Proceedings of the Institute of Acoustics

NUMERICAL TECHNIQUES FOR SOLVING ACOUSTIC TRANSPARENCY PROBLEMS

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SUMMARY

This paper deals with numerical techniques for solving acoustic transparency problems. This kind of problem is encountered in the design of all major transportation systems, such as ground vehicles, aircraft and spacecraft: noise penetration through fairings, windows and doors, and break-out from engine compartments, *etc.* Applications in structural engineering such as the cladding of buildings can also be foreseen.
Acoustic transparency means the assessment of the transmissibility of sound by vibro-acoustic interactions, from one side of a (part of a) structure to the other. This is addressed using a hybrid numerical technique based on finite element (structure) and boundary element (fluid) formulations.
Particular attention is given to the handling of specific boundary conditions and/or operating conditions, such as are encountered in test procedures and working environments: notably, installation in an infinite rigid baffle, and also excitation by a diffuse or random acoustic field.
Numerical examples show verifications of the approach which is presented.

1. INTRODUCTION

The question of vibro-acoustic transparency appears in a wide range of problems. Examples include the transmission of noise through a car door or double-glazed window, and the acoustic energy applied to a satellite inside the fairing of a launch vehicle. An experimental analysis can be performed by placing a structure into the partition wall between a reverberant chamber, where a diffuse field is generated, and an anechoic chamber. The purpose is then to measure which part of the power is transmitted through the structure from one chamber to the other. However, such experimental tests are quite expensive, so for this reason, and in order to have more information on performance and any need for engineering design changes earlier in the design and development process, predictive methods are attractive.

This paper presents an innovative technology to perform predictive transparency analysis, using numerical modelling. The method is based on a baffled boundary element formulation which can handle structural geometries lying outside the plane of the baffle, as well as co-planar with it. Numerical results are compared to analytic results and results obtained by an Infinite-Finite Element Method (IFEM). Some special attention is given to the handling of diffuse field acoustic excitation and to the prediction of the transmitted power.

2. THEORETICAL BASIS

2.1 Description of the problem

A thin structure lies in an infinite homogeneous fluid domain V which is divided into two parts V_+ and V_- by an infinite plane baffle P_∞. By definition this plane is a plane where the acoustic normal velocity is equal to zero, *ie* it is an infinite rigid plane. The structure can lie inside or outside this baffle plane (*ie*, it need not be co-planar with the baffle) and the structure may also have holes through it.

2.2 Numerical problem statement

Find the acoustic pressure $p(x)$ that satisfies the Helmholtz equation: $(\Delta + k^2)p(x) = 0 \qquad \forall x \in V$

and the Sommerfeld condition at infinity: $\frac{\partial p(r)}{\partial r} + ikp(r) = o(\frac{1}{r}), \quad r \to \infty$

where r is a suitable radial distance.

The usual boundary conditions can be presented as a particular case of a generalized transfer relation:

$$\begin{pmatrix}\frac{\partial p}{\partial n}(x)\\ \frac{\partial p}{\partial n}(y)\end{pmatrix} = \begin{pmatrix}\alpha_{11}(x) & \alpha_{12}(x)\\ \alpha_{21}(x) & \alpha_{22}(x)\end{pmatrix}\begin{pmatrix}p(x)\\ p(y)\end{pmatrix} + \begin{pmatrix}-i\rho\omega v_n(x)\\ -i\rho\omega v_n(y)\end{pmatrix} \qquad \forall (x,y) \in S_t$$

(where y is the point corresponding to x on the other side of S).

Particular cases of this relation are velocity BC's (matrix α=0) or local admittance BC's $(\alpha_{12}(x) = \alpha_{21}(x) = 0)$:

For a known velocity: $\frac{\partial p(x)}{\partial n} = -i\rho\omega v_n(x) \qquad \forall x \in S_v$

For a known admittance: $\frac{\partial p(x)}{\partial n} = \alpha(x)p(x) - i\rho\omega v_n(x) \qquad \forall x \in S_a$

where k is the wavenumber $(k = \omega/c)$ ω is the circular frequency, c is the velocity of sound and ρ is the volumic mass of the fluid.

Openings in the baffle plane are defined by the following continuity conditions:

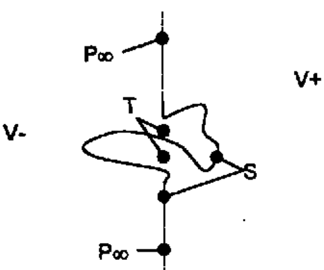
$$p(x) = p(y) \qquad \forall (x,y) \in T$$

$$\frac{\partial p}{\partial n}(x) = \frac{\partial p}{\partial n}(y) \qquad \forall (x,y) \in T$$

and finally, the BC's on the baffle itself are:

$$\frac{\partial p}{\partial n}(x) = 0 \qquad \forall x \in P_\infty \setminus (S \cup T)$$

with the notation of the figure below:



The physical properties of the materials along S (*ie* specific admittance $\alpha/(-ik)$ and transfer admittance matrix α_{ij} and the prescribed acoustic velocity v_n are assumed to be regular enough, so that there exists a solution to the formulated problem. In particular, physical considerations of reciprocity lead to the constraint $\alpha_{12} = -\alpha_{21}$ along S_T. Note that $v_n(x)$ is the structural normal velocity, which may or may not be known. If it is unknown, a coupled formulation has to be used.

As the structure S is thin, it is possible to work with its mean surface (also denoted by S in what follows). At each point of S, the two sides of S (variables or boundary conditions) are referred to by + and - signs and the normal at this point is assumed to point from the - to the + side.
We note that S is divided into 3 parts:

S^l, the part of S in V.

S^m, the part of S in the infinite plane

S^r, the part of S in V_+

2.3 Integral representation

The integral representation is built by using a Green function which satisfies:

$$\Delta G(X,Y) + k^2 G(X,Y) = -\delta(X,Y) - \delta(X',Y)$$

where X' is the symmetric of the point X.

Using Green's second formula on one half of the fluid domain, V_- or V_+ gives:

$$p(x) = \int_{S^m} \left\{ + \frac{\partial p^-(y)}{\partial n(y)}.G(x,y) \right\} dS(y) + \int_{S^l} \left\{ \mu(y).H^Y(x,y) - \sigma(y).G(x,y) \right\} dS(y)$$

$$+ \int_T \left\{ + \frac{\partial p(y)}{\partial n(y)}.G(x,y) \right\} dS(y) \qquad \forall x \in V_- \qquad (2.3.1)$$

and:

$$p(x) = \int_{S^m} \left\{ - \frac{\partial p^+(y)}{\partial n(y)}.G(x,y) \right\} dS(y) + \int_{S^l} \left\{ \mu(y).H^Y(x,y) - \sigma(y).G(x,y) \right\} dS(y)$$

$$+ \int_T \left\{ - \frac{\partial p(y)}{\partial n(y)}.G(x,y) \right\} dS(y) \qquad \forall x \in V_+ \qquad (2.3.2)$$

Where the double and single layer potentials are defined as the jump of pressure and the normal derivative of the jump of pressure, respectively:

$$\sigma(y) := \frac{\partial p^+(y)}{\partial n(y)} - \frac{\partial p^-(y)}{\partial n(y)} \quad \text{and:} \quad \mu(y) := p^+(y) - p^-(y)$$

Similar integral representations are derived for points on the 'transparent membrane' and the structure by using a limit process.

The mean values of the pressure and its normal derivative for points on S^r and S^l are expressed from operators[1] applied on μ, σ and $\frac{\partial p}{\partial n}$:

$$\overline{p}(x) := \frac{p^+(x) + p^-(x)}{2} = \overline{P}^l(\sigma, \mu, \frac{\partial p}{\partial n})(x)$$

and:

$$\overline{p}_n(x) := \frac{\frac{\partial p^+(x)}{\partial n} + \frac{\partial p^-(x)}{\partial n}}{2} = \overline{P}_n^l(\sigma, \mu, \frac{\partial p}{\partial n})(x)$$

where:

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[1] *eg*: $\overline{P} : (S \to C \times S \to C) \to (S \to C) : (\sigma, \mu) \rightleftharpoons \overline{P}(\sigma, \mu)$
with: $\overline{P}(\sigma, \mu) : S \to C : x \rightleftharpoons \overline{P}(\sigma, \mu)(x)$

$$\overline{P}^{l}(\sigma,\mu,\frac{\partial p}{\partial n})(x) := CPV\left\langle \mu(y).H^{Y}(x,y)\right\rangle_{S'(y)} - \langle\sigma(y).G(x,y)\rangle_{S'(y)}$$
$$+\left\langle \frac{\partial p^{-}(y)}{\partial n(y)}.G(x,y)\right\rangle_{S^{m}} + \left\langle \frac{\partial p(y)}{\partial n(y)}.G(x,y)\right\rangle_{T}$$

and:

$$\overline{P}^{l}_{n}(\sigma,\mu,\frac{\partial p}{\partial n})(x) := FP\langle\mu(y).E(x,y)\rangle_{S'(y)} - CPV\left\langle \sigma(y).H^{X}(x,y)\right\rangle_{S'(y)}$$
$$+\left\langle \frac{\partial p^{-}(y)}{\partial n(y)}.H^{X}(x,y)\right\rangle_{S^{m}} + \left\langle \frac{\partial p(y)}{\partial n(y)}.H^{X}(x,y)\right\rangle_{T} \qquad \text{(almost everywhere on } S^{l}\text{)}$$

Notations: CPV means the Cauchy Principal Value and FP means the Finite Part in Hadamard's sense.[2]
A similar integral equation is valid for points on S^{r}.
For the points on T or S^{m}, the representations are different if the limit process is taken from one side or from the other. For instance:

$$p^{-}(x) = P^{-}(\sigma,\mu,\frac{\partial p}{\partial n})(x)$$

where: $$P^{-}(\sigma,\mu,\frac{\partial p}{\partial n})(x) := \left\langle\left\{+\frac{\partial p^{-}(y)}{\partial n(y)}.G(x,y)\right\}\right\rangle_{S^{m}(y)}$$
$$+\left\langle\left\{\mu(y).H^{Y}(x,y) - \sigma(y).G(x,y)\right\}\right\rangle_{S'(y)} \qquad \text{for all points } x \text{ on } S^{m} \cup T.$$

The problem can then be expressed in terms of the surface potentials μ and σ on $S^{r} \cup S^{l}$, the normal derivative (normal to P_{∞}) of the pressure $\frac{\partial p}{\partial n}$ on T and in terms of the normal derivative of the pressure on both sides of S^{m}, $\frac{\partial p^{+}}{\partial n}$ and $\frac{\partial p^{-}}{\partial n}$. (If these surface values are known, the acoustic pressure field is uniquely given by the above integrals).

2.4 Surface problem

Some surface values are known a priori from the boundary conditions, as listed in Table 1:

Sub-surface	type of bc on + side	type of bc on - side	unknown local values			
S^{l}_{2} or S^{r}_{2}	relation $\tilde{\alpha}(x) \neq 0$		σ	μ		
S^{l}_{5} or S^{r}_{5}	relation $\tilde{\alpha}(x) = 0$			μ		
S^{m}_{6}	relation $rank[\alpha(x)] = 0$		No unknown			
S^{m}_{8}	relation $rank[\alpha(x)] = 2$				$\frac{\partial p^{+}}{\partial n}$	$\frac{\partial p^{-}}{\partial n}$
T	Continuity				$\frac{\partial p}{\partial n}$	

Table 1: Relations between sub-surfaces, boundary conditions and unknown potentials.

[2] $\langle f(x,y)\rangle_{S(y)} := \int_S f(x,y).dS(y)$ and $\langle f(x,y)\rangle_{S(x)S(y)} := \int_S\int_S f(x,y).dS(y).dS(x)$

The problem is to find μ defined on S_μ (= $S_2 \cup S_5$); σ defined on S_σ (= S_2); $\frac{\partial p}{\partial n}$ on T, $\frac{\partial p^+}{\partial n}$ on $S^m{}_8$ and $\frac{\partial p^-}{\partial n}$ on $S^m{}_8$ such that they verify the following integral equations:

$$\left(\overline{P_n}^s(\sigma,\mu,\frac{\partial p}{\partial n}) - \frac{\sigma\,\overline{\alpha}}{2\tilde{\alpha}} + \frac{\mu}{2}\left(\frac{\overline{\alpha}.\overline{\alpha}}{\tilde{\alpha}} - B\right) + \frac{\left(H^+ - H^-\right)\overline{\alpha}}{2\tilde{\alpha}} - \frac{H^+ + H^-}{2}\right)(x) = 0 \qquad \forall x \in S_2^s$$

$$\left(\overline{P}^s(\sigma,\mu,\frac{\partial p}{\partial n}) - \frac{\sigma}{2\tilde{\alpha}} + \frac{\mu}{2}\frac{\overline{\alpha}}{\tilde{\alpha}} + \frac{\left(H^+ - H^-\right)}{2\tilde{\alpha}}\right)(x) = 0 \qquad \forall x \in S_2^s$$

$$\overline{P_n}^s(\sigma,\mu,\frac{\partial p}{\partial n})(x) - \overline{P}^s(\sigma,\mu,\frac{\partial p}{\partial n})(x).\overline{\alpha}(x) - \frac{\mu(x)}{2}.B(x) - (\frac{H^+ + H^-}{2})(x) = 0 \quad \forall x \in S_5^s$$

$$P^+(\sigma,\mu,\frac{\partial p}{\partial n})(x) = P^-(\sigma,\mu,\frac{\partial p}{\partial n})(x) \qquad \forall x \in T$$

$$P^+(\sigma,\mu,\frac{\partial p}{\partial n})(x) = \frac{\alpha_{22}(x)}{|\alpha(x)|}\left[\frac{\partial p^+}{\partial n}(x) + i\rho\varpi v_n^+(x)\right] - \frac{\alpha_{21}(x)}{|\alpha(x)|}\left[\frac{\partial p^-}{\partial n}(x) + i\rho\varpi v_n^-(x)\right] \qquad \forall x \in S_8$$

$$P^-(\sigma,\mu,\frac{\partial p}{\partial n})(x) = -\frac{\alpha_{12}(x)}{|\alpha(x)|}\left[\frac{\partial p^+}{\partial n}(x) + i\rho\varpi v_n^+(x)\right] + \frac{\alpha_{22}(x)}{|\alpha(x)|}\left[\frac{\partial p^-}{\partial n}(x) + i\rho\varpi v_n^-(x)\right] \qquad \forall x \in S_8$$

In these equations, the fact that some surface variables are known on certain sub-surfaces is taken into account in the definition of the operators $\overline{P}^s$ and $\overline{P_n}^s$.
Note that if the structure S is only made of $S^m{}_8$ the integrals (2.3.1) and (2.3.2) reduce to the Rayleigh integral.

2.5 Variational form and stationary problem - Finite Element formulation

A symmetric variational form is obtained by multiplying the above equations by μ or σ. (The symmetry gives benefits in calculation time and storage). A stationarity principle is applied to obtain a minimization problem. The minimization problem is solved by the finite element method, which gives a solution in the space of functions defined by the shape functions on the surface approximating S U T.

The surface (structure) is modelled by surface finite elements. The topology of each element (noted e) is classically described by a transformation from a reference element, e^{ref} in R^3 to the real element in R^3:

$$\tau: S^e_{ref} \rightarrow R^3, (\varsigma,\eta) \Rightarrow \overline{x}^e(\varsigma,\eta) := \begin{pmatrix} x \\ y \\ z \end{pmatrix}^e (\varsigma,\eta) = \sum_{i=1,N^e} \begin{pmatrix} x_i \\ y_i \\ z_i \end{pmatrix}^e . M_i^e(\varsigma,\eta)$$

where $S^e_{ref} \in R^2$ is the domain of e^{ref}, N^e is the number of points on e (and e^{ref}), $\begin{pmatrix} x_i \\ y_i \\ z_i \end{pmatrix}^e$ is the image of the i^{th} point (ζ_i, η_i) of e^{ref} by τ, and $M^e{}_i$ is the i^{th} geometric shape function of e (defined by $M^e{}_i(\zeta_j, \eta_j) = \delta_{ij}$).
The unknown functions μ, σ and $\frac{\partial p}{\partial n}$ are expressed in term of their nodal values:

$$\sigma(\overline{x}) := \sum_{i \in E_e} \sigma_i . N_i(\varsigma, \eta)_i$$

$$\mu(\overline{x}) := \sum_{i \in E_e} \mu_i . N_i(\varsigma, \eta)_i$$

$$\frac{\partial p}{\partial n}(\overline{x}) := \sum_{i \in E_e} q_i . N_i(\varsigma, \eta)_i$$

Finally, the (n_u x n_u) matrix system to be solved (where n_u is the number of unknowns) is:

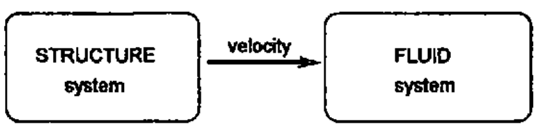
$$(A_{i,j})_{1\le i,j\le n_u} \bullet (u_i)^T_{1\le i\le n_u} = (F_i)^T_{1\le i\le n_u}.$$

Note that the left and right parts are decoupled. If the unknowns on S^m U T, S^l and S^r are denoted by $(u^m_i)^T$ $(u^l_i)^T$ and $(u^r_i)^T$ respectively, then:

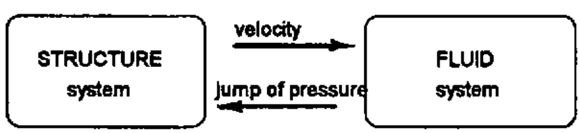
$$\begin{pmatrix} A^{mm} & A^{mr} & A^{ml} \\ A^{rm} & A^{rr} & 0 \\ A^{lm} & 0 & A^{ll} \end{pmatrix} \begin{pmatrix} u^m \\ u^r \\ u^l \end{pmatrix} = \begin{pmatrix} F^m \\ F^r \\ F^l \end{pmatrix} \begin{matrix} \Rightarrow S^m \\ \Rightarrow S^r \\ \Rightarrow S^l \end{matrix}$$

2.6 Coupled formulation and solution scheme

In the above formulation, there are no details about the velocity v_n. These values can be independent of the acoustic pressure field, if the load (added mass and/or stiffness) coming from the fluid onto the structure is small. This gives a so-called one-way coupling, with the velocity considered as a given boundary condition of the acoustic problem:



If the load from the fluid onto the structure is not negligible, the load surface density, comprising the jump of pressure at the right of the structure must be taken into account in the equations for structural equilibrium. The system is then fully-coupled:



The structural problem is solved by a variational statement. In this structural formulation, the influence of the fluid in the energy equation is the pressure-load energy:

$$\langle \mu(y).u_n(y) \rangle_{S^c(y)}$$

where S^c is the coupled part of the structure and u_n is the structural displacement ($i\omega u_n = v_n$).
After discretizing the structure by finite elements, the following symmetric system has to be solved:

$$\begin{pmatrix} Z+Z^{up} & C^m & C^r & C^l \\ (C^m)^T & A^{mm} & A^{mr} & A^{ml} \\ (C^r)^T & A^{rm} & A^{rr} & 0 \\ (C^l)^T & A^{lm} & 0 & A^{ll} \end{pmatrix} \begin{pmatrix} d \\ u^m \\ u^r \\ u^l \end{pmatrix} = \begin{pmatrix} F^s + F^{up} \\ F^m \\ F^r \\ F^l \end{pmatrix} \begin{matrix} \\ \Rightarrow S^m \\ \Rightarrow S^r \\ \Rightarrow S^l \end{matrix}$$

where d denotes the nodal structural displacements.
(Z).(d) = (F^s) is the system resulting from the corresponding uncoupled structural problem.

2.7 Handling of an incident field

A classical handling of the incident field is as follows:
The scattered field p_s is defined by the relation: $p_{total} = p_i + p_s$ where p_i is the incident field; and (by linearity) the boundary condition is then formulated in terms of p_s, *eg*:

$$\begin{pmatrix} \frac{\partial p_s}{\partial n}(x) \\ \frac{\partial p_s}{\partial n}(y) \end{pmatrix} = \begin{pmatrix} \alpha_{11}(x) & \alpha_{12}(x) \\ \alpha_{21}(x) & \alpha_{22}(x) \end{pmatrix} \begin{pmatrix} p_s(x) \\ p_s(y) \end{pmatrix} + \begin{pmatrix} -i\rho\varpi v_n(x) \\ -i\rho\varpi v_n(y) \end{pmatrix}$$

$$- \begin{pmatrix} \frac{\partial p_i}{\partial n}(x) \\ \frac{\partial p_i}{\partial n}(y) \end{pmatrix} + \begin{pmatrix} \alpha_{11}(x) & \alpha_{12}(x) \\ \alpha_{21}(x) & \alpha_{22}(x) \end{pmatrix} \begin{pmatrix} p_i(x) \\ p_i(y) \end{pmatrix}$$

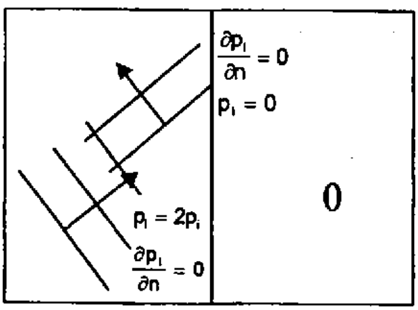
Note: This approach is quite classical, but special care must be taken, because the expression of the BC's in terms of scattered pressure gives rise to non-zero velocity BC's in the baffle plane:

$$\frac{\partial p}{\partial n} = -\frac{\partial p_i}{\partial n} \quad \text{at each point of } P_\infty\ (S^m \cup T)$$

In order to avoid integration in this infinite plane, the incident field is re-defined as an incident field such that:

$$\frac{\partial p_i}{\partial n} = 0 \quad \text{at each point of } P_\infty$$

This new definition is fully-consistent with the initial definition of the field, as shown in the following figure:



2.8 Random incident field

A particular case for the incident field is the random acoustic diffuse field. Two approaches can be used: the superposition of an infinite set of uncorrelated plane waves, or (more efficiently) an automatic extraction of principal components of the excitation [2].

2.9 Software implementation

The methodology presented here has been embodied in software, which will be added to the next revision of a general-purpose FE/BE program for vibro-acoustics [4] to use its extensive pre- and post-processing and other capabilities. A link to specialized software with poro-visco-elastic fluid-structure elements [5] will enable the analysis of multi-layer structures including porous absorbers and localized models of the damping layers themselves, with assessment of energy flows and dissipation efficiencies.

3. APPLICATIONS

3.1 Baffled plate subject to a nodal load

A simple plate example is shown to illustrate the procedure for the analysis and the types of results which are derived, and as a verification against other methods.

Plate dimensions: 1m x 1m x 0.005m
Young's modulus: 2.1E11 N/m^2
Density, ρ: 7800 kg/m^3
Modal damping, η: 0.01
Plate simply-supported in the baffle, *ie* normal displacement equal to 0 at the edges.
64 structural modes are used
Acoustic fluid: Air on both sides of the plate
Speed of sound, c: 340 m/s
Density, ρ: 1.225 kg/m^3
Normal nodal load: 1000 N amplitude, at node at (0.65,0.2,0) *ie* not at centre (baffle is in plane Z=0).

Analysis from 10 to 500 Hz in steps of 5 Hz:
The 64 structural modes are first calculated, without any fluid influence (see Figure 1; note that only the flexible structural elements, which are identical to the boundary elements, are shown: the surrounding infinite baffle is not displayed). These modes are then used in the fully-coupled calculation.
The results obtained from the Boundary Element formulation presented above, are compared with the results of an Infinite Element formulation (IFEM) [3] which may be taken as a reference result, and with the displacement results for the plate without any fluid influence.
Figure 2 shows pressure *vs* frequency, at the field point at (0.5,0.5,0.5) *ie* above the centre of the plate. Figures 3 and 4 show the UZ displacement (normal to plate) at the nodes at (0.2,0.3,0) and (0.5,0.5,0) *ie* in one quadrant and at the centre of the plate. Figure 5 shows a deformed-shape plot of the displacements of the plate at 255 Hz, and Figure 6 shows the acoustic pressure on a hemisphere with 1 metre radius, centred on the plate, in the 'receiver room'. The transmitted power (active, Real part, *ie* propagating energy) is 0.5336 watts at 255 Hz.
It can be seen that there is very good agreement between BEM and IFEM.

3.2 Baffled plate subject to an acoustic plane wave

Same plate as in 3.1, but excitation is an acoustic incident plane wave, with unit amplitude.
The results obtained from the BEM formulation are again compared with the results of an IFEM formulation and with the results for the plate without any fluid influence, together with the results of an analytical solution, which does not take into account the cross-influence between structural modes due to the linkage with the fluid.
Figure 7 shows the pressures at the field point at (0.5,0.5,0.5) and Figures 8 and 9 show the UZ displacement (normal to plate) at the nodes at (0.2,0.3,0) and (0.5,0.5,0).
It can be seen that there is very good agreement between BEM and IFEM.

4. CONCLUSIONS

The methodology presented here is very promising due to its generality and ease-of-use. It supports a wide range of boundary conditions including generalized transfer relations and holes in the structure. Random or diffuse-field excitations are supported. The acoustic system can be coupled to a part (or the whole) of a structure, which is represented in physical or modal coordinates. This structure can lie within or outside the plane of the baffle. The implementation of this methodology within a general purpose vibro-acoustic program, and a link to special software for multi-layered vibro-acoustic damping materials, creates a complete design analysis procedure for acoustic transparency studies.

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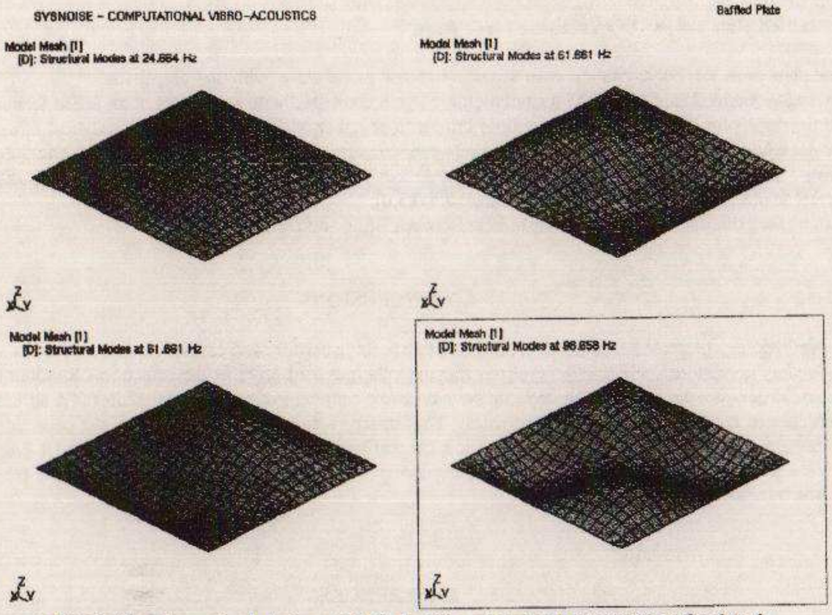

FIGURE 1: Example plate in an infinite baffle, uncoupled structural modes 1 to 4

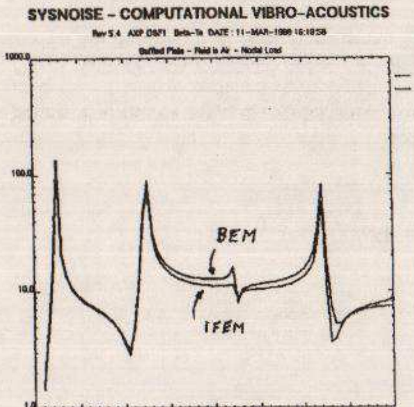

FIGURE 2: Plate with normal force, field point pressures *vs* frequency, BEM and IFEM compared

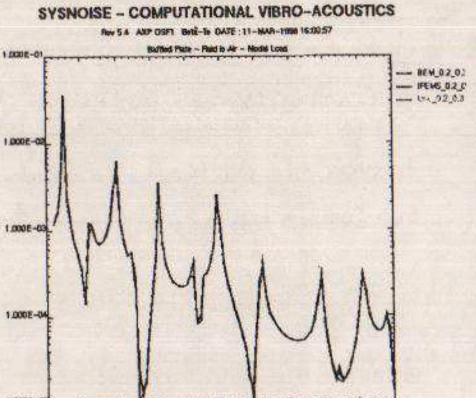

FIGURE 3: Plate with normal force, normal displacements at (0.2,0.3,0) *vs* frequency, BEM, IFEM and Uncoupled compared

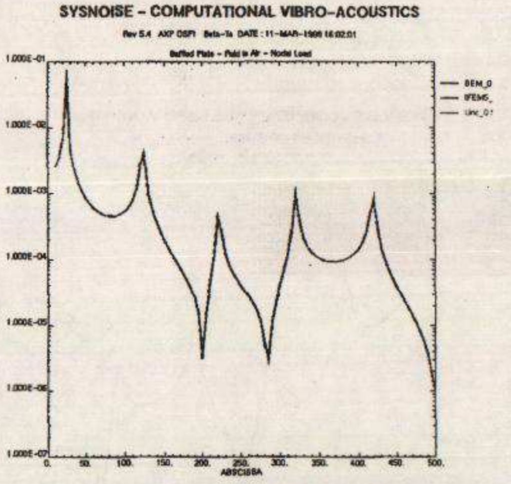

FIGURE 4: Plate with normal force, normal displacements at (0.5,0.5,0) *vs* frequency, BEM, IFEM and Uncoupled compared

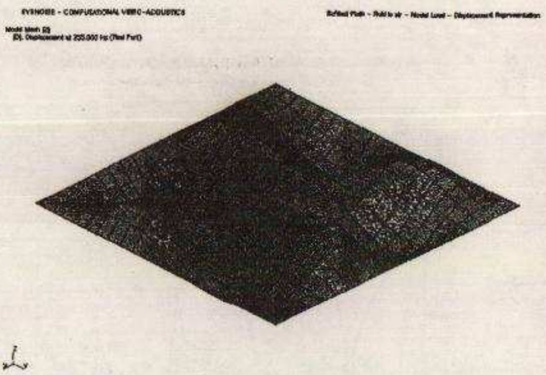

FIGURE 5: Plate with normal force, deformed-shape plot of displacements at 255Hz

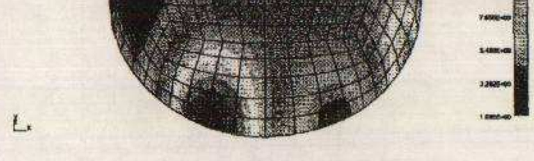

FIGURE 6: Plate with normal force, contour plot of pressure in the 'Receiver' Halfspace, 255Hz (Viewpoint: in the receiver room, baffle lower edge nearest to viewpoint)

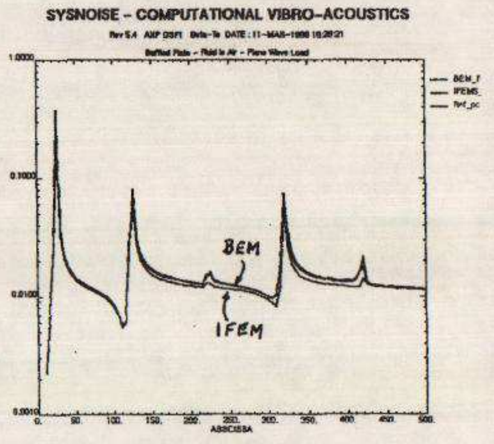

FIGURE 7: Plate with incident plane wave, field point pressures *vs* frequency, BEM and IFEM compared

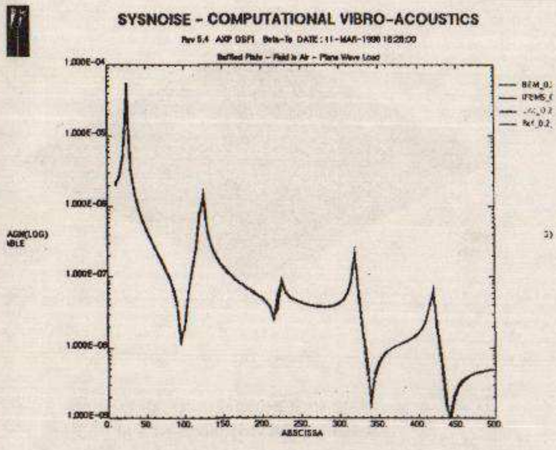

FIGURE 8: Plate with incident plane wave, normal displacements at (0.2,0.3,0) *vs* frequency: BEM, IFEM, Uncoupled and Analytical compared

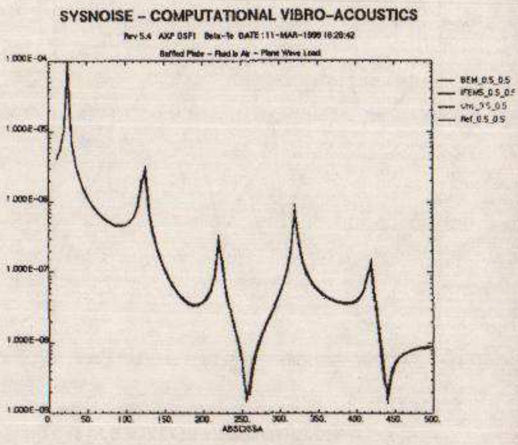

FIGURE 9: Plate with incident plane wave, normal displacements at (0.5,0.5,0) *vs* frequency: BEM, IFEM, Uncoupled and Analytical compared