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DYNAMIC BEHAVIOUR OF A RECTANGULAR UNBAFFLED PLATE INMERSED IN A DIFFUSE FIELD

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ABSTRACT

A method to estimate the response and the dynamic characteristics of an aerospace structure excited by the acoustical loads produced during a rocket launch is presented. These elevated pressure loads can be critical in the design of large lightweight structures such as solar arrays and communication reflectors where high acceleration levels can be achieved. The resulting acoustic field can be considered as a diffuse field composed of a large number of uncorrelated incident plane-waves traveling in different directions that impinge over the structure surface. A Boundary Element Method has been used to compute the pressure jump produced by an incoming plane-wave on an unbaffled rectangular plate and the fluid-structure coupling loads generated by the plate own vibrations. This method is based on Kirchhoff's integral formulation of the Helmholtz equation for the pressure field taking into account the Sommerfeld radiation condition. The generalized forces matrix due to the fluid loading is then determined taking the modes of the plate in vacuum as base functions of the structural displacement in the present problem. These modes are obtained by means of a Finite Element Model. An iteration procedure has been developed to calculate the natural frequencies of the coupled fluid-plate system. Comparisons of the present method with various experimental data and other theories show the efficiency and accuracy of this method for any support condition of the plate and the validity of the present procedure for the values of the frequency of excitation that appear in an acoustical test performed in a large reverberant chamber.

1. INTRODUCTION

Spacecraft structures located in the payload compartment are submitted to an intense acoustic field during the launcher lift-off. These high pressure levels excite primarily secondary structures of a satellite such as communication reflectors, solar arrays or thin payload panels. With the advent of new technologies and the appearance of structures made of sandwich composite materials, their mass has decreased considerably thus producing an increase in the acceleration levels due to the acoustic loads that can damage sensitive parts of the structure and the electronic equipment attached to it. For this reason, light aerospace structures are subjected to intense acoustic test in large reverberant chambers and reliable analysis methods are needed during its design to guarantee that the structure comply with all the acoustic requirements of the launcher, hence the qualification test campaign can be successfully afforded.

Two effects need to be considered in order to obtain the structure dynamic response under acoustic loads. First, the effect of the structure surrounding fluid that transmit to the exterior domain the pressure waves generated by the plate vibration. Second, acoustic waves produced by other sources, impinge over the structure surface, forcing it to vibrate. These two effects are coupled producing a continuos feedback between the structural and acoustic behavior.

The influence of the surrounding fluid on the dynamic characteristics of structures has been well known for many years. Fluid presence affects considerably its natural frequencies and normal modes, even in the case of light fluid, like air, when the structures are constructed with composite sandwich panels. References [8], [16], [18] and [20] provide good examples of it. Most of these works were concerned with underwater applications, and therefore the surrounding fluid was considered a liquid with low compressibility effects. Besides, these structures, mainly rectangular and circular plates, were modeled as "baffled", embedded in an infinitely rigid plane. Acoustic radiation of baffled rectangular plates has been studied in great detail in references [3], [7], [14], [18], [24], obtaining the acoustic pressure distribution employing either the Rayleigh integral equation [25] or using the Fourier transform of its impulse response. Only recent research [1], [11], [22] has been focused on unbaffled plates.

There are mainly two methods capable to determine the acoustic behavior of complex geometries, Boundary Element Methods (BEM) and Finite Element Methods (FEM). The last one is a powerful tool to model general structures of arbitrary shapes and is extensively employed in structural analysis. However, the application of this method to acoustical problems necessitates the discretization of the surrounding acoustic media. This leads at high frequencies or unbounded fluid domains to algebraic systems of large size which increases the computational cost and in addition, the Sommerfeld radiation condition is very difficult to impose at the external mesh boundary. To overcome the previous limitations, infinite acoustic wave envelope elements [6] have been recently developed for simple cases, but they are not fully implemented yet. B. E. M. is an alternative method to study problems on fluid-structure interaction in which the fluid is unbounded. This method is fully described in references [2], [5], [9], [10] where a variational approach is adopted to solve the integral formulation of the Helmholtz equation for the pressure field, combined with a normal modes analysis of the structural response. Other methods as [28] are based on the time domain instead of the frequency domain, but it seems that frequency domain methods are more efficient for the low to medium frequencies range in which we are more interested. These two methods are only valid for frequencies from 0 to about 400Hz. At higher frequencies modal densities are so high that is very difficult to identify and calculate normal modes with accuracy, with the subsequent increase in computational time if modal analysis techniques are performed. Therefore, at high frequencies statistical methods have been used like SEA [15], [23]. This method consists in a statistical analysis of the energy or power flow between subsystems submitted to random loading.

Different studies [26], [30] have shown that the acoustic lift-off noise of a launcher can be characterized as a diffuse field. To take this loading into account on the structure response, realistic modellisation of the acoustic field has to be accomplished. Several authors [17], [27], [29] describe statistically the sound distribution of reverberant sound fields. Nelisse et al [21] presents an analytical method to calculate the structural response of a unbaffled panel placed in a rectangular cavity coupling the deformation modes of the structure with the influence of the chamber walls, considered these as rigid. References [1], [4], [12], [22] assume that the diffuse field is composed of a superposition of uncorrelated incident plane waves traveling in different directions. Each direction is characterized by a pressure spectral density.

In this paper, the response of an unbaffled rectangular plate with arbitrary boundary conditions immersed in a diffuse field is calculated, modeling the acoustic field as a large number of uncorrelated plane waves. A Boundary Element Method has been used to compute the pressure jump produced by an incoming plane-wave over the plate surface and the loads generated by the plate own vibrations. This method is based on Kirchhoff's integral formulation of the Helmholtz equation for the pressure field that uses an elemental solution that satisfies the Sommerfeld radiation condition. The integral equation is solved by means of a collocation technique and the finite part of the singular integral is obtained analytically. The generalized forces due to the fluid loading are determined using the vacuum modes of the plate, obtained with a F.E.M model, as base functions of the structural displacement. An iteration procedure has been developed to calculate the natural frequencies of the plate surrounded by a compressible fluid. The response of the plate forced by an unitary pressure wave traveling in a specific direction is then computed and the contributions due to all the different directions are combined to obtain the global spectral density for a particular degree of freedom of the structure.

2. PROBLEM FORMULATION

A thin, flat rectangular plate with any type of support condition placed at the z = 0 plane is considered. The plate is surrounded by an infinite fluid domain and an acoustic plane wave traveling at a direction defined by the angles θ and ϕ (see fig. 1.) impinges over its surface.

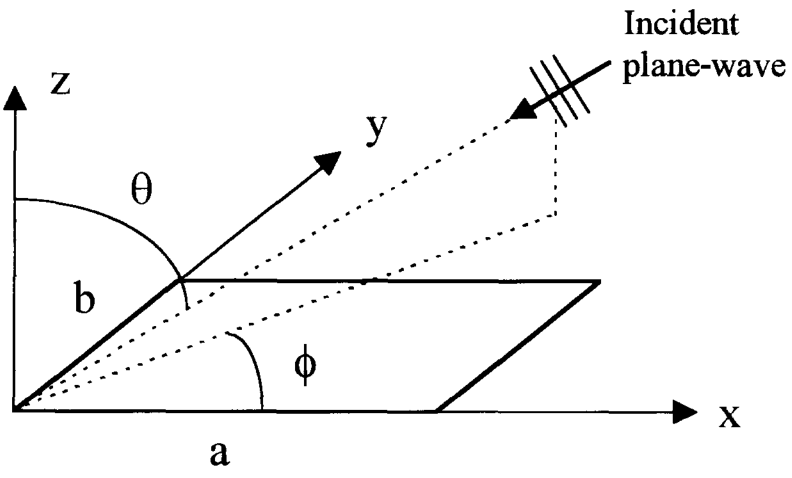

Figure 1: Geometrical representation of an unbaffled plate submitted to a plane-wave.

The differential equation of motion for the transverse displacement w(x,y,t) of the plate when the thickness is constant can be written as:

$$D\nabla^4 w + \rho_M \frac{\partial^2 w}{\partial t^2} = \Delta P_{vib}(x,y,t) + \Delta P_{wave}(x,y,t) \tag{1}$$

Where ΔP_{vib} is the pressure jump due to the plate vibration and ΔP_{wave} is the pressure jump due to the incident plane wave over its surface when the plate is considered as rigid. Assuming small perturbations in a compressible, inviscid and irrotational fluid, these pressure distributions can be calculated by solving the wave equation on the fluid domain, given by:

$$\nabla^2 p - \frac{1}{a_\infty^2}\frac{\partial^2 p}{\partial t^2} = 0 \tag{2}$$

Application of the momentum equation at the surface of the plate yields the boundary condition:

$$\frac{\partial p}{\partial z} = -\rho_\infty \frac{\partial^2 w}{\partial t^2} \qquad \text{at } z = \pm 0 \tag{3}$$

At a large distance from the plate, the Sommerfeld radiation condition has to be satisfied. For the determination of the dynamic characteristics of the coupled fluid-structure system it will be assumed that the motion is harmonic for both the fluid and the structure. The deformation of the latter will be expressed as function of the normal modes of the plate in vacuum, which have been determined by the finite element model provided by MSC/NASTRAN. Let these modes be W_{mn}., as they are only known at the nodes of the FEM model, a curve fit in terms of Lagrange polynomials is used to obtain an analytical expression for them. The deformation of the plate is then expressed as:

$$w(x,y,t) = \sum_{m=1}^{M}\sum_{n=1}^{N} q_{mn}^0 \, e^{\imath\omega t} \, W_{mn}(x,y) \tag{4}$$

In what follows the independent variables x,y,z and ξ,η are nondimensionalized by the structure characteristic length l defined as $l = \sqrt{a \cdot b}$.

2.1 CALCULATION OF THE PRESSURE JUMP ΔP_{vib}

For purely harmonic motion and applying Green's theorem, equation (2) can be expressed [10] as a distribution of dipoles placed on the plate surface:

$$P(x,y,z) = -\frac{1}{4\pi}\iint_{S_P} \Delta P(\xi,\eta) \cdot \frac{\partial}{\partial z}\left(\frac{e^{-\imath kR}}{R}\right) d\xi d\eta \tag{5}$$

where $\Delta P(\xi,\eta)$ represents the pressure difference between the lower and the upper surface of the plate, $\frac{e^{-\imath kR}}{R}$ is the fundamental solution of equation (2) for harmonic motions which satisfies Sommerfeld radiation condition.

By applying the boundary conditions on the plate surface, the following integral equation is obtained for the determination of the pressure jump across the plate

$$\left.\frac{\partial P(x,y,z)}{\partial z}\right|_{z=0} = \rho_\infty \omega^2 \sum_{m=1}^{M}\sum_{n=1}^{N} q_{mn}^o \, W_{mn}(x,y) = -\frac{1}{4\pi}\iint_{S_P} \Delta P_{vib}(\xi,\eta) \left.\frac{\partial^2}{\partial z^2}\left(\frac{e^{-ikR}}{R}\right)\right|_{z=0} d\xi \, d\eta \tag{6}$$

In this equation for each mode W_{mn} a pressure distribution $(\Delta P_{vib})_{mn}$ will be obtained.

2.2 CALCULATION OF THE PRESSURE JUMP ΔP_{wave}

For harmonic motions, the pressure imposed by a plane wave at any point of space can be written [22] as:

$$P_i(x,y,z) = P_o e^{-ik(\sin\phi\cos\theta + \sin\phi\sin\theta + z\cos\phi)} \tag{7}$$

The pressure field (in this case the sum of the incident plane wave field and the diffracted field) must comply with equations (2) and the sommerfeld radiation condition with the following boundaring condition at the plate surface, this considered as perfectly rigid.

$$\frac{\partial p}{\partial z} = 0 \qquad \text{at } z=\pm 0 \text{ in } S_p \tag{8}$$

By applying Green's theorem, the equation for the pressure field can be written as:

$$P(x,y,z) = P_i(x,y,z) + \frac{1}{4\pi}\iint_{S_p} \Delta P_{wave}(\xi,\eta)\cdot\frac{\partial}{\partial z}\left(\frac{e^{-ikR}}{R}\right) d\xi\, d\eta \tag{9}$$

By imposing the boundary condition (8) the following integral equation to calculate ΔP_{wave} is obtained

$$ik\cos\phi P_o e^{-ik\sin\phi(\cos\theta x+\sin\theta y)} = -\frac{1}{4\pi}\iint_{S_p} \Delta P_{vib}(\xi,\eta)\frac{\partial^2}{\partial z^2}\left(\frac{e^{-ikR}}{R}\right)\Bigg|_{z=0} d\xi\, d\eta \tag{10}$$

2.3 BOUNDARY ELEMENT METHOD

Integral equations (6) and (10) are solved utilizing the BEM described in [11]. The plate is divided into rectangular elements and on each element the unknown pressure distribution is assumed constant. By satisfying the integral equation at $I*J$ control points placed at the center of each rectangular element, a linear system of $I*J$ equations is obtained for the determination of the pressure jump distribution (either ΔP_{vibe} *or* ΔP_{wave}). The coefficients for the linear system of equations are the double integrals over each element ij, of the fundamental solution of the wave equation. Thus, the integration to be performed will be:

$$\iint_{S_{i,j}} \frac{\partial^2}{\partial z^2}\left(\frac{e^{-ikR}}{R}\right)\Bigg|_{z=0} d\xi\, d\eta \tag{11}$$

In this integral special care must be taken of the singularity that occurs when the element of integration coincides with the element where the boundary condition is satisfied. After applying the second derivative with respect to z, the following kernel function $K(x_{rs}-\xi, y_{rs}-\eta)$ is obtained:

$$K(x_{rs}-\xi, y_{rs}-\eta) = \frac{\partial^2}{\partial z^2}\left(\frac{e^{-ikR}}{R}\right)\Bigg|_{z=0} = -ik\frac{e^{-ikR|_{z=0}}}{R^2|_{z=0}} - \frac{e^{-ikR|_{z=0}}}{R^3|_{z=0}} \tag{12}$$

where (x_{rs}, y_{rs}) are the coordinates of the control point.

At this stage, two cases can be distinguished. The first, is when the point of control is located outside the panel over which the integration is being undertaken, and no singularity is present. Integration of the kernel given at (12) is then numerically evaluated by means of a double Gaussian integration procedure. The second is when the point of control is located within the limits of the panel over which the integration is being undertaken, in this case the integral is singular and the singular part of the kernel $K(x_{rs}-\xi, y_{rs}-\eta)$ must be extracted when $\xi \to x_{rs}$ and $\eta \to y_{rs}$. The integration method to avoid this singularity consists in adding and subtracting the singular part K_s from the kernel K. The first integral which is non-singular is evaluated numerically with a gaussian method as was done for the nonsingular case, and the second is obtained analytically.

$$\iint_{S_p} K(x_0,y_0)\partial x_0 \partial y_0 = \iint_{S_p} (K(x_0,y_0) - K_s(x_0,y_0))\partial x_0 \partial y_0 + \iint_{S_p} K_s(x_0,y_0)\partial x_0 \partial y_0 \tag{13}$$

Once the integration on each element is performed, a linear system of equations is determined to compute the pressure jump desired. Eqs. (6) and (10) can be written then as:

$$-\frac{1}{4\pi}[aic]\{\Delta P_{vib}\}_{mn} = \rho_\infty \omega^2 l q^o_{mn}\{W_{mn}\} \tag{14}$$

$$-\frac{1}{4\pi}[aic]\{\Delta P_{wave}\} = \{P_i\} \tag{15}$$

where $[aic]$ is a square matrix of $(IxJ)x(IxJ)$ elements, $\{\Delta P_{vib}\}$, $\{W_{mn}\}$, $\{\Delta P_{wave}\}$ and $\{P_i\}$ are vectors of IxJ elements. Solving both linear system of equations the pressure distribution of the plate can be obtained.

The generalized force for the uv^{th} mode is defined as the work done by the total pressure distribution considering all the modes of vibration on the uv^{th} mode, and can be expressed as

$$Q_{uv} = \iint_{S_p} \Delta P\, W_{uv}\, d\sigma = \iint_{S_p} \Delta P_{vib}\, W_{uv}\, d\sigma + \iint_{S_p} \Delta P_{wave}\, W_{uv}\, d\sigma \tag{16}$$

The equations of motion for the generalized coordinates associated with the modes of vibration of the plate, with the forces exerted by the plane wave, can then be expressed as

$$\left(-\omega^2[M]+[k]\right)\{q\} = \{Q\} = \left[Q_{mn}^{vib}\right]\{q\} + \left\{Q^{wave}\right\} = \omega^2[M]_f\{q\} + \left\{Q^{wave}\right\} \tag{17}$$

and rearranging equation (16) it can be expressed as

$$\left(-\omega^2([M]+[M]_f)+[k]\right)\{q\} = \left\{Q^{wave}\right\} \tag{18}$$

The solution of this linear system leads to the response of the plate to an incident plane wave. If no external forces are considered except those produced by the surrounding fluid due to the free vibration of the plate the resultant homogeneous system will provide the natural frequencies and the normal modes of the coupled fluid-structure system.

2.4 CALCULATION OF THE DYNAMIC RESPONSE OF A STRUCTURE INMERSED IN A DIFFUSE FIELD.

A diffuse field can be modeled by the superposition of plane waves traveling through different directions[4]. Let's assume that the elemental power spectral density of the incident wave in the "ij" direction, defined by the angles θ and ϕ of fig. 1., is $W_{pij}(w)$ in Pa^2/Hz. If the different directions of the incident waves aren't correlated the total power spectral density W_p would be the addition of all the elemental densities. As a launcher acoustic requirements are given in NdB per octave [30] for a specific bandwidth, the total power spectral density W_p is assumed constant over such bandwidth, obtaining the following expression:

$$N = 10 \cdot log\frac{P^2}{P_r^2} = 10 \cdot log\frac{W_p \Delta f}{P_r^2} \tag{19}$$

Where $P_r = 2*10^{-5}$ Pa is the reference pressure. From this equation it can be obtained that the power spectral density can be written as:

$$_p = \frac{P_r^2}{\Delta f}(10)^{\left(\frac{N}{10}\right)} \tag{20}$$

The space field is divided into a finite number of directions θ_i, ϕ_j and associated to each direction there is a solid angle a_{ij}. It is assumed that the power spectral density of the ij direction is $a_{ij}*W_p(\omega)$, where W_p is the total power spectral density.

Once all the problem parameters are defined, the calculation of the structural response to the diffuse field is obtained as follows:

let's $H_{r,ij}$ (w) be the transfer function between the r degree of freedom of the structure and the plane wave of ij direction. The power spectral density induced by all the plane waves occupying the solid angle corresponding to the ij direction for the r^{th} degree of freedom is:

$$W_{r,ij} = \left|H_{r,ij}(\omega)\right|^2 W_{pij} = \left|H_{r,ij}(\omega)\right|^2 a_{ij} W_p \tag{21}$$

As all directions are uncorrelated, it can be demonstrated for a random field [31] that the structural response is obtained as the sum of the elemental spectral densities.

$$W_r(\omega) = W_p(\omega)\sum_{i,j}\left|H_{r,ij}(\omega)\right|^2 a_{ij} \tag{22}$$

Thus, a relation between the power spectral density of the r structure degree of freedom, the acoustic power spectral density W_p and the transference function between the r degree of freedom and the plane waves composing the diffuse field is derived.

2.5 PROCEDURE FOR THE COMPUTATION OF THE NATURAL FREQUENCIES

The natural frequencies of the system are determined by making the determinant formed with the two mass matrices and the stiffness matrix equal to zero. However, the added-mass matrix $[M]_f$ depends on the frequency of oscillation of the plate and, therefore, an iteration procedure needs to be used in order to obtain the natural frequencies of the coupled fluid-structure system.

The iteration scheme developed is as follows (for more details see [11]):

First, the natural frequencies of the system are computed assuming that the surrounding fluid is incompressible.

A set of reduced frequencies are then determined, defined as $k_j = \dfrac{\omega_{j\,incomp} \cdot l}{a_\infty}$ where ω_j is the j^{th} natural frequency of the coupled system incompressible fluid-structure.

By taking these results as an initial guess and letting now the added-mass matrix to be a function of k, the natural frequencies of the system are recalculated until convergence has been achieved

For each natural frequency the procedure converges in two or three iterations. It should be noted that for a compressible fluid the natural frequencies are obtained one by one while for an incompressible fluid all of them are obtained at the same time.

Once the natural frequencies of the coupled fluid-structure system are determined, the normal modes can be computed and expressed as a linear combination of the normal modes of the structure in vacuum.

3. RESULTS

Next some results from the method developed will be presented.

To validate the B.E.M numerical code, the case of the diffraction of a plane-wave impinging over a rigid plate with dimensions of 0.8m x 0.6m has been selected, and the results obtained have been compared with the method developed by Nelisse et al [22] .The orientation of the incident plane-wave is θ=0° and ϕ =0° (perpendicular to the plate) and its amplitude is the unity. Figure 2 shows the absolute value of the sound pressure jump at the point (0.27,0.3) on the panel as a function of the frequency, between 0Hz and 500Hz. Results show an excellent agreement between both predictions. As the influence coefficients matrix is common for both the rigid and the vibrating pressure jump calculation, these result also validates the code developed to calculate the natural frequencies.

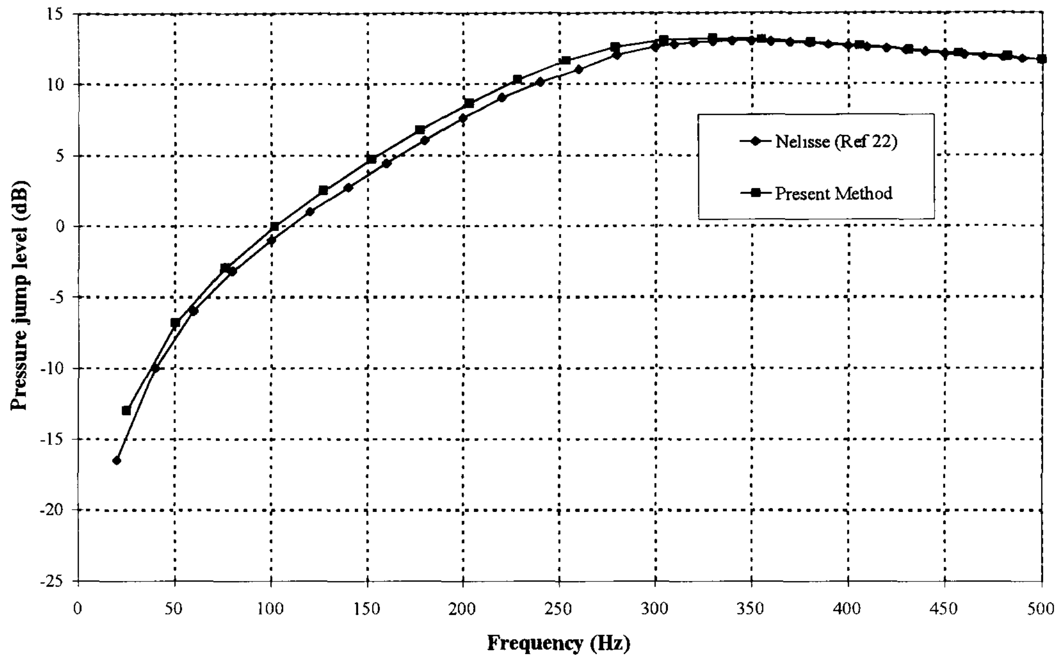

Figure 2: Pressure jump across a rigid plate for a particular point.

In the next case, the effect of the surrounding fluid density on the first four frequencies of a rectangular sandwich plate is presented. The plate is divided into 20 x 13 elements, to represent correctly the plate modes over the frequency range. This type of structure is employed in the aerospace industry because it is very light and very stiff and this is the case where the fluid effects rise considerably. Two different boundary conditions are selected, a simply supported and a free-free plate. The plate's dimensions are 0.6m x 0.386m and is made of a honeycomb core of thickness 15mm and two skins of CFRP(Carbon Fibre Reinforced Plastic) of 0.39mm thickness each one. The density of the plate is 129 Kg/m^3 and the Young's module E has an average value of $9x10^9 N/m^2$. The natural frequencies of the plate in a vacuum, submerged in air considered as an incompressible fluid (k=0) and a compressible fluid are presented in Table 1.

	Simply Supported Plate			Free-Free Plate		
Mode	Vacuum	Incompres.	Compres.	Vacuum	Incompres.	Compres.
1	590.7 Hz	550.9 Hz	570.1 Hz	366.0 Hz	356.0 Hz	355.0 Hz
2	1058.0 Hz	1009.6 Hz	1052.0 Hz	403.7 Hz	390.8 Hz	388.4 Hz
3	1521.0 Hz	1467.0 Hz	1518.0 Hz	795.2 Hz	776.1 Hz	767.1 Hz
4	1760.0 Hz	1702.0 Hz	1758.6 Hz	923.6 Hz	905.3 Hz	897.3 Hz

Table 1 : Natural frequencies of a simply supported and free-free plate surrounded by air.

The natural frequencies of the plate decrease from its vacuum value due to the effect of the air. This effect is larger in the free-free plate than in the simply supported plate. In the latter case, as we consider higher modes their natural frequencies are closer to the vacuum value. The explanation can be found if we plot the diagonal terms of the added-mass matrix as a function of the reduced frequencies (Fig.3 and Fig.4). These curves show the real part and the imaginary part of those terms. The real part causes the reduction of the frequencies and the imaginary part represents the fluid damping. The added-mass increases its value from the incompressible (k=0) case untill it reaches a maximum and then it decreases towards zero. If the reduced frequency of the mode considered in a vacuum is located before the maximum, the natural frequencies are reduced due to the fluid presence but if the maximum is placed after it the frequencies tends to reach the vacuum value.

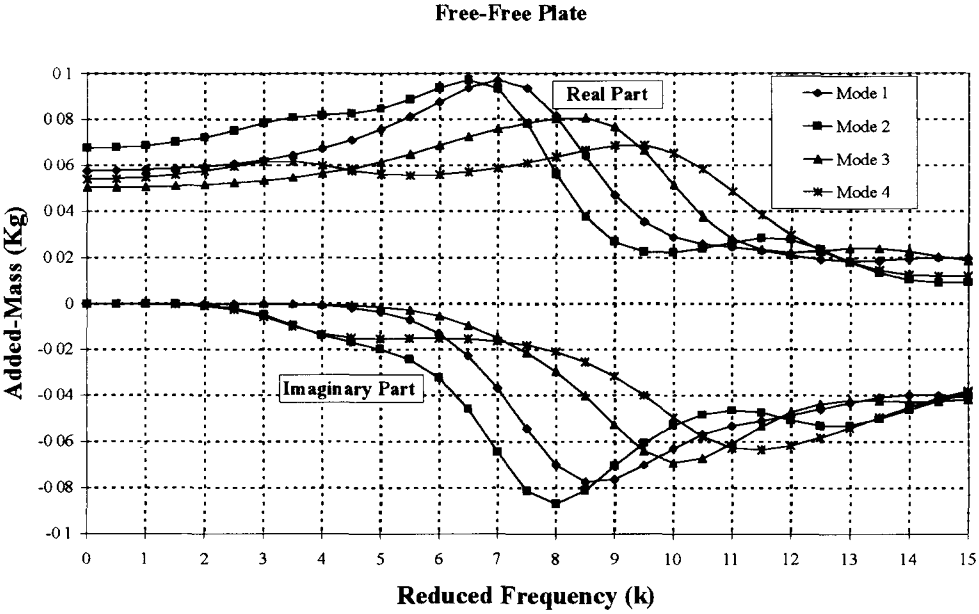

Figure 3 : Added-mass terms as a function of the reduced frequency for the free-free plate

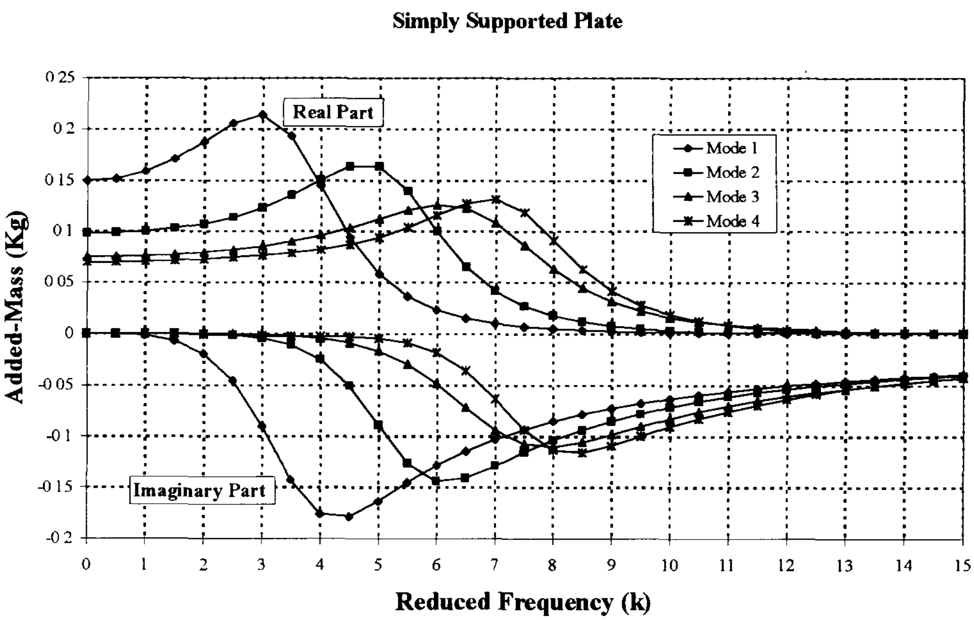

Figure 4 : Added-mass terms as a function of the reduced frequency for the simply supported plate

Figures 5 and 6 show the fluid damping ratio of the first four modes. It is defined as $\xi = \dfrac{-m_i \cdot \omega}{C_{cr}}$ where m_i is the imaginary part of the diagonal terms of the added mass matrix for each mode and C_{cr} is the critical damping of the corresponding mode. The fluid damping ratio has a maximum value of 0.096 for the simply supported plate and 0.137 for the free-free plate. Considering that sandwich panels have a structural damping ratio between 0.01 and 0.02 the acoustical damping cannot be neglected at very high frequencies. Those plots are also very useful to determine the contribution of fluid damping in the total damping measured in a test, been able to separate then the structural and acoustical damping terms.

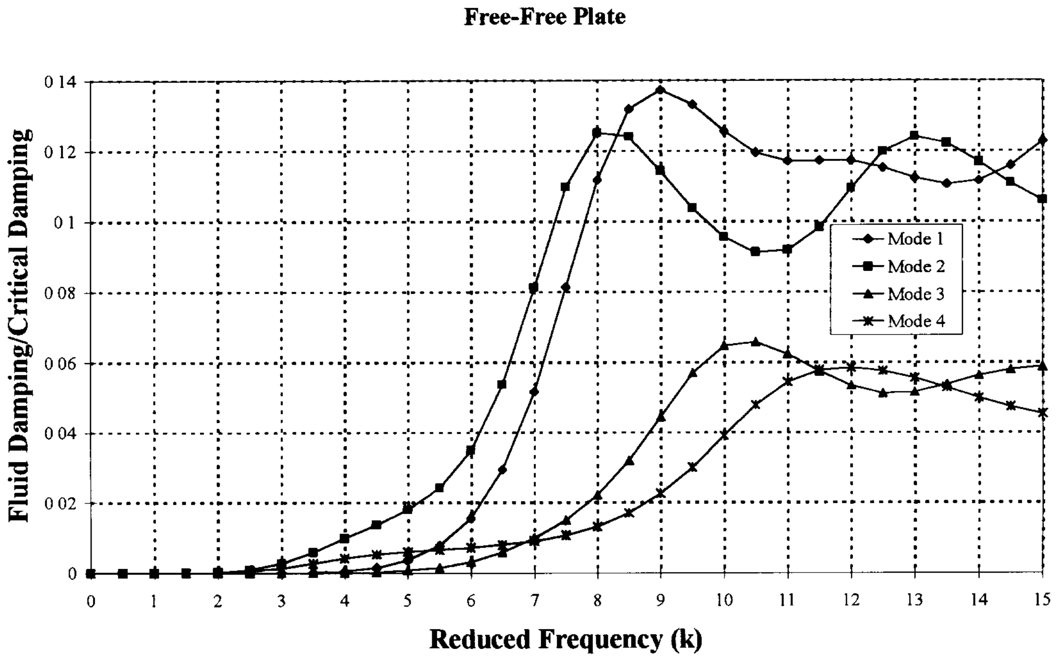

Figure 5: Fluid damping ratio as a function of the reduced frequency for the free-free plate.

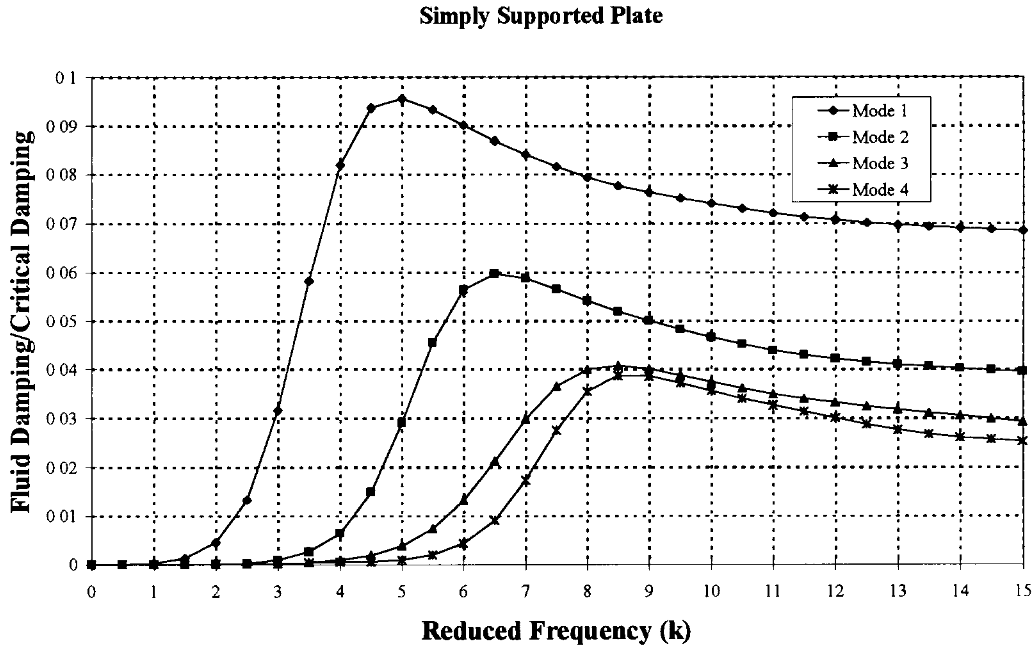

Figure 6: Fluid damping ratio as a function of the reduced frequency for the simply supported plate.

4. CONCLUSIONS

A method to calculate the dynamic characteristics of a rectangular plate of any type of support condition immersed in a diffuse field has been presented valid for low to medium frequencies. The numerical integration procedure of the kernel function has been validated with existing data and presents an excellent efficiency and accuracy. The iterative calculation of the natural frequencies shows that depending on its vacuum value and boundary condition, the natural frequencies are considerably reduced by the surrounding fluid or are similar to the vacuum ones in a sandwich plate. The added-mass and fluid damping modal coefficients as a function of the reduced frequency has been described in physical terms and its influence cannot be neglected in dynamic calculations of large frequency range. The calculation of the response of the plate to a diffuse field load with the formulation described in this paper and the extension to more general structures is currently underway.

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