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TUNED LOUDSPEAKER ENCLOSURES FOR ACTIVE NOISE CONTROL

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INTRODUCTION

The proper design of active control systems requires suitable electroacoustic transducers, and more particularly suitable loudspeakers. When designing active control systems the assumption is often made that these loudspeakers behave in a prescribed manner and are linear. In practice, nonlinearity can be a major problem and is one of the reasons why installed (as opposed to laboratory) systems can fail to provide an adequate amount of noise reduction. Moreover, the requirements for the loudspeakers used as secondary sources in active control systems are in general different from those needed for high quality music reproduction. For example, to control actively the blade passing tone in helicopters, it is necessary to drive high power levels (with low distortion) over only a narrow bandwidth. The best solution to achieve such requirements seems to be the use of vented-box loudspeakers. The aim of this paper is to investigate the behaviour of such narrow band enclosures, and to suggest ways in which the nonlinearities of these systems can be reduced.

DESIGN OF A LOW FREQUENCY, HIGH POWER VENTED-BOX SYSTEM

Beranek [1] suggested a simplified acoustical circuit of vented-box loudspeaker, valid for low frequencies (figure 1). In this circuit : e_g is the open circuit voltage of the generator (Volts); Bl is the force factor of the loudspeaker (N/A); R_c is the electrical resistance of the coil (Ω), measured with the voice-coil movement blocked; S_d is the effective area (m^2) of the loudspeaker diaphragm; M_{as} is the acoustic mass of the diaphragm and the voice coil (kg / m^4); R_{as} (N.s / m^5) and C_{as} (m^5 / N) are respectively the acoustical resistance and compliance of the suspension; C_{ab} is the acoustical compliance due to the air in the box; M_{as} is the sum of the acoustical mass of the diaphragm and the voice coil, of the acoustical radiation mass for the front side of the loudspeaker, and of the acoustic mass loading on the rear side of the diaphragm, due to the air in the box; M_{av}, the total air mass of the vent, is the sum of M_{ap}, the acoustic mass of air in the port (including the inner end correction) and of M_{a2} , the acoustical radiation mass for the outer end of the port; R_{ap} is the acoustical resistance of the air in the port. It is important to note that R_{ap} is generally negligible when the system is fed by small amplitude input signals; U_c, U_p and U_b are respectively the volume velocities of the diaphragm, of air in the port and air in the box (m^3/s). The circuit of figure 1 is equivalent to a two degrees of freedom system. Figure 2 shows the computed volume velocities versus frequency for both the driver, the box and the port of a vented system whose characteristics are : driver : Bl = 44 N/A; S_d = 0.053 m^2; mechanical resistance of suspension = 20 mech. Ohms; mechanical compliance of suspension = 1.6 10^{-4} m/N; box : volume = 0.548 m^3; vent : length = 0.586m; internal radius = 0.093 m. Note that volume velocities of the vent and the driver are out of phase below 14.5 Hz and in phase above this frequency. Figure 2 A. reveals an interesting mode of working of the system, that is achieved when the frequency of the exciting force is equal to the natural frequency of the enclosure sub-system (stiffness of air in the enclosure, and mass of air in the port). This mode is often called the Helmholtz mode and occurs at 14.5 Hz in this case. At this frequency the diaphragm does not vibrate at all (provided that both the acoustical resistance of the vent and the enclosure losses are negligible) and all the acoustic power is generated by the port. Tuning the system on the Helmholtz mode is therefore a good way to avoid driver nonlinearities, the only potential source of nonlinearities that remains being the vent (note that a phenomenon of thermal power compression is still likely to occur at the driver, although it will not be discussed in this paper). For the active control of blade passing tone in helicopters this tuning strategy can require the resonant frequency of the system to be as low as 14.5 Hz.

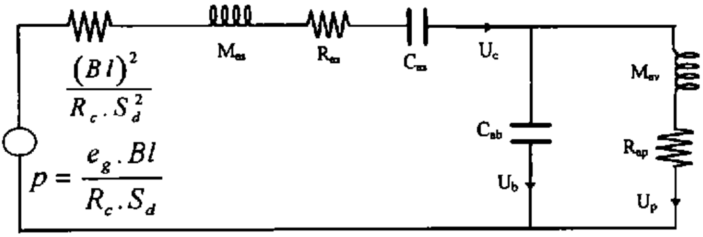


Fig. 1. *Simplified acoustical circuit of vented-box loudspeaker, valid for low frequencies.*

LARGE SIGNALS ANALYSIS OF THE VENTED LOUDSPEAKER

To understand the behaviour of the vented-box loudspeaker when driving

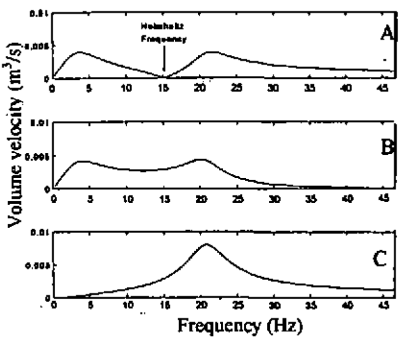
large volume velocities, a large signals analysis of the unit is necessary. When driving large amplitude signals, the port acoustic reactance and resistance are likely to be nonlinearly dependant on the port volume velocity and the acoustic resistance is no longer negligible in the circuit of figure 1. The acoustical characteristics of the vent can be calculated by measuring the volume velocity of the loudspeaker diaphragm,
which is now no longer equal to zero at the Helmholtz frequency. If Z_1, the impedance of the driver, Z_2, the impedance of the cabinet volume and Z_3, the impedance of the vent are defined as follow :

$$Z_1 = \frac{(Bl)^2}{R_e . S_d^2} + R_{as} + j\omega M_{as} + \frac{1}{j\omega C_{as}}, \quad Z_2 = \frac{1}{j\omega C_{ab}}, \quad Z_3 = R_{ap} + j\omega(M_{ap} + M_{a2}),$$

then the impedance of the vent can be written as :

$$\boxed{Z_3 = \left(p - Z_1 . U_c\right) . Z_2 / \left\{\left(Z_1 + Z_2\right) . U_c - p\right\}}$$

The real part of Z_3 gives the value of R_{ap}, while the imaginary part of Z_3 gives the value of $(M_{ap} + M_{a2})$. The values of R_{av} and M_{ap} were measured for various loudspeaker input currents at 14.5 Hz. The results are given in figures 3 and 4 where the non-dimentional port velocity is defined as : $U_p(non\,\mathrm{dim.}) = U_p / \left(S_v . \sqrt{\nu . \omega}\right)$, in which S_v is the vent area and n is the kinematic viscosity of air (1.56 10^{-5} m^2/s in normal conditions).



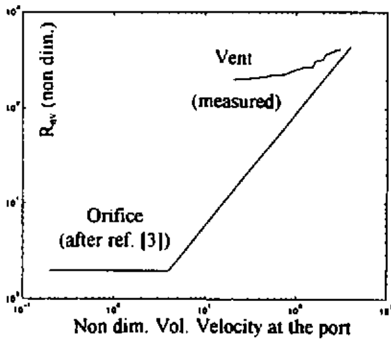


Fig. 2. *Volume velocity of the vented-box loudspeaker (A). Generated by the diaphragm (B). Generated by the port alone, and (C). Generated by the whole system.*

Fig. 3. *Comparison of the nonlinear acoustic resistance of the vent with the acoustic resistance of an orifice. Data Relative to the orifice are taken from ref [3].*

The acoustical resistance (figure 3) increases nonlinearly with the port velocity, because the flow in the vent becomes turbulent. Note however that this nonlinear effect is not very pronounced : an experimental study revealed that this nonlinear dependence of the vent acoustical resistance on the flow velocity in the vent has almost no influence on the loudspeaker output. The magnitude of the resistance is, however, considerably higher then that predicted for the orifice, presumably because of the finite length of the vent. Figure 4 shows that the acoustical reactance of the vent decreases slightly with the port velocity for the experimental enclosure. This result is consistent with the literature [3]. The main effect of this nonlinear dependence is to lead the loudspeaker tuning frequency to slightly increase with the loudspeaker input current, as illustrated on figure 5. The tuning frequency was equal to 14.4 Hz for an input current of 0.4 A (rms) while it was equal to 14.7 Hz for an input current of 2 A (rms). The total volume velocity produced by the enclosure at the Helmholtz frequency was about $0.1 m^3 s^{-1}$ at 1 A (rms), which is substantially more than could have been obtained from the loudspeaker in a closed enclosure at this frequency.

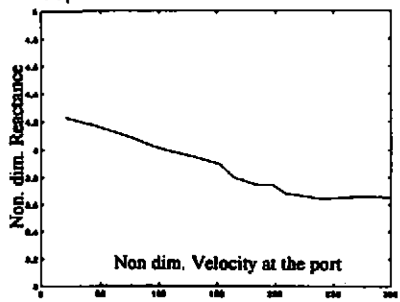


Fig. 4. *Acoustical reactance of the vent, in the nondimensional form suggested in [3].*

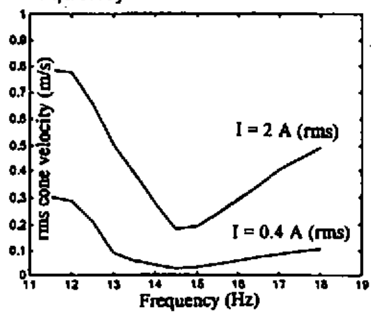


Fig. 5. *Diaphragm linear velocity Versus frequency for two input currents.*

CONCLUSION

Vented-box loudspeakers are good candidates as secondary sources in active noise control systems especially if the noise to cancel is tonal. They can derive relatively large volume velocities even at very low frequencies. The best way to reduce the nonlinear behaviour of the driver in a vented-box is to tune the loudspeaker on the Helmholtz frequency. In this case, the vent is the only part in the system that is likely to exhibit a nonlinear behaviour. The influence of the nonlinear acoustical resistance of the vent on the loudspeaker performance is almost negligible in the very large enclosure tested here, but may be significant for enclosures with smaller vents. However, the decrease of the vent acoustical reactance with the input level leads to an increase of the tuning frequency. This effect must be taken into account when designing the loudspeaker.

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