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ABSTRACT

In a reciprocating hermetic compressor. the suction system is a noise source related to the compressor fundamental frequency and higher orders, as well as the frequencies generated by suction valve lift. In this way the intake noise is a source for airborne noise, and forces the acoustic gas cavity modes, which excite the compressor shell that is responsible of the radiated noise. In the development of a suction muffler numerical or analytical acoustical codes are used to predict the acoustical behaviour of the system as far as the noise reduction is concerned.

This paper shows a correlation of the theoretical results of a 3D numerical code, and a 1D analytical code, compared to the experimental results. As a result it is possible to conclude that a 1D acoustical analytical code is accurate enough if some geometrical conditions are fulfilled. Furthermore the application of a concentric-hole cavity resonator is shown.

NOMENCLATURE

c_o = Speed of sound	$j = \sqrt{-1}$ Imaginary unit
$\mathbf{i}, \mathbf{j}, \mathbf{k}$ = Axis versors	$\nabla = \mathbf{i}\frac{\partial}{\partial x} + \mathbf{j}\frac{\partial}{\partial y} + \mathbf{k}\frac{\partial}{\partial k}$
K = Wave Number	l = Duct Length
p = Pressure	ρ = Density
s = Duct Section	SPL = Sound Pressure Level
u = Mean Flow Velocity	Vol = Volume of Expantion Chamber
v = Acoustical Mass Velocity	vs = Acoustical Mass Generator
Z_R = Radiation Impedance	Zs = Internal Source Impedance
Zo = Transfer Impedance	

INTRODUCTION

Intake noise

The opening of the suction valve during the movement of the piston from the top dead centre (TDC) to the bottom dead centre (BDC), impresses to the gas in the suction system an acceleration throughout the components that make up the system. The gas is further accelerated and decelerated by the elastic behaviour of the suction valve. During the closing of the suction valve, the velocity of the gas is decreased and modulated by the elastic behaviour of the valve; than the gas is decelerated till a zero velocity is reached. The fluctuation of the signal is naturally modulated by the acoustic response of the inlet system. Then the components which make up the system contribute to the spectral noise distribution associated with the periodicity of the suction process.

Experimental verification

Experimental tests of compressor artificial excitation (with electrodynamic shaker).were conducted exciting a compressor body with a signal previously recorded during compressor running. Test results in fig. 1 show that it is not possible to achieve the same acceleration levels on the compressor shell with the structural excitation only; so, airborne noise in a certain frequency range do not depend by a structural path. Furthermore, in fig. 2 are shown spectral estimations of the SPL inside the gas cavity between a compressor whose suction system was externally connected to an auxiliary tank (no interaction with compressor cavity) and a compressor with a standard suction system. From fig 1 and 2 it is possible to define for the intake noise a low frequency range from 160Hz to 400Hz, and a high frequency range from 3000Hz to 5500Hz

ACOUSTICAL MUFFLER DESIGN

Simulation methods

The application of simulation models in the design phases of an acoustical silencer allows a faster definition of the components, more choices among different design alternatives and in some cases, a severe reduction of the experimentation. Acoustic simulation of silencers are based on different analytical methodologies. Among those are:

Solution of the acoustic wave equations

Finite Element Methods

Generally the acoustic behaviour of each component which makes up the circuit, is determined by the solution of the system of equations for compressible, inviscid, fluids with the hypothesis of isentropicity. The equations of the system in the general case of 3D propagation are (1) :

MASS CONTINUITY $$\rho\nabla\cdot\mathbf{u}+\frac{\partial\rho}{\partial t}=0$$

DYNAMICAL EQUILIBRIUM $$\rho\frac{\partial\mathbf{u}}{\partial t}+\nabla p=0$$

ENERGY EQUATION $$\left(\frac{\partial P}{\partial\rho}\right)_s=c_0^2$$

Those equations are linearized assuming small perturbances superimposed to the equilibrium state, eliminating u from the first two equations this lead to the homogeneous equation :

$$\frac{\partial^2 p}{\partial t^2}-c_0^2\nabla^2 p=0$$

Furthermore, if we assume that the variation in time of the pressure at any given point is sinusoidal.

we obtain the Helmholtz equation: $\nabla^2 p + k^2 p = 0$ where $k = \omega/c_0$ is the wave number and $p = pe^{j\omega t}$ the complex pressure amplitude.

Nevertheless for frequency below some critical values, depending of the transversal dimensions of the muffler 'wave propagation can be assumed plane. Thus, the previous equations can be projected along the streamlines of the flow, corresponding to the line of propagation of the acoustic disturbances. The 1D analytical code that we used in muffler design, uses the electroacoustic analogies (2). The mathematical model of the muffler is represented by an electrical network where the noise source is expressed by an acoustical mass generator v_S or pressure generator p_S, with internal impedance Z_S; the acoustical system is then closed on the radiation impedance Z_R, of the external field (see fig. 3). Obviously v_S, p_S, Z_S, are univocally tied to the acoustic characteristics of the circuit. In this formalism the muffler circuit is described by a transfer matrix connecting the pressure p_1 an the mass velocity v_1 at the input to the corresponding physical observables at the output. The elements of the global transfer matrix are calculated for suitable combinations of the elements of the transfer matrices for basic acoustic components. The used elements are the following:

Constant area duct, variable area duct, expansion chamber, inversion flow chamber, extended tube resonator, concentric hole-cavity resonator.

In fig. 4 is represented an example of transfer matrix; In fig.5 is represented a scheme of a suction system of an hermetic compressor.

Muffler acoustical performance

In the 1D acoustical code used for muffler design, the performance of a muffler is measured in terms of one of the following parameters:

Transmission Loss $\quad TL = 10LOG\dfrac{W_i}{W_t}$

Noise Gain $\quad NG = 20LOG\dfrac{p_o}{p_i}$

Transfer Impedence $\quad Z_o = 20LOG\dfrac{p_o}{v_i}$

Where w_i is the incident acoustical Power, w_t is the transmitted acoustical Power. The NG parameter is used in comparison with experimental data; the parameter Z_0 provides the indication of the resonant frequencies of the system. In the next table is shown a comparison between the resonance frequencies up to 3000 Hz calculated with the 1D analytical code and with a 3D numerical code for the circuit of fig.5. It is to point out the presence of the 1° transversal mode (shown by the 3D code) at the frequency of 2253 Hz due to the transversal dimension of the large central expansion chamber. Furthermore the good correlation between 1D and 3D codes in the response at longitudinal modes becomes evident. It is necessary to point out that for the comparison with 3D code were not considered damping effects in the circuit.

MODE	1D code (Hz)	3D code (Hz)
1	77	72
2	440	411
3	1010	1001
4	1790	1645
5	2290	2253
6	------	2351*
7	------	2363*
8	2747	2709

* Transversal mode

Experimental setup

To verify theoretical data, methodological experimental tests have carried out in laboratory controlled conditions for a correct evaluation of the acoustic performance of mufflers; this was done in the perspective of real working. This methodology (see fig.6) provides the acquisitions of input pressure p_i, and output pressure p_O, generated from the excitation of the muffler circuit by an acoustical mass generator positioned in correspondence of the suction valve section. The muffler circuit is filled of R134A refrigerant at a reference pressure. In this configuration the experimental NG is evaluated as the transfer function of the two pressures, precisely $NG = |H_1(f)| = \left|\frac{G_{io}}{G_{ii}}\right|$, where G_{io} is the Cross spectrum of input output, and G_{ii} is the Power spectrum of the input. Particular attention has to be paid to the comparison between experimental data and theoretical data. The input pressure has been measured in a different point compared to the real inlet point, so the simulation technique has to consider the real points of input and output. The inlet microphone always feels the effect of the standing wave that is present in the place where the microphone is set: this might mean an increasing or a decreasing of the pressure amplitude and then a wrong Transfer Function for that frequency. Then the Transfer Function analysis data must be carried on with the input and output Power spectra analysis data. In fig. 7 a comparison between experimental NG and calculated NG for the circuit of fig. 5 has been made.

Optimization of a new muffler

With the experimental and design tools above mentioned we have obtained the development of suction muffler that follows those simples rules: a number of expansion chambers that give the better TL at a given frequencies number. The higher expansion ratio compatible with the available geometrical dimension and flowdinamics performance. In fig. 8 a comparison between the TL of the new muffler design and muffler design of fig. 5 has been made.

Furthermore in fig 9 is shown a comparison between the TL of the new muffler design and a version of the same with the terminal application of a concentric hole-cavity resonator (muffler external dimensions don't change).

CONCLUSIONS

The aspects of the intake noise generation, as well as the experimental techniques to characterise the intake noise frequency range, has been pointed out.

The theoretical aspects of a simulation model has been shown, as well as the mathematical approach of the uni-dimensional acoustical code used in muffler design. Experimental set-up method and problems concerning to the comparison of the experimental data and calculated data has been discussed

The comparison between the NG calculated with a uni-dimensional model and the measured one shows a good agreement in the frequency range where the sound propagation can be assumed in the form of plane waves. Similar conclusion can be made for the comparison between a three-dimensional model and a uni-dimensional model

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(2) G. SCHELLINO; "Manuale d'uso per programma di simulazione e calcolo SISASP" C.R.F. (FIAT Research Centre)

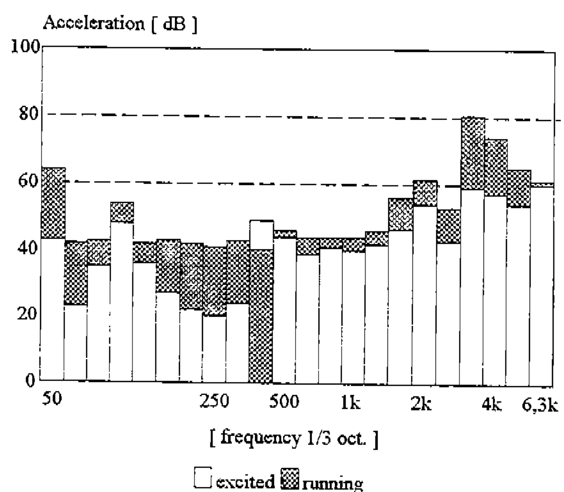

Fig.1 Artificially excitated compressor test

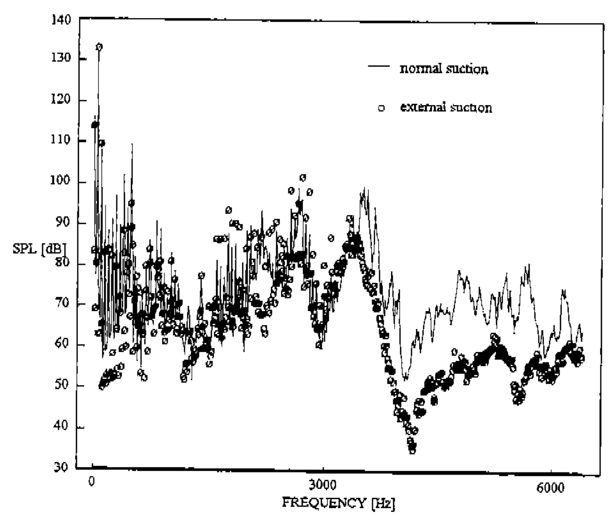

Fig. 2 SPL inside compressor gas cavity

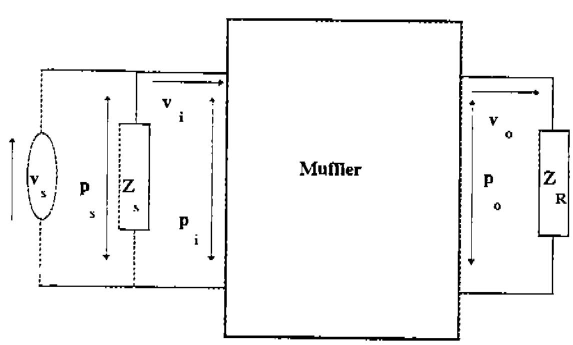

Fig. 3 Analitical model

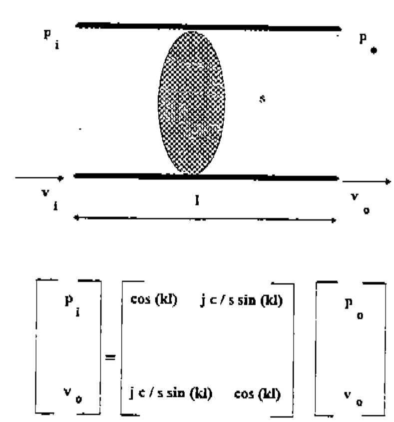

Fig. 4 Example of Transfer Matrix

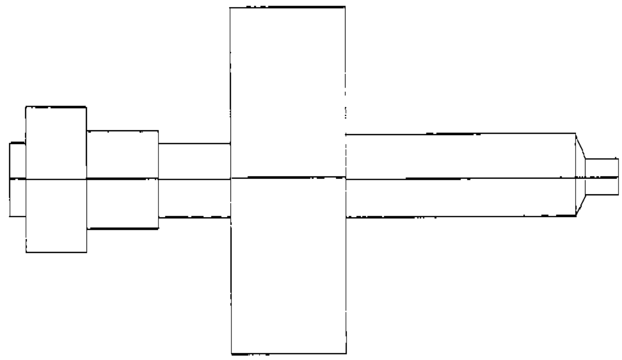

Fig. 5 Uni-dimensional suction system scheme

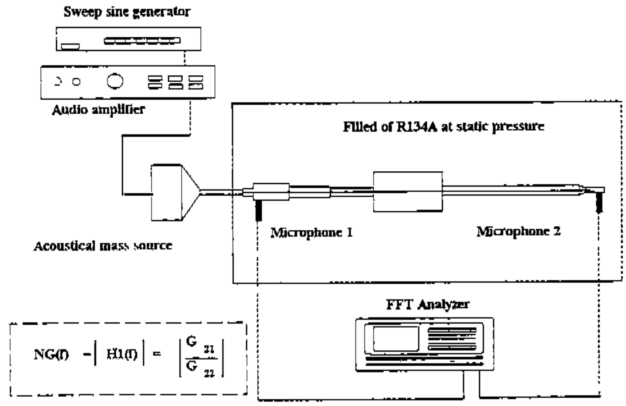

Fig. 6 Experimental Noise Gain determination

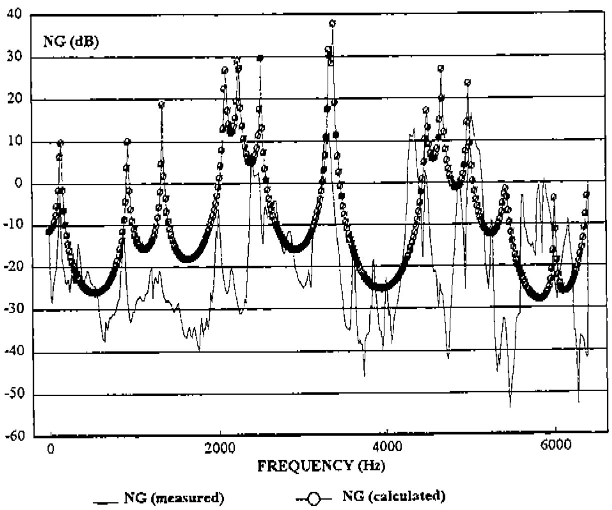

Fig. 7 Measured and calculated Noise Gain

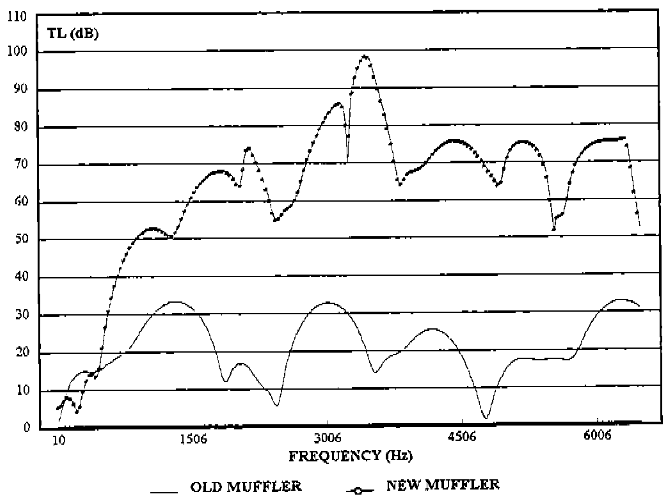

Fig. 8 T. L. of new and old muffler design

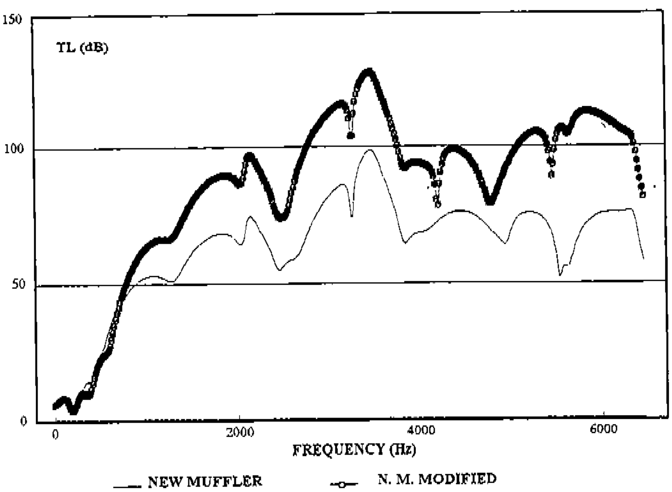

Fig. 9 T.L . of the new muffler with a hole-cavity resonator