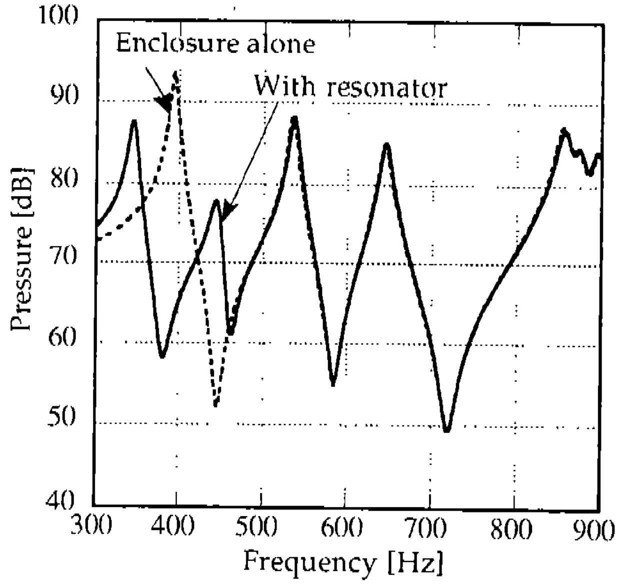

Figure 3: Response spectra ($\Psi_1(R)$=1, $\Psi_i(R)$=0.3 i=2,6)

The effect of a Helmholtz resonator tuned to the first cavity frequency was simulated. Since resonator frequency $f_R = 1/2\pi\sqrt{c^2 A_R / l_R V_R}$ was fixed, only two geometric parameters of the resonator were independent; they were specified by means of the adimensional parameters $\sigma=V_R/V$ and $\tau=l_R/V^{1/3}$; the selected values were: $\sigma=0.033$, $\tau=0.242$. Resonator critical damping ratio was set equal to 0.02, this value is in agreement with several experimental results [1].

As the coupling of the resonator with the cavity modes does not depend on the kind of source, the vibration of a small part of the boundary surface ($A_S=1\times10^{-4}m^2$) was considered being the only source of excitation. This source was assumed to be perfectly coupled with the six cavity modes ($\Psi_i(S)=1$ i=1,6) and to have constant spectral amplitude ($|\ddot{w}_S|=10\ m/s^2$) in a wide range of frequencies. The response spectra of the cavity were calculated in a point P which is an anti-node of the six cavity modes ($\Psi_i(P)=1$ i=1,6).

In the first calculation the resonator was considered coupled with the first mode ($\Psi_i(R)=1$) and not coupled with the other modes. Results are represented in figure 2, where for comparison the response spectrum of the cavity alone is represented too. They show that the resonator cuts down the first resonance peak and produces two new peaks having lower amplitude, but does not affect the response spectrum above 500Hz. In the second calculation a more real case was considered: the resonator was perfectly coupled with the first mode ($\Psi_1(R)=1$) and weakly coupled with the other modes ($\Psi_i(R)=0.3$ i=2,6). The response spectra are represented in figure 3. The frequencies of the two new peaks are not significantly modified, but the amplitude of the first peak increases of about 2 dB, whereas the amplitude of the second peak decreases of about 5 dB. The peaks of higher order modes shows little modifications and their frequencies increase of 1 ÷ 2Hz.

These effects can be explained looking at the response spectra of the six modes, which are represented in figure 4. As $\Psi_i(R)\neq0$ i=1,6, the resonator links together the six modes and not only the first mode is strongly modified by the resonator, but also the other modes are influenced in a frequency range near f_1.

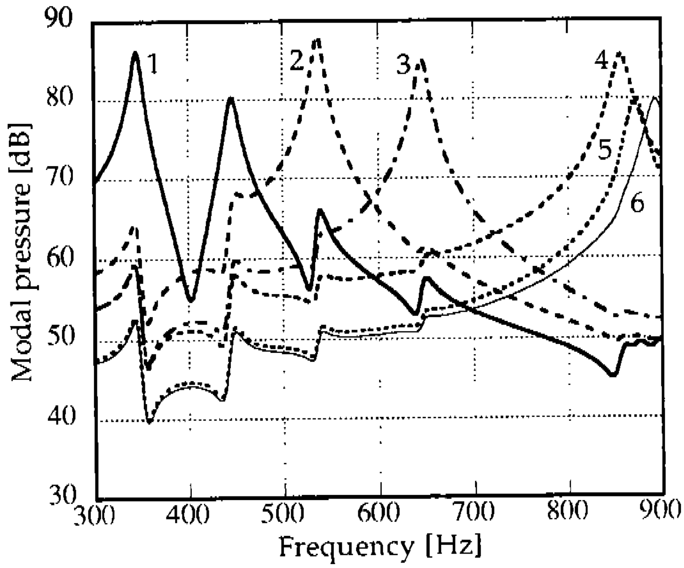

Figure 4: Modal responses (Ψ_1(R)=1, Ψ_i(R)=0.3 i=2,6)

EFFECT OF TUNING ERRORS

A series of resonators with the same resonator/cavity volume ratio (σ=0.033), the same critical damping ratio (ζ_R=0.02) but with different natural frequencies were considered. Their frequencies were chosen in order to produce tuning errors with the first natural frequency of compressor cavity.

The response spectra of the cavity coupled with these resonators were calculated with the assumptions described in the previous section, Ψ_1(R) was set to 1, whereas Ψ_i(R) i=2,6 were set to 0.3. Figure 5 collects the response spectra; the tuning error Δ, which is defined as $(f_R - f_1)/f_1$, is the parameter of the curves. With Δ=-20% the resonator splits in two the first mode of the cavity, but the frequencies of the new resonance peaks are shifted towards the lower frequencies,

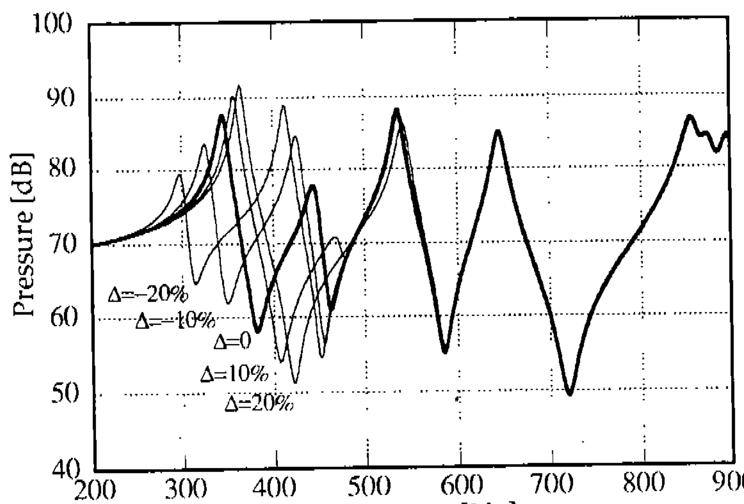

Figure 5: Effect of tuning errors

the second peak is higher than the first.If Δ=-10% the two resonance peaks of the new modes have approximately the same height and are lower than the highest peak which appears when Δ=0. The system exibits this good behaviour because the tuning error probably compensate for the effect of higher order modes.

With positive values of the tuning error the two resonance peaks which are caused by the splitting of the first cavity mode move towards the higher frequencies; the first peak becomes higher, the second becomes smaller and for Δ=20% it merges with the resonance peak of the second mode of the cavity. In the case Δ=20% the resonance peak of the second mode of the cavity is appreciably influenced by the resonator.

EXPERIMENTAL TESTS

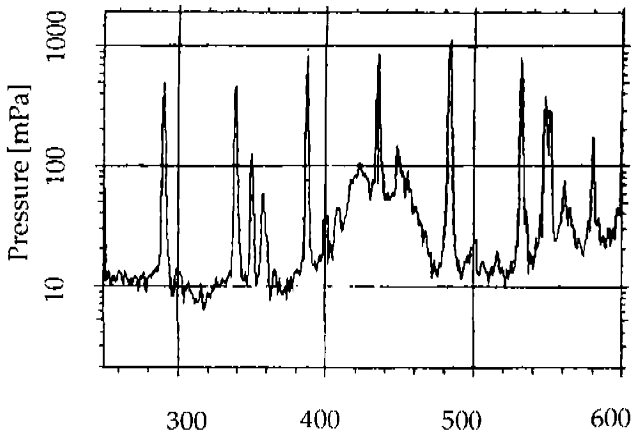

Figure 6: Measured response spectrum of the cavity

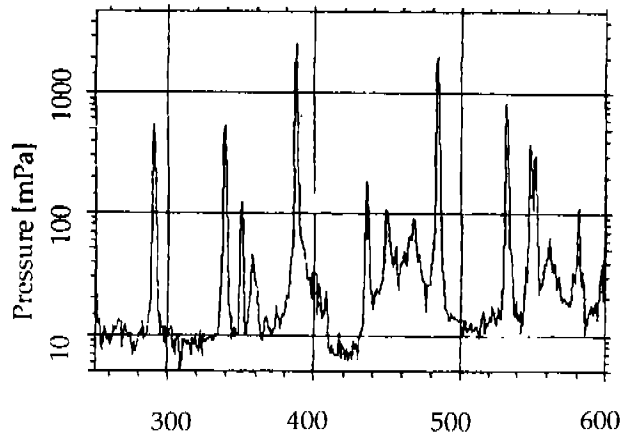

Figure 7: Measured response spectrum of the coupled system

The experimental tests were carried out on a compressor cavity during the actual working of the compressor. The refrigerating fluid was R12, the inlet pressure (cavity pressure) was 1.24 Bar, whereas the outlet pressure was 13.7 Bar, the temperature inside the cavity was 70 c^0, therefore ρ=5.4 kg/m^2 and c=161 m/s. The compressor shell had an elliptical cross section, the first mode of the cavity was axial in the direction of the major axis of the ellipse, whereas the second mode was axial in the direction of the minor axis of the ellipse. The resonator orifice was located along the major axis of the ellipse, in this way it was well coupled with the first mode of the cavity and very weakly coupled with most of the higher order modes. Resonator duct had fixed dimensions, whereas resonator volume was adjustable and allowed to achieve the optimum tuning of the resonator with the first cavity mode. A microphone pressure transducter was located near the resonator orifice in a position not far from the major axis of the elliptical cross section; therefore it was well coupled with the first mode and weakly coupled with the second. Microphone signal was amplified by a charge amplifier and then analized by a narrow band spectrum analyzer.

The response spectrum of the cavity alone is represented in figure 6. Compressor working causes background noise, a series of periodic spectral peaks (period $\approx$50 Hz) and some irregular peaks at about 350 Hz and 550 Hz. The resonance peak of the first mode is at about 435 Hz and it amplifies one of the periodic peaks of excitation. The second resonance peak is at about 560 Hz and is less clear than the first

After some attempts the following optimum values of resonator parameters were selected: f_R=425Hz, σ= 0.011, τ=0.169. The measured response spectrum of the cavity coupled to the resonator (figure 7) is rather different from the response spectrum of the cavity alone. The resonance peak at 435 Hz disappears, whereas the spectral amplitudes increase in the ranges of frequency around 400 Hz and around 460 Hz, the periodic peaks which are encompassed in these frequency ranges become higher.

COMPARISON BETWEEN ANALYTICAL AND EXPERIMENTAL RESULTS

A series of calculations was performed with the aim of verifying if the mathematical model was able to simulate the measured behaviour of the cavity. As the experimental results showed that the spectrum of the exciting machanisms contains a series of periodic peaks and background noise, two sources of excitation were considered in the calculations. The first source (a small vibrating surface) had a spectrum with a series of periodic peaks like those measured experimentally. The second source (a small vibrating surface) had a constant spectral amplitude.

In the first simulations only the first mode of vibration of the cavity was taken into account and it was considered perfectly coupled with the sources and with the microphone. The response spectra of the cavity alone and of the cavity perfectly coupled with the resonator were calculated at microphone location. They are represented in figure 8 and show that the introduction of the resonator eliminates the original resonance peak at 435Hz and produces two new resonance peaks at 398Hz and 464 Hz respectively; hence the component of the forcing spectrum at 435Hz is cut down, whereas the components at 385 Hz and at 485Hz are amplified.These results are in agreement with those obtained experimentally and show that the single-mode analysis foresee most of the effects produced by the resonator. Above 500Hz the differences between the predicted and the calculated responses are greater because the higher order modes are not taken into account in the model.

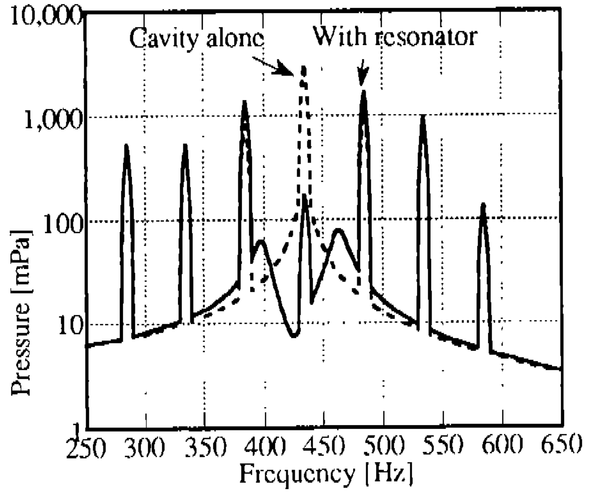

Figure 8: Simulation of experimental results, one mode in the model

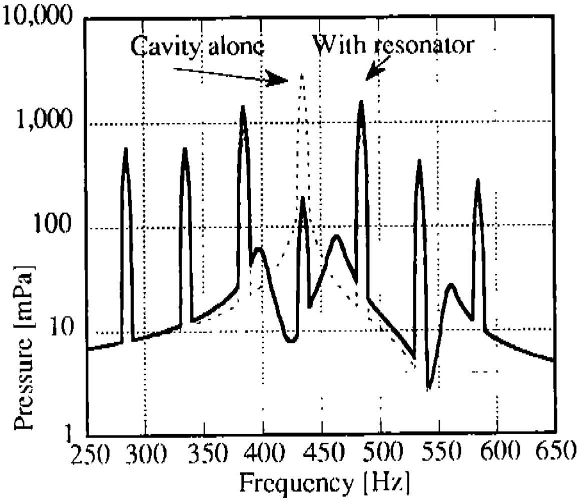

Figure 9: Simulation of experimental results, two modes in the model

Other calculations were then carried out considering the first and the second acoustic modes of the cavity. In order to simulate the experimental apparatus, where the microphone is not exactly in the nodal plane of the second mode, the microphone was assumed perfectly coupled with the first mode and weakly coupled with the second; the resonator was assumed coupled only with the first mode. The sources of excitation were the same of the previous calculations and were coupled with both modes. The response spectra which are presented in figure 9 show that the introduction of the second cavity mode in the mathematical model improves the agreement between calculated and experimental results above 500Hz.

CONCLUSIONS

The influence of a Helmholtz resonator on the acoustic vibration of the cavity of a domestic refrigerator compressor was studied both analitically and experimentally. The analytical model was useful to analyze separately some particular effects which are present when a resonator is coupled to a real compressor cavity. Results showed that in most cases the resonator was still effective, even if tuning errors and multi-mode coupling alter its performance. The experimental tests showed that the introduction of a resonator tuned to the most excited mode of a real compressor cavity caused a remarkable modification of the response spectrum with a large reduction of the resonance peak. Owing to the many sources of excitation, which are present in the actual compressor, the cavity response spectrum is rather complex; nevertheless a good agreement was found between experimental and theoretical results which confirmed the validity of the multi-mode analytical model.

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