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NOISE REDUCTION OF HERMETIC COMPRESSOR BY IDENTIFICATION OF THE GAS CAVITY RESONANCE

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ABSTRACT

Recently, a new reciprocating compressor for domestic appliances has been developed at I.R.E. Compressor Development Dept., by way of its high efficiency, quiet operation, light weight and small size. This so called PB compressor ranges from 40 W to 140 W (mini line of reciprocating compressors of I.R.E. Production) by 5 different models referring to the displacement and the cooling capacity.

This compressor has been mainly conceived for the European market, where a very low operating noise emission is needed to be considered as a Top Value Product. Hence, the acoustic engineers made big efforts on noise reduction by suggesting special design of the mechanical parts to guarantee an extremely low noise level.

An interesting approach to reduce the noise emission at 500 Hz (1/3 octave band) occurred among different kinds of investigations on the overall noise reduction activity.

This paper presents as follow: the identification of the transmission path, the description of the law describing the behaviour of the 500 Hz frequency (first with experimental tests and then with Finite Element Method calculations) and finally the implemented technical solutions.

INTRODUCTION

The hermetic shell of the compressor generally serves as a noise container, but the cavity between the compressor and the shell, filled with gas, has its own natural frequencies that may reinforce the pumping harmonics (Acustic Resonances).

As the gas is in contact with a large area of the shell, when the Gas Cavity natural frequencies are excited by the pumping harmonics, sufficient pressures are developed in the Gas Cavity to drive the shell and radiate noise.

Generally, the characteristic impedance of the gas is significantly less than the impedance of the compressor shell and therefore a little sound energy is transmitted via this path to the shell.

This transmission path can be instead significant since the internal compressor system has many strong excitation sources in direct contact with the gas, whose frequencies could be aligned with the Gas Cavity natural frequencies.

This paper deals with the investigations on the Gas Cavity resonance close to the 11th pumping harmonic (532 Hz at 60 deg.C and 2920 r.p.m.) of the PB compressor.

COMPRESSOR DESIGN

The design of the PB reciprocating compressor is shown in Fig.1 (see Appendix). Besides the high performances required, the aim of this new model was a strong reduction of the component parts in order to make an automatic assembly.

The cross section (Fig.1 in Appendix) give enough details to understand the main patented features of the compressor e.g.: four suspension springs directly coupled with the stator screws, discharge muffler far from the cylinder to increase the heat transfer from the cylinder, suction side yet arranged for adopting direct suction, suction muffler location on the special design of the cylinder head and a new original design of the connecting rod (spherical bushing coupled with a sintered connecting rod via a metallic spring as a retainer).

NOISE TRANSMISSION PATH IDENTIFICATION

The PB6 compressor (a 60 W cooling capacity model; LWS target: 33.5 dBA at bench), yet at the beginning of the noise measurements, displayed an unceasing high level at 500 Hz (1/3 octave band) either at bench or on appliances (usually a 362 litres Double Door refrigerator).

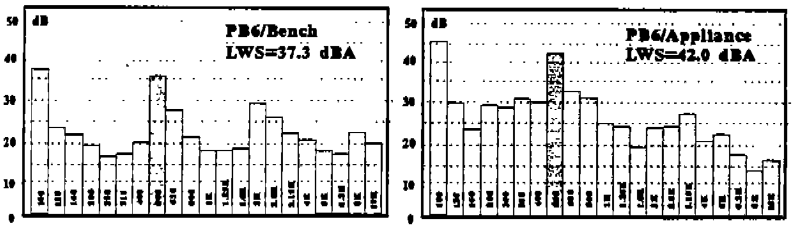

FIG.1 Sound power level spectrums of PB6 type compressor. At bench (left) and on appliance.362 litres Double Door (right)

All the measurements have been carried out in a reverberation room of about 150 cubic meters and all the noise tests have been performed according to CECOMAF Specifications (-25 deg.C evaporating temperature , and +55 deg C. condensing temperature) at bench and according to ISO 3741 Standard on appliances.

By considering the 500 Hz (1/3 octave band) reduced to the more usual level of 30 dB instead of 42.3 dB as in Fig.1 above (right side), because of the high weight of this 1/3 octave band, the final Sound Power Level could became 38.5 dBA i.e. 4 dBA less.

The first approach to analize this noise emission around 500 Hz, has been a statistical analysis by collecting all the noise measurements of many PB6 compressors either assembled on different types of appliances or tested alone at bench.

First considerations have been made after organizing all the available data : a high 500 Hz (1/3 octave band) didn't occur at bench measurements at all time, but no compromise seemed possible : or 500 Hz (1/3 octave band) just around 30 dBA or higher than 40 dBA (of course occasionally some cases showed 35-37 dBA, but out of the statistical significance level).

The situation seemed more clear by considering the noise spectrums of different families of appliances equipped with the PB6 compressor i.e. the amplitude of the 500 Hz (1/3 octave band) was primarily depending on the type of appliances.

Was then evident that the families of appliances could be divided into two families : with or without a high level of the 500 Hz (1/3 octave band) frequency.

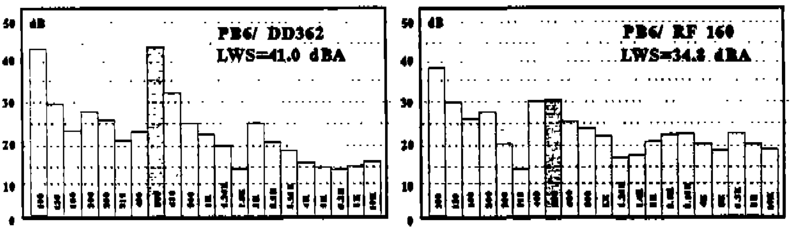

FIG.2 Sound power level spectrums of D.D.362 litres and 160 l refrigerators both equipped with the same PB6 compressor.

Fig.2 give a typical noise spectrum of the 362 litres Double Door refrigerator and the 160 litres refrigerator.

All the measurements on the DD 362 showed a high level of the 500 Hz (1/3 octave band), and all the measurements on the 160 litres refrigerators, pointed out a low 500 Hz level. This was true even if the compressor, disconnected from a cabinet (e.g.160 litres) with a very low 500 Hz (1/3 octave band) level, was assembled on a DD 362 family cabinet (Fig.2).

Lastly, even by a proper organization of all noise measurements, seemed right to conclude that there was not a strong correlation between the values of the 500 Hz (1/3 octave band) amplitude on the compressor tested at bench or tested on appliances (a low/high 500 Hz noise level at bench didn't always mean a low/high 500 Hz noise level on appliances and vice versa).

Neither by testing the PB6 compressor at bench at the same suction and discharge pressure of the 362 litres and the 162 litres refrigerators (i.e. 1.23/13.64 Bar abs. and 1.14/12.87 Bar abs. respectively), was possible to reproduce the 500 Hz (1/3 octave band) behaviour of the compressor tested on appliances.

On the contrary, the same behaviour of the 500 Hz (1/3 octave band)occurred both at bench and on appliances by modifying the compressor position from the usual horizontal position.

Indeed both at bench and on appliances the 500 Hz (1/3 octave band) noise level immediately dropped when the compressor (or the cabinet) was tilted of about 25-30 deg. around the axial line (piston axis), toward the name plate side (Fig.3).

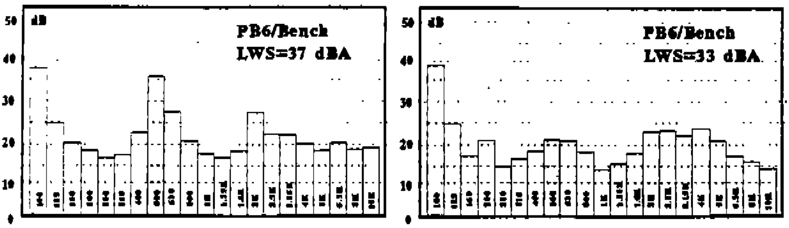

FIG.3 Noise level spectrum PB6 compressors at bench.
Normal position (left), 30°tilted (right).

The 500 Hz (1/3 octave band) noise level immediately became to the high previous value again when the compressor (or the refrigerator) got back to the steady-state position. Fig.3 could be read indifferently from left to right an from right to left both for noise test at bench and on appliances.

After that, some considerations have been made in order to discover the parameters involved in the behaviour of the 500 Hz (1/3 octave band) in all the above situations.

GAS CAVITY RESONANCE

Taking into account all these informations, a list of different hypothesis have been drawn up and a set of different tests have been arranged in order to refuse or accept the influence of each weighted hypothesis.

Oil level	Suction noise	Shock loop stresses
Rotor gravity force centre	Suction tube position	Load on suspension springs
Oil pump draft	Winding immersion level	Oil pump position
Pressures ratio	Gas cavity temperature	Gas cavity shape
Gas cavity volume	Shell resonances	Gas properties

By adopting the direct method and by taking into account the informations from measuring the sensitivity of the compressor noise with the compressor operational parameter modified, it has been possible to significantly reduce the hypothesis number.

During these tests only a single operating parameter was varied to the nominal value each time. Within the repeability of the noise measurement, any change in the compressor noise was then ascribed to the compressor sensitivity toward the tested parameter.

Major changes in the 500 Hz (1/3 octave band) amplitude occurred by adopting different kinds of gasses. For practical reasons R12, R22, R134A, Helium and Dry Air have been used.

In order to deeply investigate this phenomenon some structured experiments have been set-up with two special compressor configurations (Fig.2 in Appendix).

In both cases a small loudspeaker (response from 200 Hz to 5000 Hz) was fixed inside the compressor housing just above the compressor body (Fig.2 in Appendix).

The shell vibrations were measured by using some small accelerometers opportunely fixed at many different shell locations.

The first configuration took into account the only compressor housing filled with gas (with the pump group and the electric motor removed). The oil level was fixed to the standard value. The second configuration took into account the standard PB compressor (i.e. mechanical parts included) filled with gas at the right oil level value again.

No important differences between the vibrations level of the 500 Hz frequency (1/3 octave band range) were recorded in these two configurations (i.e. the volume of the cavity, filled with the same gas type, seemed no significant on the vibration level measured on the shell). Larger changes were on the contrary obtained by modifying the cavity shape when tilting the compressor as described above (i.e. the unchanging horizontal oil surface, modified on the contrary, the shape of the cavity).

The behaviour of the Gas Cavity resonance was mostly affected by the gas properties; by using R12 the resonance peak occurred at 520 Hz while with Helium and Dry Air the resonance peaks respectively occurred at 750 Hz and 700 Hz. Furthermore the highest shell vibrations level occurred when filling the compressor housing with R12.

The high shell vibrations level around 500 Hz could be explained as an alignment between a loudspeaker exciting frequency and the gas cavity natural frequency just around 500 Hz (even if the shell hadn't any natural frequency close to 500 Hz). Further experiments strenghtened the theory of the Gas Cavity Resonaces.

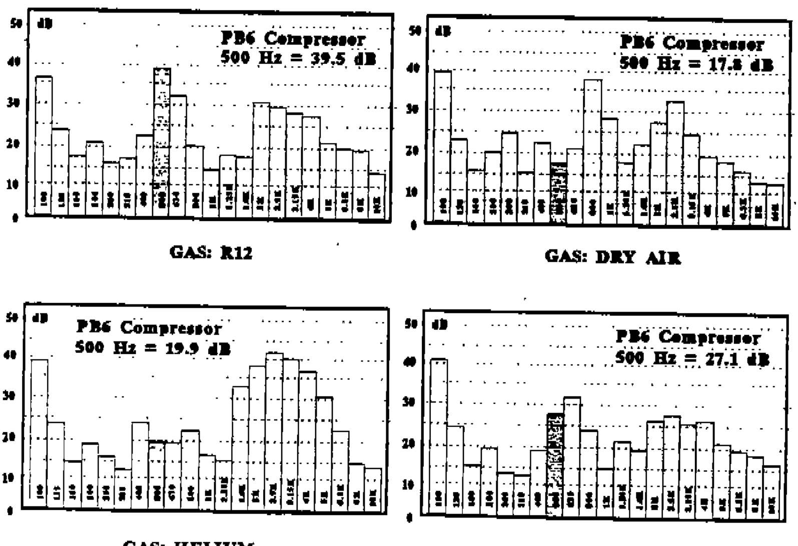

FIG.4 Noise spectrums at bench by using different types of gas on operating PB6 compressor(1.23 /13.64 bar Abs.pressure)

The effect of the refrigerant properties has been hence directly investigated by measuring that any change in the sonic velocity produced a corresponding change in the cavity resonant frequency at running conditions. To confirm this,different gasses have been substituted to Refrigerant 12 inside the shell cavity and the noise, at Standard Conditions, measured on the operating compressor (Fig.4).

Hence, both considering the loudspeaker excitation or the pumping harmonics excitation, under fixed thermodynamical conditions, the resonance around 500 Hz (Gas: R12) was strong enough to excite the shell and then radiate loud noise.

GAS CAVITY LOCATION

As the gas temperature inside the shell rises, the sonic velocity of the gas increases causing a shift in the Gas Cavity resonant frequency.

The gas temperature was hence varied and the effects have been examined as far as to localize the resonace conditions from the narrow band noise analysis. The vibrations coming out from small accelerometers mounted on fixed locations on the housing, are given in Fig.6. For practical reasons the gas temperature has been assumed equal to the measured top shell temperature. All the measurements have been carried out at bench under the same operating conditions (-25 deg.C, +55 deg.C).

The Gas Cavity Resonace was hence identified near the 11th pumping harmonic by changing the gas temperature of the cavity and by examining the shift of the 21.8 dB peak located between the 10th and the 11th pumping harmonic (first at 40 C). At 60 deg.C the Gas Cavity resonance occurred at 532.5 Hz frequency (Fig.3 in Appendix).

Additional measurements have been also arranged in a special diagram where both the noise level of the 500 Hz (1/3 octave band) and the Gas Cavity narrow band analysis have been plotted at different gas temperatures (Fig.5).

The 500 Hz noise level (1/3 octave band) versus gas temperature is described on the left side of the graph, and the Gas Cavity natural frequency (narrow band analysis) versus temperature is described on the right side.

On the same graph the shift of the 11th pumping harmonic of the compressor assembled on appliance (a 362 liters Double Door) from startup to normal operating condition has been also described.

The continuous curve in the middle of the darkened area, describes the behaviour of the 500 Hz (1/3 octave band) of the PB compressor, which Gas Cavity natural frequency was 532.5 Hz at 60 deg.C (related to the continuous curve of the shaded area on the right side).

The shaded area represents the family lines describing the shift of the Gas Cavity frequency of a large number of conventional PB6 compressors on appliances, the darkened area means the alignment between the 11th pumping harmonic and the Gas Cavity frequency in the temperature range from 57 deg.C to 73 deg.C i.e. resonance just at the operating compressor conditions on appliances.

The two lines, both on the right and on the left of the above mentioned family lines, show the behaviour of the modified compressor after studying the Gas Cavity behaviour with Finite Element Method calculations (far enough from the conventional PB6 Gas Cavity area).

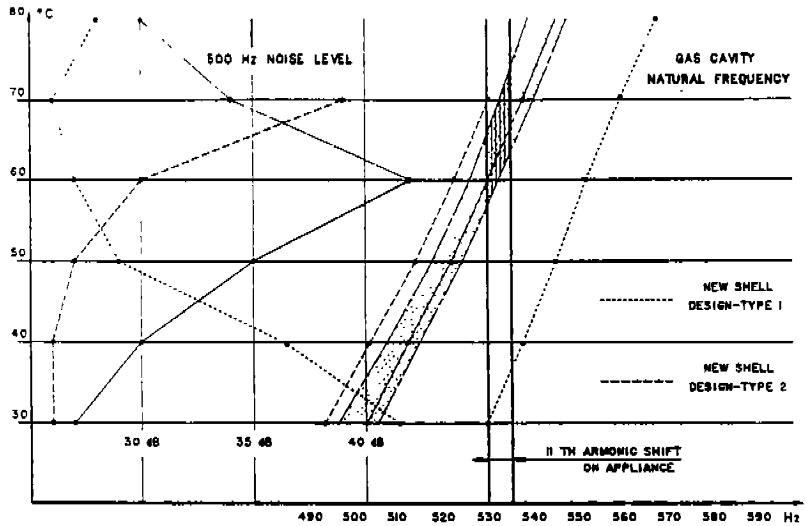

FIG.5 Relationship of 500 Hz noise level (1/3 octave band) and Gas Cavity drift with temperature.

GAS CAVITY SIMULATION

Taking into account the Gas Cavity behaviour, the first attempt was to shift the driving frequency by modifying the electric motor characteristics. The gain got on the 11th harmonic was only about 5 Hz, not enough to completely solve the problem.

Hence two approaches were only possible: or modify the mechanical compressor response (i.e. shift of the pumping harmonics) or modify the shell shape (i.e. shift of the gas cavity frequency).

Besides experimental investigations, a Finite Element Analysis on compressor components has been carried out in order to make a numerical simulation of both structural and acoustic system response.

The Finite Element Code "ABAQUS" Version 4.7 has been used. Since one of the origin of noise was due to the mechanical vibrations, able to excite the eigenfrequencies of the Gas Cavity, the analysis was separated in three different steps as follow:

- Dynamic analysis on compressor body, submitted to a cyclic pressure load, to compute the vibrations of the components in terms of displacement and frequency i.e. attempt to shift the 11th pumping harmonic (not included in this paper).
- Modal Analysis of the Gas Cavity in order to know the eigenfrequencies and the eigenmodes of the Gas Cavity.
- Coupled acoustic-structural analysis in order to know which of the modes are excited on the coupled system (Gas Cavity and compressor shell).

MODAL ANALYSIS OF THE GAS CAVITY

A modal analysis of the gas cavity was made in order to know the first modes, suspected to be the main cause of noise for some frequencies, particularly the 532.5 Hz found from the previous noise analysis.

The wave lenght corresponding to the frequencies around 532 Hz (when the gas is at the steady-state pressure and temperature) was near to the distance between the principal opposite walls of the shell i.e. parallel and perpendicular to the piston axis. Therefore the two principal cross sections of the cavity shape were considered.

The two Finite Element models are shown in Fig.4 in Appendix. "Acoustic" in plane elements have been used, available in ABAQUS, for modelling an acoustic medium undergaing small pressure changes.

The analysis results are shown in Fig.5 and Fig.6 in Appendix. In the larger section the first mode (462 Hz) acts between the shell surfaces along the greatest distance, while the second mode (520 Hz) is in proximity of the 530 Hz frequency. In the shorter section the two first modes act in the same directions of the preceeding areas, with respectively 523 Hz and 535 Hz frequencies.

Three modes may be the cause of noise: the second in the larger section and the two first in the shorter section, because of their proximity to the 532.5 Hz frequency of the pumping harmonic (Fig.4 and Fig.5 in Appendix).

COUPLED ACOUSTIC-STRUCTURAL ANALYSIS

In order to know which of the modes are excited in the real system a "steady-state dynamic analysis" on the half 3-D model was made.

The analysis consisted in computing displacements and pressure acoustic levels based on the harmonic excitation, selected by the user.

In the previous analysis, only the acoustic cavity was considered, while in this case the complete system is needed: structural (i.e. the shell, able to radiate in space the vibrations of the gas inside) and acoustic (gas and oil).

In the model, shown in Fig.7 in Appendix, the three systems are linked by some "interface" elements able to couple the structural and acoustic variables of modes (displacements and pressure). A certain number of harmonic excitations were imposed to the shell points corresponding to the locations of the suspension springs, in the 400 Hz / 800 Hz range.

The response of the system is shown in Fig.8 in Appendix in which the pressure level is plotted in the frequency domain, for some significant points in proximity of the shell surface.

Only one mode is excited; the second mode in the shorter section, with a frequency of 532 Hz, acting between the opposite surfaces of the shell.

The 535 Hz mode was hence highlighted, and starting from this result some modifications on the shell shape have been analyzed in order to obtain a shift of this 535 Hz natural frequency.

The graphs in Fig.5 show the results of modifications on the PB compressor shell after the Finite Element Analysis: the two plotted solutions show a shift of this longitudinal second mode frequency, respectively left and right, far enough from the 532.5 Hz frequency.

CONCLUSION

The different noise level of a compressor tested at bench or on apliiance can be explained by considering the different trend of the gas cavity temperature in these two tests. The measurements at bench are collected in a steady-state temperature sometimes different from the gas temperature of the gas cavity on appliances.

Furthermore, the standard method for noise measurements on appliances consider a weighted average of many noise spectrums collected during a complete cycle of the cabinet. During this cycle the temperature of the gas cavity moves in a broad range of values from the lowest value at startup to the highest value just before the switching off of the thermostat.

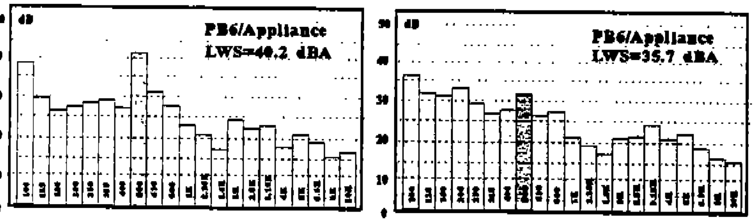

FIG.6 Typical sound power level spectrums.First shell design (left), modified shell design (right)

This aspect of noise control must be taken into account in the compressor design. Big improvements have been reached in the PB compressor design by modifying the shell shape. A 3 dBA reduction in the Sound Power level has been obtained and 10 dB less in the 500 Hz (1/3 octave band) (Fig.6).

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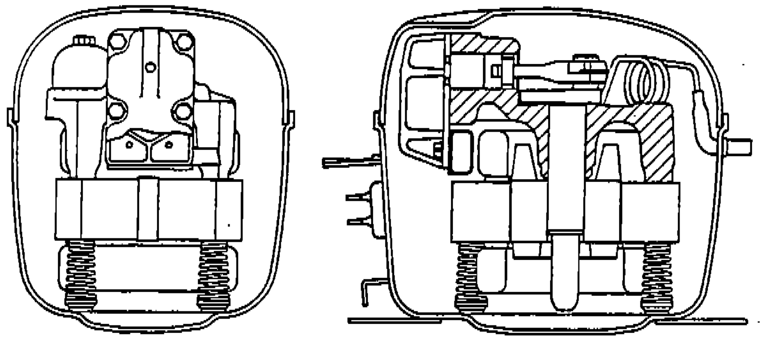


FIG.1 Appendix. Cross sectional views of the PB compressor.

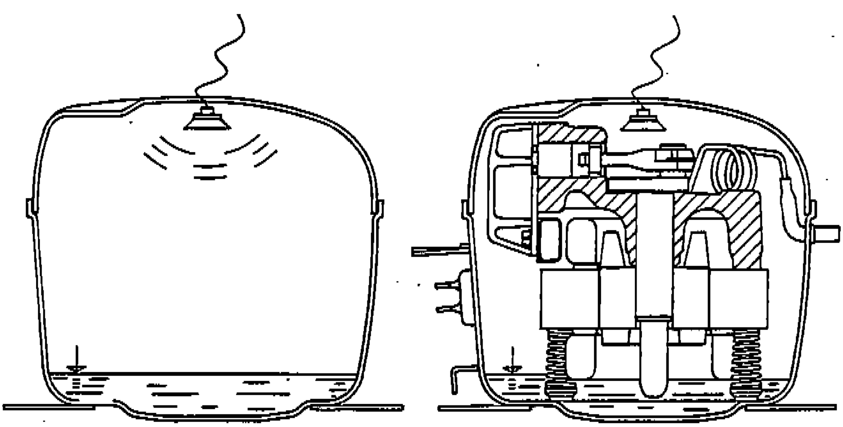
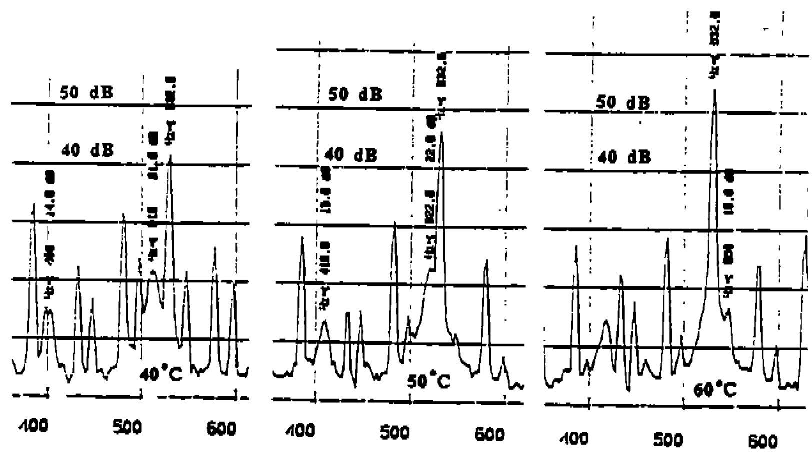
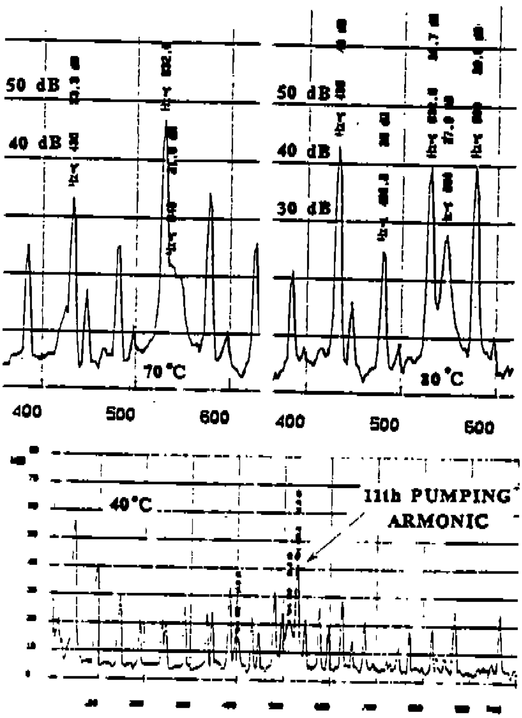


FIG.2 Appendix. Experiments set up.





GAS CAV. Temp. °C	GAS CAV. Freq. Hz	GAS CAV. Vib. dB	11th Armonic Vib. dB
40	515	21.8	41.8
50	522	22.8	46.4
60	532	54.1	54.1
70	540	21.8	46.6
80	560	27.8	38.7

532.5 Hz = 11th armonic

FIG.3 Appendix. Drift of the Gas Cavity natural frequens versus temperature. Vibration level of the compressor shell.

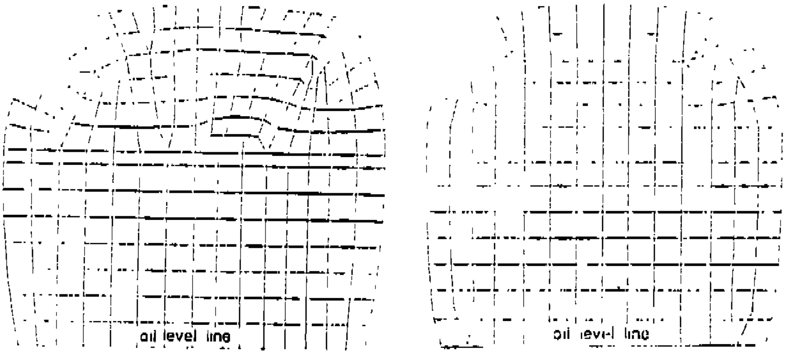


FIG.4 Appendix. Finite elements model.
Longitudinal section (left), transverse section (right).

FIG.5 Appendix. Gas Cavity eigenfrequencies longitudinal section.
First mode left(462 Hz), second mode right (520 Hz).

FIG.6 Appendix. Gas Cavity eigenfrequencies. Transverse section.
First mode left(523 Hz), second mode right (535 Hz).

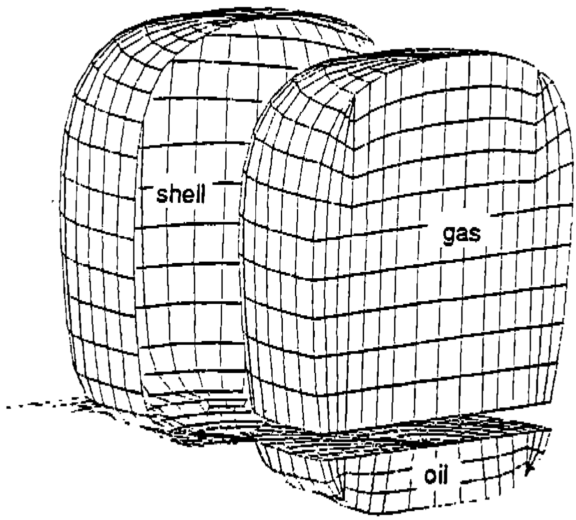


Fig.7 Appendix. Coupled acoustic–structural system.

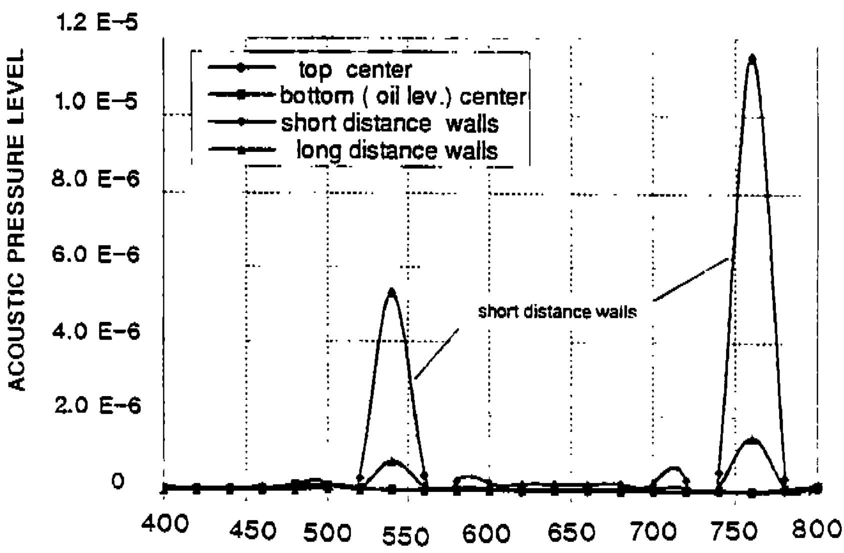


FIG.8 Appendix. Coupled acoustic–structural system. Gas Cavity acoustic pressure in the frequency domain.