

# USE OF FEM/BEM TO IMPROVE HANDSFREE TELEPHONE AUDIO QUALITY

Stéphane Dedieu and Philippe Moquin

Mitel Networks, 350 Legget Drive, Kanata, Ontario, CANADA, K2K 2W7

## INTRODUCTION

The telephone audio band ranges from 300 to 3400 Hz for traditional telephony and 150 to 7000 Hz for wideband audio. In spite of such a wide bandwidth numerical methods offer a valuable design tool. Previous work has investigated main factors to be taken into consideration in the model [1]. This paper presents a case study on a telephone conference unit, and will show some interesting aspects of the design process required. First some theoretical basics of the full structure/acoustic problem are presented and commented. Secondly, the Finite and Boundary element model is presented. Results are analysed and practical solutions are brought to improve the acoustic receive performance.

## THEORETICAL BACKGROUND

The telephone can be considered as an elastic structure  $S$  enclosing a fluid cavity and radiating sound in the external domain. Let  $P_1$  and  $P_2$  be the pressure in the internal and external domain,  $f$  the force on the loudspeaker diaphragm, and  $U$  the normal diaphragm displacement field. To solve the full coupled vibro-acoustic problem a mixed variational (structure/internal fluid) and integral formulation (external fluid) is used. After discretization by finite and boundary elements, it leads to the following linear system [2]:

$$\begin{bmatrix} K - \omega^2 M + A & -C_1 & B + \frac{C_2}{2} \\ -C_1^t & \frac{H}{\rho_1 \omega^2} - \frac{Q}{\rho_1 c^2} & 0 \\ \left(B + \frac{C_2}{2}\right)^t & 0 & \frac{D}{\rho_2 \omega^2} \end{bmatrix} \begin{Bmatrix} U \\ P_1 \\ P_2 \end{Bmatrix} = \begin{Bmatrix} f \\ 0 \\ 0 \end{Bmatrix} \quad (1)$$

Where  $K$  and  $M$  are the structural stiffness and mass matrices,  $A$  the acoustic admittance matrix,  $C_1$  the structure/internal fluid coupling matrix,  $C_2$  and  $B$  the structure/external fluid coupling matrices and  $D$  the external fluid admittance.  $H$  and  $Q$  are the internal fluid matrices linked to the internal compression and kinetic energies.  $U$ ,  $P_1$ ,  $P_2$ ,  $f$  are nodal displacement, pressure and force field vectors. Using the cavity and the elastic structure eigen modes, and eliminating  $P_1$ ,  $P_2$ , the structure modal displacement field vector  $d$  is solution of the linear system:

$$[Z_m - Z_{ar} - C_r Z_f^{-1} C_r^t] \{d\} = \{g\} \quad (2)$$

where:  $Z_m = \Omega_s^2 - 2i\omega\eta_s\Omega_s - \omega^2 I$  is the modal mechanical

impedance matrix,  $Z_f = \frac{1}{\rho_1 \omega^2 c^2} [\Omega_f - 2i\omega\eta_f\Omega_f - \omega^2 I]$  and

$Z_{ar}$ , the modal acoustic impedance matrices for the internal and external fluid.

$\Omega_s$ ,  $\Omega_f$  are the natural angular frequencies matrices,  $I$  the identity matrix,  $C_r$  the modal structure/internal fluid coupling matrix and  $d$  and  $g$  the modal displacement field and force vectors.  $P_1$  and  $P_2$  can be determined depending on  $d$ .

Notice that the modal coupling matrix  $C_r$  terms do not depend on frequency but only on the geometry of the cavity and structure modeshapes. According to equation (2), when the forcing frequency is close to a cavity natural frequency, one term in  $Z_f^{-1}$  tends to infinity, so that the correspondent displacement vector  $d$  component must tend to zero. This means that close to a cavity resonance and depending on the shape of the coupling matrix  $C_r$ , the structure displacement field can be "blocked". This will likely happen when the coupling between an acoustic mode in the cavity and a structural mode is important, and may have an impact on the external frequency response.

## ACOUSTIC MODEL OF A CONFERENCE UNIT

The system is a telephone conference unit with a diameter of about 16 cm and a height of 6.5 cm. The loudspeaker (64 mm diameter) has its first resonance frequency  $f_0$  close to 260 Hz. The first step is to construct a simplified model where unnecessary details are eliminated. For example, since in normal telephony the maximum frequency is 3400 Hz, details much smaller than 10 cm are eliminated in the cavity as they are not relevant to the model. Also it has been previously shown [1] that from medium frequencies and above, the housing has little influence on the acoustic radiation compared to the loudspeaker diaphragm. The structure is meshed with shell elements, the acoustic cavity with volume finite elements and the external fluid with a layer of boundary elements. IDEAS Vibro-Acoustic with Rayon solvers were used for the simulation.

First cavity  
eigenfrequencies:

0 Hz  
1200 Hz  
2000 Hz  
2300 Hz

First structure  
eigenfrequencies:

210 Hz  
260 Hz (loudsp.)  
475 Hz  
640 Hz  
935 Hz (loudsp.)  
.....

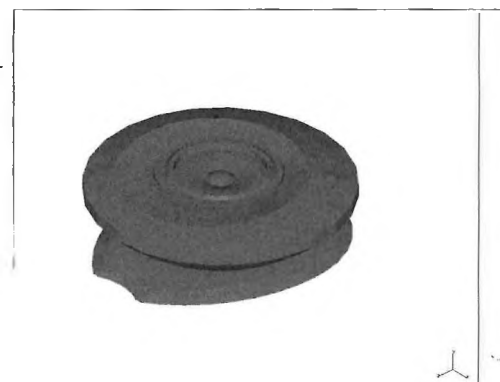


Figure 1: Description of the Conference Unit on its stand. Boundary Elements mesh. (IDEAS – Vibro-Acoustics)

#### Effect of acoustic modes on the loudspeaker response:

The loudspeaker's light diaphragm has a high mobility due to a low stiffness and relative low damping in this case (about 11 % for the 1<sup>st</sup> diaphragm mode  $f_0$ ). Therefore, more than the housing (2.7 mm thick plastic), the diaphragm is very sensitive to acoustic resonance in the cavity. After computation and as expected, recalling equation (2) above, we notice that close to cavity resonance, significant notches appeared in the sound pressure response at the receiving position (50 cm). The most critical notch appeared close to the third resonance, where the acoustic mode has an axi-symmetric shape with an antinode in the centre, at the position of the loudspeaker. As the coupling term between this cavity mode and the first diaphragm mode is important, the phenomenon described at the end of section 2 is very strong. This result was confirmed by measurement.

As manufacturing constraints prevent the use of porous material in the cavity, we decided to solve this problem by designing a cap to de-couple the loudspeaker from the housing cavity. A closed cap proved to stiffen the diaphragm, shifting up the first resonance. A cap with a leak was designed to properly "load" the diaphragm shifting down  $f_0$ .

#### Breathing mode:

The in-vacuum structure modal analysis exhibited a kind of housing "breathing" mode at about 1.2 kHz likely to be driven by the loudspeaker vibration. The initial design of the conference unit had only screws around the perimeter top and bottom parts. Despite the fact the housing vibration levels are much smaller than those of the diaphragm, we took no risks and decided to add a central post linking the cap and the bottom to prevent this kind of structural resonance, and avoid any buzzing noise.

## MEASUREMENTS

Once plastic parts were manufactured we were able to verify our modelling predictions. We performed a measurement of the conference unit receive characteristics. Figure 2 shows the receive frequency response at 50 cm, in a semi-anechoic condition as specified in ITU P.340. We can clearly see the notches at 1.5kHz, 2.0kHz and particularly at 2.2kHz illustrating the computational prediction. Also shown is the frequency response of the system with the cap and the post as we designed using the modelling results. The frequency response is clearly improved and meets the standard for speakerphones.

## CONCLUSIONS

Provided that the physics is mastered and given some simplifying assumptions, numerical modelling can be effectively used to model telephony type enclosures and provides reasonably good results even at medium frequencies and above. The major advantage is that many acoustical problems can be solved before any physical model is made. This saves money, as we do not have to modify plastic tools. It also saves significant design time, as the acoustic design is more likely to work in the first iteration as in this case.

## REFERENCES

- [1] Joris Brun-Berthet; *Utilisation de plans d'expérience pour choisir la modélisation acoustique d'un téléphone mains-libres*, Proceedings CAA 1999, pp 102-103.
- [2] Philippe Jean; *Une méthode variationnelle par équations intégrales pour la résolution de problèmes intérieurs et extérieurs de couplage élasto-acoustique*, Ph.D. Thesis, Université de Compiègne, France, 1985.

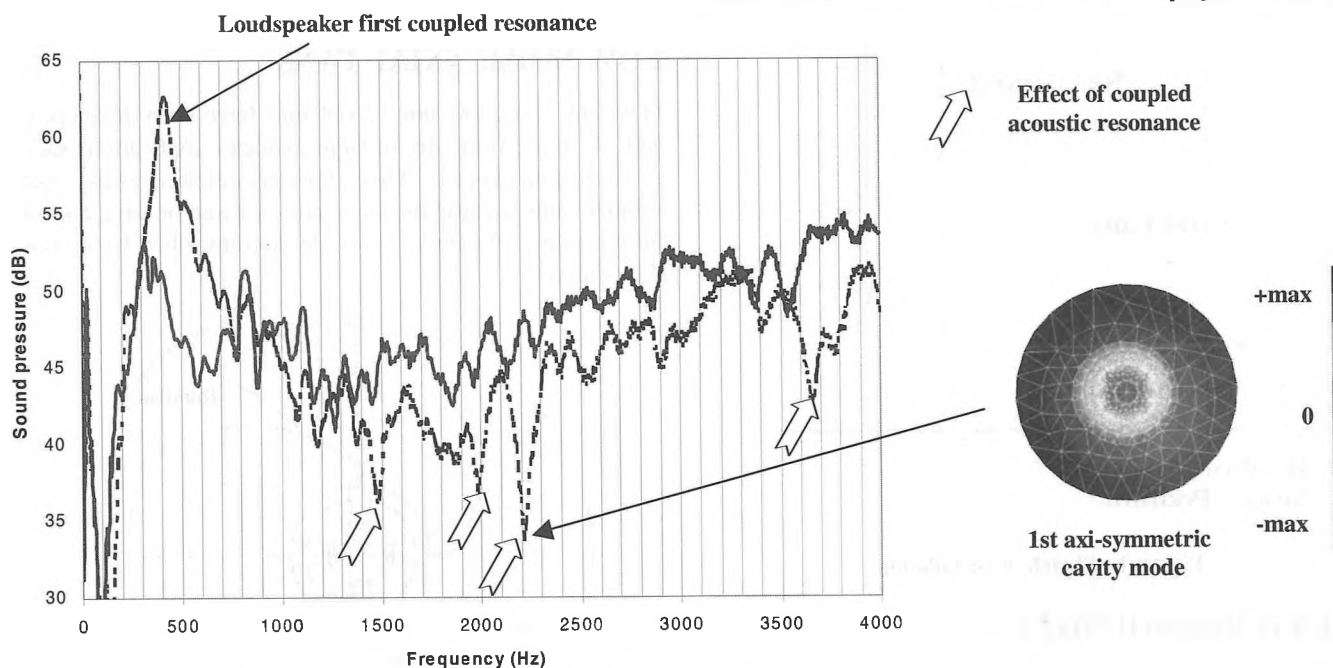


Figure 2: Measured receive response at 50 cm (Ear Reference Point-ITUP340). Initial response (dotted line) and improved response after use of a loudspeaker cap (solid line)