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Structural Acoustic Calculations in the Low-Frequency Range

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The structural acoustic research activities pursued at the Aircraft Design Institute in collaboration with the Italian Aerospace Research Center during the last 5 yr are collected and summarized in this article. The fluid-structural interaction is approached from several viewpoints, paying attention to the theoretical analysis of the problem, as well as to its practical and realistic applications. The research results are presented with a view to outline both the future necessary developments of the theoretical approach, mainly looking at the possible extension of the deterministic approach, and the actual numerical-experimental comparisons that have been completed for gaining a necessary confidence level of the methodologies when facing real design problems.

Nomenclature

A = cross-sectional area of the pipe
$( )_A$ = acoustic term
A_C = coupling area, fluid-structural interface
A_F = part of the boundary of the integration domain where the acoustic impedance is specified
AN = nodal area matrix
B = general damping matrix
$B_{i,j}$ = dimensional coupling factor
c = speed of sound of the fluid
D = stress-strain relationship
$( )_D$ = discrete unknowns
E = Young modulus
ec = length of the expansion chamber
$\boldsymbol{F}$ = general loading vector
$F_{i,j}$ = nondimensional coupling factor
f = general modal loading vector
$(\dot{g})$ = time derivative of g
ip = length of the inlet pipe
K = general stiffness matrix
k = general modal stiffness matrix
L = total length of the muffler
M = general mass matrix
Mach = Mach number
m = general modal mass matrix
$\boldsymbol{n}$ = normal vector
p = unknown acoustic pressure
R_E = dynamic resistance
$( )_S$ = structural term
$( )^T$ = transpose operator
t = time
U = stream velocity
u = x component of the displacement vector in Oxyzt
$\boldsymbol{u}$ = unknown displacement vector
v = y component of the displacement vector in Oxyzt
w = z component of the displacement vector in Oxyzt
Z_n = specific normal acoustic impedance
∇^2 = Laplacian operator
β = $\sqrt{1 - \text{Mach}^2}$
ε = strain
ρ = mass density of the fluid
ρ_E = effective density of the fluid inside porous material
σ = stress
v = Poisson modulus
Φ = general modal coupled area matrix
ϕ_{iS} = ith structural eigenvector
ϕ_{jA} = jth acoustic eigenvector
Ω_S = surface porosity
Ω_V = volume porosity
ω = excitation frequency
ω_{iS} = ith structural eigenvalue
ω_{jA} = jth acoustic eigenvalue

I. Introduction

THIS article will collect and show several experiences in the structural acoustics field where a large amount of theoretical and numerical analysis, mainly finite element based, were performed in the last 5 yr. An attempt will be made to summarize the main steps of the activities, some of the obtained results, and to underline the problems evidenced during the activities. The research was driven by the need to evaluate and reduce the interior noise levels inside the fuselage of commercial jet and propeller aircraft[1–8] and advanced propeller aircraft.[9] The achievement of an augmented level of interior comfort was a real challenge for the whole aerospace scientific community, together with growing attention for fluid-structure interaction understanding and modeling. The large amount of papers published in the last decade[1–28] attests to this interest.

In this article more attention will be devoted to the general methodology and experience developed around the analysis of the fluid-structure interaction. The latter refers to the low-frequency range problems, in which modal representations could be suitable. In fact, the term "low frequency" more appropriately should be replaced by the term low modal density, and clearly it could be applied independent of the frequency range. The applications herein shown, are therefore related to low modal density ones. Practical issues like the computational costs, the effectiveness of the simulation of the dissipative effects, and the relation between the skill of the user and the required skill to properly follow the solution of a general structural acoustic problem, are also discussed.

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II. Governing Equations and Finite Element Approach for the Fluid-Structure Analysis

The governing equations of motions including the structural dynamics, the acoustics, and their coupling, are recalled. For a continuous solid, the classical Navier elasticity relationships, without dissipative terms are[10,24]

$$\partial D \partial^T u - \rho_s \frac{\partial^2 u}{\partial t^2} = F \tag{1}$$

where ρ_s is the mass structural density. The D matrix is the classical Hooke stress-strain relationship that for a plane stress case is the following:

$$D = \frac{E}{(1+v)(1-2v)} \begin{bmatrix} 1-v & v & 0 \\ v & 1-v & 0 \\ 0 & 0 & \dfrac{1-2v}{2} \end{bmatrix}$$

$$\partial = \begin{vmatrix} \dfrac{\partial}{\partial x} & 0 & 0 & \dfrac{\partial}{\partial y} & \dfrac{\partial}{\partial z} & 0 \\ 0 & \dfrac{\partial}{\partial y} & 0 & \dfrac{\partial}{\partial x} & 0 & \dfrac{\partial}{\partial z} \\ 0 & 0 & \dfrac{\partial}{\partial z} & 0 & \dfrac{\partial}{\partial x} & \dfrac{\partial}{\partial y} \end{vmatrix}$$

$$\boldsymbol{u} = [u(x, y, z), v(x, y, z), w(x, y, z)]^T$$

$$\boldsymbol{F} = [f_x, f_y, f_z]^T$$

Note that ∂ is a differential operator and $\boldsymbol{F}$ is the external force vector (including the inertial ones). The forces acting on the external surface of the domain F_s provide the boundary conditions for the problem:

$$D\partial^T u = F_s$$

The constrained displacements, if present, provide the other boundary conditions (Dirichlet conditions).

The Helmholtz equation, for the acoustic pressure dynamic behavior is:

$$\nabla^2 p = \frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} \tag{2}$$

The coupling conditions at the interface (fluid-vibrating structure) are the following:

$$\frac{\partial p}{\partial n} = \rho \frac{\partial^2 u_n}{\partial t^2} \tag{3}$$

$$F_s = pA_c \tag{4}$$

The second effect is called fluid loading. It is possible to show that the fluid-structure interaction equations, Eqs. (1) and (2), jointly with the coupling conditions, Eqs. (3) and (4), in matrix finite element form,[24] are the following:

$$\begin{aligned} M_s \ddot{u}_D + K_s u_D &= -(AN)p_D + F_s \\ M_A \ddot{p}_D + K_A p_D &= \rho (AN)^T \ddot{u}_D + F_A \end{aligned} \tag{5}$$

Including linear dissipation effects, one gets the following general expression:

$$\begin{aligned} M_s \ddot{u}_D + B_s \dot{u}_D + K_s u_D &= -(AN)p_D + F_{sD} \\ M_A \ddot{p}_D + B_A \dot{p}_D + K_A p_D &= \rho (AN)^T \ddot{u}_D + F_{AD} \end{aligned} \tag{6}$$

where M, B and K are assembled using the finite element method (FEM). In Ref. 23, a refined analytical approach, based on the variational principle, is also presented. At this level the B_A matrix should include only the contribution of the damping of the fluid, neglecting the dissipative surface effects. It should be noted that if the fluid loading effect is neglected, the acoustic field can be solved by using the boundary element method,[17,29,30]

The most important aspect of FEM problems centers around the evaluation of the structural damping, particularly in the linearized approach. In fact, in the finite element analysis the real behavior of the system is strongly governed by the internal/external connections modeling the dissipative effects. By using experimental measurements it is possible to better tune the numerical parameter and to refine the computational model. The analysis of an enhanced damping model,[31] together with the studies of nonlinear behavior of the constraints are needed, because the FEM still remains among the more efficient and reliable computational tools for the low-frequency range, postdesign applications.

III. Soundproofing Problem

For the acoustic field, it is possible to theoretically model the effect of a dissipative surface posed on a boundary of the acoustic domain. The boundary can be also a vibrating wall. An analysis of the literature about the topics[10,12,13,20,21] shows that is possible to use the following approaches.

A. Field Approach

In this approach the acoustic field inside the soundproofing layer is calculated. A proposed governing equation for the acoustic behavior of the material (neglecting its elastic contribution) is the following:

$$\nabla^2 p = \frac{R_E \Omega_V^2 \rho_E}{c^2 \Omega_s^2 \rho}\frac{\partial p}{\partial t} + \frac{R_E \Omega_v}{c^2 \Omega_s \rho}\frac{\partial^2 p}{\partial t^2} \tag{7}$$

where c is the speed of sound in the fluid.

It is emphasized that ρ_e and R_E are both frequency dependent. The pressure and its gradient must be continuous at the interface with the acoustic field, Eq. (2), and again, Eqs. (3) and (4) define the coupling at the interface with the structure.

B. Impedance Approach

The second approach to simulate the soundproofing materials is to generalize the boundary conditions, Eq. (3), as follows:

$$\frac{\partial p}{\partial n} = \frac{\rho}{Z_n}\frac{\partial p}{\partial t} \tag{8}$$

where Z_n characterizes the soundproofing material over a surface A_F [the dimensions: $[Z_n]$ = (force time/length^3) or (force/velocity)(1/area)]. The classical boundary conditions using Eq. (8) become, rigid wall ($Z_n \to \infty$), and the nonreflecting boundaries, ($Z_n = \rho c$). A simple application of the last technique,[28] was proposed in an analytical case in which the eigenvalues of a one-dimensional acoustic domain were found. Tables 1 and 2 respectively show the results for the case of conservative boundary conditions

$$(p)_{x=0} = 0, \qquad \left(\frac{\partial p}{\partial x}\right)_{x=L} = 0 \tag{9}$$

and for nonconservative boundary conditions

$$(p)_{x=0} = 0, \qquad \left(\frac{\partial p}{\partial x}\right)_{x=L} = -\frac{\rho}{Z_n}\left(\frac{\partial p}{\partial t}\right)_{x=L} \tag{10}$$

The results were obtained using the following set of values: $\rho = 0.125$ kg s^2/m^4, $Z_n = 1 + j$ kg s/m^3, $L = 1$ m (length of

Table 1 One-dimensional-eigenvalues (conservative)[28]

i[a]	Theory	FEM	Error %
1	85	85.00	0.00
2	255	254.91	0.03
3	425	424.57	0.10
4	595	593.81	0.20
5	765	762.47	0.33
6	935	930.38	0.49

[a]Mode number.

Table 2 One-dimensional-eigenvalues (nonconservative)[28]

i	Theory	FEM	Error %
1	168.74	168.70	0.02
2	338.74	338.51	0.07
3	508.74	508.00	0.15
4	678.73	676.97	0.26
5	848.69	845.29	0.40
6	1018.76	1012.75	0.59

the one-dimensional domain), and $c = 340$ m/s, where j is $\sqrt{-1}$ the imaginary unit. The FEM calculations are in good agreement with the analytical results for both the conservative and nonconservative boundary conditions. Clearly, the presented approaches are not unique, but they are probably the most simple to use. The common problems in the use of these techniques are related to the link between the numerical/experimental information and the real operating conditions, and to the required computational effort.

The field approach needs all parameters, about the materials, that could be measured in a laboratory. The impedance approach can be used if specific impedance measurements are available and assuming the same operating behavior when actually assembled on the real structure. It was relatively simple to solve both approaches with the finite element method, as well as using the integral formulation of the boundary element method,[17] but the problem of the simulation of the real behavior of the soundproofing layer still remains. Furthermore, it should be noted that the solution of Eq. (7) requires the inclusion of more degrees of freedom (DOF) for the thickness of the soundproofing layer in the finite element approach. This also adds to the total computational cost.

IV. Acoustic Structural Coupling Interface Analysis Code

The various steps necessary to optimize the phases of the numerical simulations discussed above has indicated the need for a generic code useful for fluid-structure interaction analysis. The generic code should be usable by structural acoustic specialists as well as general designers using some level of care. In response to this need, the acoustic structural coupling interface analysis code (ASCIA) was developed.

ASCIA can receive from the CAD phase the structural and acoustic finite element models and the equivalent nodal areas necessary to translate in discretized form the nondissipative boundary conditions [Eqs. (3) and (4)]. The procedure is strongly linked with NASTRAN card formats, because it has also the possibility to automatically run the final files. During execution ASCIA is able to: 1) search for the structural and acoustic interfacing nodes; 2) evaluate the coupling coefficients in dimensional and nondimensional form; 3) generate the damping matrix for the soundproofing simulation following the field or the impedance approach; 4) generate an acoustic load set of NASTRAN cards from existing numerical and/or experimental data, searching and associating the loaded nodes; 5) prepare the executable NASTRAN file for the modal coupled analysis and/or direct frequency response; and 6) generate equipment reports to form a short history of the performed analysis.

ASCIA allows the nonexperienced user to optimize the numerical analysis. It should be noted that these operations can be performed by other procedures, but by using ASCIA it was verified that the man-time devoted to preparing files, searching for syntax errors, checking for accuracy, and waiting for calculations is significantly less, allowing more time for analysis and interpretation of results. Enhancements of ASCIA that are in verification phase include, evaluation of the coupling areas when the structural mesh is different from the acoustic one at the interface, and adoption of the boundary element method for the acoustic field, when it is possible to neglect the fluid loading.

An enhancement to allow the user to perform modal frequency vibroacoustic response studies has also been scheduled.

V. Effect of the Mass Convection

The natural extension of the finite element analysis of the room acoustics is the simulation of the convected conservative acoustic wave equation. The governing one-dimensional equation[32–36] for the acoustic wave problem including the convection effect is the following:

$$-\frac{A}{\rho}\frac{\partial^2 p}{\partial x^2} + \frac{A}{\rho(c^2 - U^2)}\left[\frac{\partial^2 p}{\partial t^2} + 2U\frac{\partial^2 p}{\partial x \partial t}\right] = 0 \tag{11}$$

For $U = 0$, the Helmholtz equation [Eq. (2)] is returned. The problem can be solved using a discrete number of sectors in which the distribution of velocity U is known. The required two steps, 1) aerodynamic and 2) acoustic evaluations, can be easily performed using a standard commercial finite element code, allowing user modifications of the matrices. In fact,[32] the one-dimensional convection effect can be simulated including the following unsymmetrical damping matrix, B_{element}, for each one-dimensional element:

$$B_{\text{element}} = \frac{UA}{\rho(c^2 - U^2)}\begin{vmatrix} -1 & 1 \\ -1 & 1 \end{vmatrix} \tag{12}$$

The governing equation to be solved with finite element technique becomes

$$\beta^2\frac{\partial^2 p}{\partial x^2} - \frac{1}{c^2}\frac{\partial^2 p}{\partial t^2} - \frac{2U}{c^2}\frac{\partial^2 p}{\partial x \partial t} = 0 \tag{13}$$

where $\beta = \sqrt{1 - \text{Mach}^2}$.

A simple classical muffler geometry was selected as the test case. In Ref. 30 some basic trends are reported. Two trends were investigated[36]: 1) the first natural frequency (or the eigenvalue) of the system and 2) the transmission loss. The results are reported in Fig. 1. The abscissa xa represents (ip/

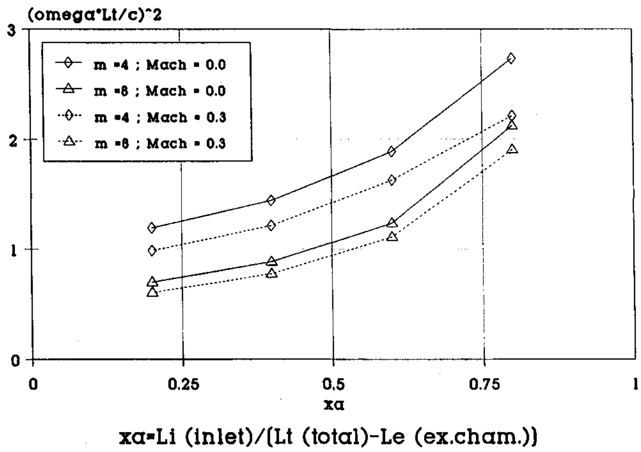


Fig. 1 Effect of the expansion chamber on the fundamental frequency of a muffler.

$L - ec$), the length of the inlet pipe measured in terms of total length minus the length of the expansion chamber. The parameter m is the area ratio. The y axis in Fig. 1 represents the ratio $z = (\omega L/c)^2$, a nondimensional form of the first natural frequency. The effect of the mass convection is evident and the frequency shift of the point of minimum transmission loss (minimum efficiency of the muffler), (very useful information in the muffler design phase), is also outlined and well-correlated with literature.[34] The used finite element model had 121 nodes and 120 one-dimensional elements (120 DOF). The feasibility, reliability, and simplicity of such acoustic analysis using the finite element method is also evident. It could allow the extension to a general acoustic finite element code useful for more complex internal geometries, using refined finite element (e.g., the exact pipe element[35]), for designing and verifying networks.

VI. Optimized Modal Approach and Coupled Eigensolutions

It is common experience that when the finite element approach is used for the dynamic response, and moreover, for the coupled structural-acoustic, the CPU time could reach unacceptable levels. One solution for lowering the cost is to use the modal approach, that for the conservative coupled system, Eq. (5), is

$$m_s\ddot{\eta} + k_s\eta = -\Phi\psi + f_s$$
$$m_A\ddot{\psi} + k_A\psi = -\rho\Phi^T\ddot{\eta} + f_A \tag{14}$$

where η and ψ are the structural and acoustic modal coordinates, respectively.

In this way a significant reduction in the DOF of the system can be obtained. However, some criticisms about the modal approach, for general application, still arise from the following considerations.[37,38]

1) The modal coordinates have no physical meaning, neither do the terms of the modal stiffness and the mass matrices.

2) It is always necessary to get a reference solution, usually based on numerical (e.g., direct response) or experimental tests.

3) The uncertainties related to damping and internal and/or external constraints are such that the modal representation could lead to an incomprehensive model.

It is evident that when the reference solution is available, the modal approach is justified. For the fluid-structure interaction analysis it could be useful to analyze a nondimensional coupling factor in order to get preliminary information, simply starting from the uncoupled modal analysis. These coefficients are defined as[22]

$$F_{ij} = \frac{1}{1 + [(\omega_{jA} - \omega_{iS})^2/4](1/\Gamma_{ij}^2)} \tag{15}$$

$$\Gamma_{ij} = \sqrt{\left(\frac{\rho c^2}{\rho_s\Lambda_{ii}\Lambda_{jj}}\right) B_{ij}^2} \tag{16}$$

$$B_{ij} = \int_{A_c} \phi_{iS}\phi_{jA}\,\mathrm{d}S \tag{17}$$

where $\Lambda_{ii} = \int_{A_c} \phi_{iS}\phi_{iS}\,\mathrm{d}S$ and $\Lambda_{jj} = \int_V \phi_{jA}\phi_{jA}\,\mathrm{d}V$, with V being acoustic domain.

The parameter B_{ij} can be interpreted as the geometrical component of the coupling, while F_{ij} is interpreted as the weighted nondimensional value with respect to the parameters of the problem being tested. For simple configurations analytical solutions for B_{ij} and F_{ij} are available. A specific nondimensional value (e.g., $F_{1,2}$), gives the relative importance of the second acoustic eigenvector when coupled with the first structural mode shape. The equation for writing F_{ij}, and clearly for B_{ij}, were generalized to include them in a finite element environment (ASCIA), allowing their evaluation for a general coupling between an elastic solid and an acoustic limited volume, interfacing over a surface A_c. Examination of the complete nondimensional matrix coefficients can give some preliminary information about the coupled model, allowing the selection of the smallest number of most significant pairs of modes while maintaining a significant dynamic content of the system. The application of this criterion to a simple model, such as a flexible plate with an acoustic box, gave good results in comparison with the analytical results. Also, in more complex configurations, the use of these coefficients has been very useful, even though further analysis is required. In writing the reduced scheme, it can be easily shown that the modal coupling area matrix can be read as a rectangular matrix containing the B_{ij} coefficients

$$\Phi_{ij} = \phi_{iS}[AN]\phi_{jA}$$

where $[AN]$ is the matrix of the equivalent nodal area.

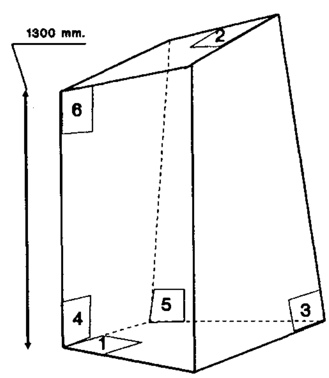
The evaluation of the coefficients starts from the uncoupled structural and acoustic eigenvectors, therefore a deeper investigation is required for better assessing this hypothesis.

VII. Effectiveness of the Numerical Simulations in the Design Phase

A joint research effort (ASANCA) involving the Italian Aerospace Research Center (CIRA) and the University of Naples "Federico II" focused on the effectiveness of the numerical analysis of acoustoelastic behavior of structures in the preliminary and design phase of a new project. The suggested model, used as test case, was a metallic box, with six slanting walls (Fig. 2), whose shape and dimensions were selected with the following requirements: 1) high modal density from the lowest possible frequencies; 2) simplified connections for representing the standard structural joints; 3) two structural elements (beams and plates); and 4) acoustic cavity dimensions close to the typical ratios of a helicopter cabin.

On this test article, excited by a mechanical shaker, the comparison with the experimental results were carried out with three different methodologies: 1) a finite element method for both the acoustics and the vibrations—at CIRA; 2) a finite element method for the vibrations and boundary element method for the acoustics (the fluid loading effect was neglected)—at Agusta; and 3) a statistical-energy approach—at the Defense Research Agency (DRA). For the CIRA activities, two finite element models, coarse and fine, were assembled (Table 3).

In the present work only the results related to the CIRA applications, in collaboration with the Institute of Aircraft Design, have been presented. For the preliminary analysis, the ratio between the computational costs and the reliability of the results promptly clarified the usefulness of adopting the coarse model. The selected model was updated in various



Mechanical shaker on Plate Nr.5

Fig. 2 Sketch of ASANCA box.

Table 3 Finite element models for ASANCA box

Model definition	DOF		
	Structural	Acoustic	Total
Coarse	1,530	495	2,025
Fine	8,930	5,985	14,915

Table 4 Modal analysis of the simply supported plate no. 5

Mode number		Natural frequencies, Hz	
Experimental	FEM	Experimental	FEM
1	1	19.72	20.00
4	2	31.11	30.15
8	3	49.25	47.35
13	4	85.63	87.00

Table 5 Modal analysis of the free frame

Mode number		Natural frequencies, Hz		
	FEM		FEM	
Experimental	Model A	Experimental	Model A	Model B
1	——	20.20	——	——
2	1	31.65	28.60	——
5	2	48.07	47.04	——
8	3	69.68	65.50	65.68
10	4	85.49	86.00	——

Table 6 Position (mm) of the microphones and the exciter, and list of the excitation conditions

	X	*Y*	*Z*
Microphone number			
1	390	200	1065
2	190	235	840
3	215	710	310
4	300	500	1035
5	155	620	1225
6	360	460	320
Exciter number			
1	0	300	850
2	0	550	850
3	0	550	600
4	0	300	600
Excitation conditions			
a) Exciter no. 1 active			
b) All exciters active			

steps, to improve matching the real characteristics, such as the mass of the rivets, the effect of the torsion of the beams, etc., always using the drawings of the box in a way to replay the design phase of the industrial product.

A. Modal Analysis

The experimental modal analysis of the box[39] addressed an attempt to separate the dynamic behavior of the elements of the test article; the plates, the frame, and the total model, always starting from using the complete model, in order to allow an easier understanding of the numerical results. Using the experimental mode shapes and eigenvalues, it is possible to summarize the results for plate no. 5 (Table 4).

The analysis of the isolated frame element is a little more difficult. In fact, the numerical models are really separated from the global structure, whereas in the experimental test they were extrapolated. Two finite element models (A and B) were assembled with the same degrees of freedom, but with a different mass distribution along the longerons of the removable plate (plate no. 3). The results are reported in Table 5.

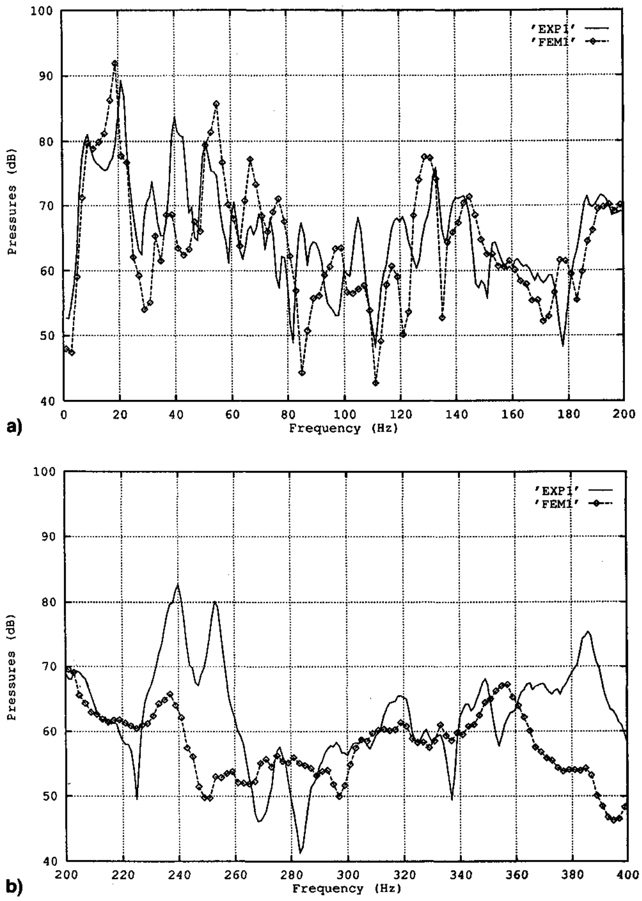


Fig. 3 Experimental vs numerical results for ASANCA box, microphone no. 1: a) 0–200 and b) 0–400 Hz.

The identification of the mode shapes and the comparison with the related eigenvalues was relatively simple for model A. The modification to the mass distribution caused a high sensitivity to the geometry of the mode shapes, thereby reducing the possibility of correct mode identification using the experimental results. It should be remembered that the extrapolation technique of the frame modes clearly does not allow similar confidence for the interpretation of the results based on experiments, as that coming from the numerical ones. However, where it was possible to compare the mode shapes, the numerical results are in good correlation with the experimental ones. As a further check, the numerical and the real weight of the ASANCA box were found to be in good agreement (525 N vs 530 N).

B. Frequency Response

A mechanical shaker exciting located in the middle of plate no. 5 was used as the forcing input, and the internal noise was measured at six microphone locations. For the sake of clarity the list of positions of the microphones and the mechanical shaker is reported (Table 6), considering that the *yz* plane of the reference system Oxyz is approximatively coincident with two edges of plate no. 5, and the origin is placed in the corner among the plates nos. 1, 3, and 5 (Fig. 2). The results coming from the numerical simulation are referred only to the excitation condition a) in Table 6.

The results [Figs. 3–7 (8 frequency responses and 1 graph for the octave band SPL of the microphone no. 1] show that in the 0–200-Hz frequency range the numerical-experimental comparison is quite satisfactory. The pressure trends are increasingly different in the 200–400-Hz frequency range, as expected. The results remain acceptable even in the high-frequency range, when comparing the acoustic results. Sound pressures in octave bands and overall values are very similar from the finite element calculations and the experiments. This

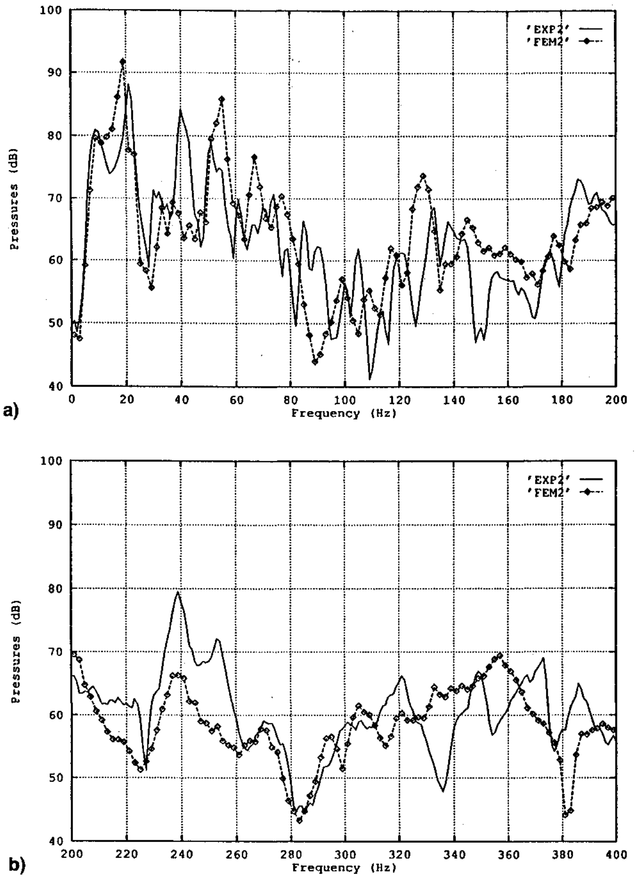


Fig. 4 Experimental vs numerical results for ASANCA box, microphone no. 2: a) 0–200 and b) 0–400 Hz.

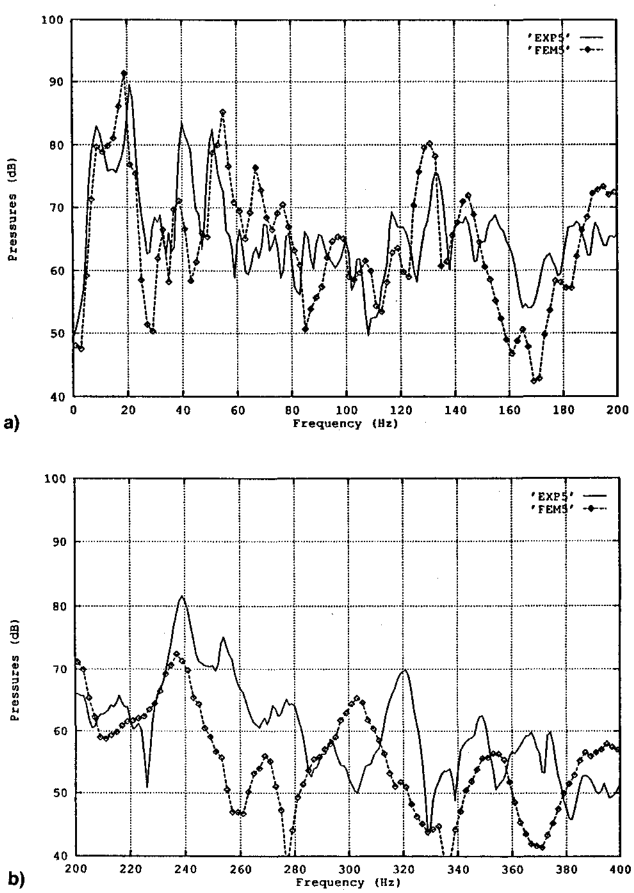


Fig. 5 Experimental vs numerical results for ASANCA box, microphone no. 5: a) 0–200 and b) 0–400 Hz.

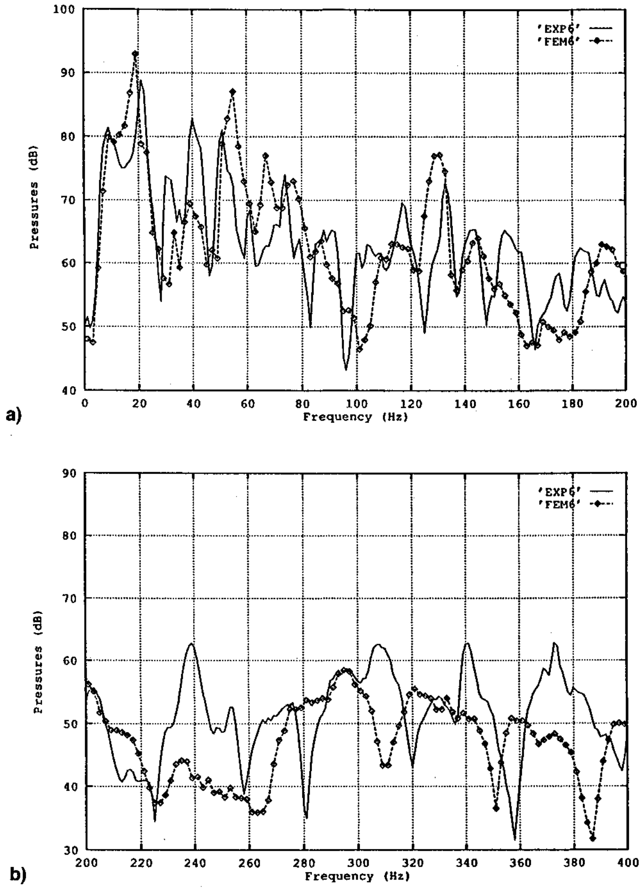


Fig. 6 Experimental vs numerical results for ASANCA box, microphone no. 6: a) 0–200 and b) 0–400 Hz.

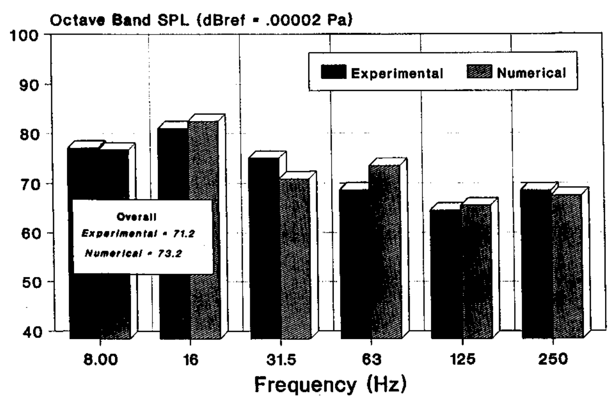


Fig. 7 Experimental vs numerical results for ASANCA box, microphone no. 1 octave band SPL.

is mainly due to the different ways of comparing the structural and acoustical behaviors, or similar to the broadband and narrow-band analysis. Thus the numerical dynamic behavior of the coupled system is globally well-predicted.

VIII. Conclusions

The present methodologies, together with those used for the high-frequency range, may be selected during the design phase, since they can help in selecting different acoustic and/or structural solutions. They appear as delicate tools that could drive toward inaccurate results. More applications to real structures are therefore necessary to improve their use. From the FEM point of view it will be important to establish whether or not a significant increase in the DOF of the models improves the quality of the results, or a different approach, perhaps based on statistics, must be adopted. The possibility to refine the basis of the finite element approach for the dynamic problems, by using oscillating terms as shape functions, has been also investigated,[27] but the results were completely unsatisfactory. In the continuation of the ASANCA project, intermediate phase of BRITE/EURAM, we will address the following items for the box problem: 1) a better identification of the boundary conditions and plate joints effects, and 2) refinement of the mode identification and the modal density with the finite element approach.

ASANCA was a severe test for the FEM, but it is the opinion of the authors that it can be still used as reference model.

This article reported the state of the art of the structural acoustic activities developed at the University of Naples "Federico II," in collaboration with CIRA and supported by Alenia, and in the final phase by the EEC through the BRITE/EURAM Research Project. The main analytical steps have been reviewed for both conservative and dissipative governing equations of the fluid-structural coupling problem, even with different boundary conditions. Particular attention has been devoted to the improvement of the numerical tools from several points of view: an increase of the modeling complexities, the extension of the procedures to include similar order problems, the improvement of the modal approach for limiting computer costs, the easy applicability for nonexpert general end-users, and comparison with experimental results for applicability in the design phase of a new project. As a result of these activities, the numerical procedures evidenced some discrepancies, pointing out the need for further improvement, but at the same time showing, wherever necessary, the validity of the FEM approach in the low-frequency band. Further data, mainly of the experimental type on real structures, are necessary and are now under definition for the improvement of the numerical prediction as well as to permit the use of other numerical procedures for the next and necessary enhancements.

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