# BEM-FEM Acoustic-Structure Interaction For Modeling and Analysis of Spacecraft Structures Subject to Acoustic Excitation

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**Abstract**

*Spacecraft structures are subject to acoustic load and high frequency vibration, particularly during launch, which can impose severe and adverse affect ot the structures of the spacecraft and their payloads. For many classes of structures exhibiting a plate-like vibration behavior, such as antennas and solar panels, their low-order mode response is likely to be of greatest importance. Based on these considerations, the present work is focused on modeling and analysis of spacecraft structure subject to acoustic excitation, and for this purpose, the acoustic-structure interaction problem is modeled and analyzed using boundary and finite element coupling. The analysis has been developed using three parts of approach: the calculation of the acoustic radiation from the vibrating structure, the finite element formulation of structural dynamic problem, and the calculation of the acoustic-structural coupling using coupled BEM/FEM techniques. The development of the computational scheme for the calculation of the structural dynamic response of the structure using coupled BEM/FEM will be elaborated. Some generic examples typical for spacecraft structure are elaborated.*

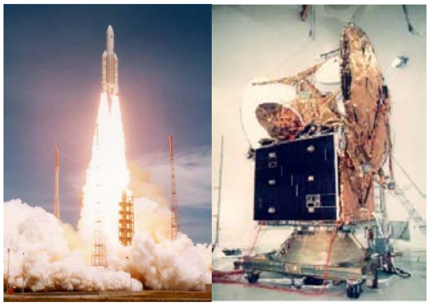
## 1. Introduction

The propagation of vibration through structures, the radiation of sound from vibrating structures, and fluid/structure interaction are all elements that are significant in structural acoustic problems on aerospace applications [1]. The loads transmitted to the spacecraft structure from the launch vehicle (LV) in the first few minutes of flight are far more severe than any load that a payload experiences on orbit. Therefore, payloads are qualified by subjecting them to loads whose magnitude and frequency contents are representative of the launch environment.

The methods and related numerical computation codes in structural acoustics for the prediction of noise emitted by structural vibration in all the audible low-, medium- and high-frequency band, play a very important role in the design and the conception of industrial products. These methods allow the design to be improved before construction and optimization with respect to the acoustical problems.

Acoustic loads are a major component of the launch environment for spacecraft structure. Exterior sound pressure levels on a spacecraft structure during launch as depicted in Fig.1a can reach 150 dB [1][2] depending on the vehicle and the launch configuration. The magnitude of the acoustic loads transmitted to the payload is a function of the external acoustic environment as well as the design of the spacecraft structure and its sound absorbing treatments.

For many classes of structures exhibiting a plate-like vibration behavior, such as antennas and solar panels as depicted in Fig.1b, their low-order mode response is likely to be of greatest importance.



(a) (b)
Fig. 1 (a) Spacecraft structure during launch, and (b) spacecraft structure in acoustic test configuration

In the Structural-Acoustic Analysis for Aerospace Structure Design problems, it is recognized that Computational Structural Mechanics (CSM) is efficient for structural-acoustic prediction in low-frequency ranges. Analysis of the detailed behavior of individual modes is possible using finite-element and classical methods.

Addressing the acoustic structural interaction problem for dealing with the influence of acoustic excitation on the structure, the coupled finite element method/boundary element method (FEM/BEM) is a convenient means of computing responses for an arbitrarily elastic structure submerged in a fluid subjected to alternating external forces and/or acoustic excitation, which has been addressed by the authors in previous work [3][4][5] as well as by other investigators [6][7][8]. The FEM is used to describe the dynamic behavior of the structure, while the BEM is used to represent the surface acoustic loading on the structure. The coupling boundary conditions between the fluid and structure are the continuity of wetted surface normal velocity and the surface pressure acting as a loading on the structure.

The formulation of the basic problem of acoustic excitation and vibration of elastic structure in a coupled fluid-elastic-structure interaction developed thus far will be summarized. The approach consists of three parts. The first is the formulation of the acoustic field governed by the Helmholtz equation subject to the Sommerfeld radiation condition for the basic acoustic problem without solid boundaries. The interface between the acoustic domain and the surface of the structure poses a particular boundary condition. Boundary element method will be utilized for solving the governing Helmholtz equation subject to the boundary conditions for the calculation of the acoustic pressure on the interface boundary.

The second part deals with the structural dynamic problem, which is formulated using finite element approach. The third part involves the calculation of the acousto-elasto-mechanic fluid-structure coupling, which is formulated using coupled BEM/FEM techniques.

The acoustic loading on the structure is calculated on the part of the boundary of the acoustic domain, which coincides with the structural surface as defined by the problem.

## 2. Helmholtz Integral Equation for the Acoustic Field

For an exterior acoustic problem, as depicted in Fig. 2, the problem domain $V$ is the free space $V_{ext}$ outside the closed surface $S$. $V$ is considered enclosed in between the surface $S$ and an imaginary surface $\Lambda$ at a sufficiently large distance from the acoustic sources and the surface $S$ such that the boundary condition on $\Lambda$ satisfies Sommerfeld's acoustic radiation condition as the distance approaches infinity.

Green's first and second theorem provide the basis for transforming the differential equations in the problem domain $V$ and the boundary conditions on the surface $S$ into an integral equation over the surface $S$.

For time-harmonic acoustic problems in fluid domains, the corresponding boundary integral equation is the Helmholtz integral equation [9][10][11].

$$cp(R) = \int_S \left( p(R)\frac{\partial g}{\partial \hat{n}_0} - g(|R-R_0|)\frac{\partial p}{\partial \hat{n}_0} \right) dS \qquad (1)$$

where $\hat{n}_0$ is the surface unit normal vector, and the value of $c$ depends on the location of $R$ in the fluid domain, and where $g$ the free-space Green's function. $R_0$ denote a point located on the boundary $S$, as given by

$$g(|R-R_0|) = \frac{e^{-ik|R-R_0|}}{4\pi|R-R_0|} \qquad (2)$$

To solve Eq. (1) with $g$ given by Eq. (2), one of the two physical properties, acoustic pressure and normal velocity, must be known at every point on the boundary surface. The specific normal impedance, which describes the relationship between pressure and normal velocity, can also serve as a boundary condition. At the infinite boundary $\Lambda$, the Sommerfeld radiation condition in three dimensions can be written as [9]:

$$\lim_{|R-R_0|\to\infty} r\left(\frac{\partial g}{\partial r} + ikg\right) \Rightarrow 0 \text{ as } r \Rightarrow \infty,\ r = |R-R_0| \qquad (3)$$

which is satisfied by the fundamental solution.

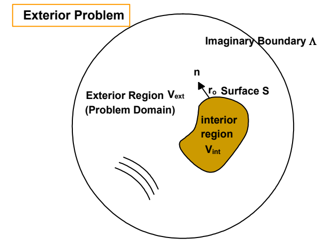



Fig. 2 Exterior problem for homogeneous Helmholtz equation

For scattering problems, Eq. (1) only gives the solution to the scattered wave. The boundary conditions are however given in terms of the total wave, which is the sum of the incident and the scattered waves. It is necessary to modify the equation to include the incident wave. For exterior scattering problems, the modification can be carried out by adding to the scattered wave integral equations the result of applying the interior Helmholtz equations to the incident wave $p_{inc}$. Then the integral equation for the total wave is given by

$$cp(R) - p_{inc}(R) = \int_S \left[ p(R)\frac{\partial g(R-R_0)}{\partial \hat{n}_0} - \frac{\partial p(r_q)}{\partial \hat{n}_0} g(R-R_0) \right] dS \qquad (4)$$

where $p = p_{inc} + p_{scatter}$, and where

$$c = \begin{cases} 1 & , R \in V_{ext} \\ 1/2 & , R \in S \\ \Omega/4\pi & , R \in S \quad \text{(non smooth surface)} \\ 0 & , R \in V_{int} \end{cases} \qquad (5)$$

The Helmholtz equation than can be discretized by dividing the continuous system into a discrete one with $N$ number of elements. Following the standard procedure in defining the elements on the boundary surface $S$, the discretized boundary integral equation becomes,

$$cp - p_{inc} - \sum_{j=1}^{N} \int_{S_j} p \frac{\partial g}{\partial \hat{n}} dS = -\sum_{j=1}^{N} \int_{S_j} g \frac{\partial p}{\partial \hat{n}} dS \qquad (6)$$

where $i$ indicates field point, $j$ source point and $S_j$ surface element $j$. The discretized equation forms a set of simultaneous linear equations, which relates the pressure $p_i$ at field point $i$ due to the boundary conditions $p$ and $v$ at surface $S_i$ of the source element $i$ and the incident pressure $p_{inc}$. It is convenient to write Eq.(6) in the following matrix form:

$$\mathbf{H}p = i\rho_0\omega\mathbf{G}v + p_{inc} \qquad (7)$$

where, **H** and **G** are two $NxN$ matrices of influence coefficients, while $p$ and $v$ are vectors of dimension $N$ representing total pressure and normal velocity on the boundary elements. This matrix equation can be solved if the boundary condition $\partial p/\partial n$ and the incident acoustic pressure field $p_{inc}$ are known.

## 3. BEM-FEM Acoustic-Structural Coupling

For convenience, following procedure described in [12], the BE region is treated as a super finite element and its stiffness matrix is computed and assembled into the global stiffness matrix and is identified as the coupling to finite elements. In the second approach, the FE region is treated as an equivalent BE region and its coefficient matrix is determined and assembled into the global coefficient matrix and is identified as the coupling to boundary elements. The state of affairs is schematically depicted in Fig. 3.

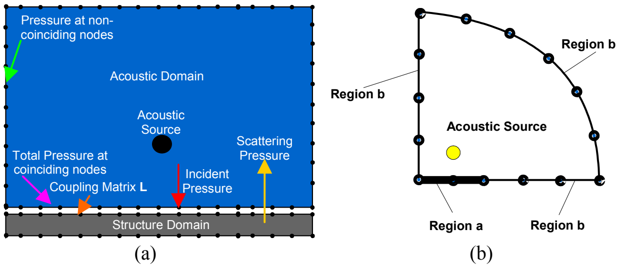



Fig 3. Schematic of Fluid-Structure Interaction Domain

The FEM leads to a system of simultaneous equations which relate the displacements at all the nodes to the *nodal forces*. In the BEM, on the other hand, a relationship between nodal displacements and *nodal tractions* is established.

The elastic structure can be represented by FE model. For a dynamic structure, the equation of motion is given by [13][14][15]

$$[\mathbf{M}]\{\ddot{x}\}+[\mathbf{C}]\{\dot{x}\}+[\mathbf{K}]\{x\}=\{\mathbf{F}\} \tag{8}$$

where **M**, **C**, and **K** are structural mass, damping and stiffness, respectively, which are expressed as matrices in a FE model. **F** is the given external forcing function vector, and *{x}* is the structural displacement vector. Taking into account the acoustic pressure $p$ on the structure at the fluid-structure interface as a separate excitation force, the acoustic-structure problem can be obtained from Eq.(8) by introducing a fluid-structure coupling term given by $\mathbf{L}\,p$ [6][12]. It follows that

$$[\mathbf{M}]\{\ddot{x}\}+[\mathbf{C}]\{\dot{x}\}+[\mathbf{K}]\{x\}+[\mathbf{L}]\{p\}=\{\mathbf{F}\} \tag{9}$$

where **L** is a coupling matrix of size $M$ x $N$ in the BEM/FEM coupling thus formulated. $M$ is the number of FE degrees of freedom and $N$ is the number of BE nodes on the coupled boundary.

The global coupling matrix **L** transforms the acoustic fluid pressure acting on the nodes of boundary elements on the entire fluid-structure interface surface a, to nodal forces on the finite elements of the structure. Hence **L** consists of n assembled local transformation matrices $\mathbf{L_e}$, given by

$$L_e=\int_{S_e} N_F^T n N_B dS \tag{10}$$

in which $N_F$ is the shape function matrix for the finite element and $N_B$ is the shape function matrix for the boundary element. It can be shown that:

$$N_F=\begin{bmatrix}1&0&0&0&0\\0&1&0&0&0\\0&0&1&0&0\end{bmatrix}[N_i] \tag{11}$$

The rotational parts in $N_F$ are neglected since these are considered to be small in comparison with the translational ones in the BE-FE coupling, consistent with the assumptions in structural dynamics as, for example, stipulated in [15].

For the normal fluid velocities and the normal translational displacements on the shell elements at the fluid-structure coupling interface a relationship has to be established which takes into account the velocity continuity over the coinciding nodes:

$$v=i\omega(\mathbf{T}x) \tag{12}$$

Similar to **L**, **T** ($n \times m$) is also a global coupling matrix that connects the normal velocity of a BE node with the translational displacements of FE nodes obtained by taking the transpose of the boundary surface normal vector $n$ [6][12]. The local transformation vector $\mathbf{T_e}$ can then be written as:

$$\mathbf{T}_e=n^T \tag{13}$$

The normal fluid velocities of the acoustic problem and the normal translation of the structural surface-fluid interface have to satisfy a certain relationship which takes into account the velocity continuity over the coinciding nodes.

The presence of an acoustic source can further be depicted by Fig.3b. Two regions are considered, i.e. $a$ and $b$ ; region a is the afore mentioned fluid-structure interface region, where FEM mesh and BEM mesh coincide and region b is the region where all of the boundary conditions (pressure or velocity) are known.

For the coupled FEM-BEM regions, Eq. (9) and Eq. (7) apply at $\{x\}\in$ a. Based on BEM-FEM model, BEM equation can now be written as:

$$\begin{bmatrix}\mathbf{H}_{11}&\mathbf{H}_{12}\\\mathbf{H}_{21}&\mathbf{H}_{22}\end{bmatrix}\begin{Bmatrix}p_a\\p_b\end{Bmatrix}=i\rho_0\omega\begin{bmatrix}\mathbf{G}_{11}&\mathbf{G}_{12}\\\mathbf{G}_{21}&\mathbf{G}_{22}\end{bmatrix}\begin{Bmatrix}v_a\\v_b\end{Bmatrix}+\begin{Bmatrix}p_{inc_a}\\p_{inc_b}\end{Bmatrix} \tag{14}$$

Using Eq.(12) for $v_a$, the BEM Eq. (14) can be modified accordingly as:

$$\begin{bmatrix}\mathbf{H}_{11}&\mathbf{H}_{12}\\\mathbf{H}_{21}&\mathbf{H}_{22}\end{bmatrix}\begin{Bmatrix}p_a\\p_b\end{Bmatrix}=i\rho_0\omega\begin{bmatrix}\mathbf{G}_{11}&\mathbf{G}_{12}\\\mathbf{G}_{21}&\mathbf{G}_{22}\end{bmatrix}\begin{Bmatrix}i\omega(\mathbf{T}.x)\\v_b\end{Bmatrix}+\begin{Bmatrix}p_{inc_a}\\p_{inc_b}\end{Bmatrix} \tag{15}$$

or:

$$\begin{aligned}\mathbf{H}_{11}p_a+\mathbf{H}_{12}p_b&=-\rho_0\omega^2\mathbf{G}_{11}\mathbf{T}x+i\rho_0\omega\mathbf{G}_{12}v_b+p_{inc_a}\\\mathbf{H}_{21}p_a+\mathbf{H}_{22}p_b&=-\rho_0\omega^2\mathbf{G}_{21}\mathbf{T}x+i\rho_0\omega\mathbf{G}_{22}v_b+p_{inc_b}\end{aligned} \tag{16}$$

If the velocity boundary condition on $b$ ($v_b$) and the incident pressure on $a$ and $b$ are known, reorganizing the unknowns to the left side, Eq.(16) become:

$$\begin{aligned}\rho_0\omega^2\mathbf{G}_{11}\mathbf{T}x+\mathbf{H}_{11}p_a+\mathbf{H}_{12}p_b&=i\rho_0\omega\mathbf{G}_{12}v_b+p_{inc_a}\\\rho_0\omega^2\mathbf{G}_{21}\mathbf{T}x+\mathbf{H}_{21}p_a+\mathbf{H}_{22}p_b&=i\rho_0\omega\mathbf{G}_{22}v_b+p_{inc_b}\end{aligned} \tag{17}$$

Since the pressure $p$ on FEM equation lies in region a, Eq. (9) can be written as

$$[\mathbf{M}]\{\ddot{x}\}+[\mathbf{C}]\{\dot{x}\}+[\mathbf{K}]\{x\}+[\mathbf{L}]\{p_a\}=\{\mathbf{F}\} \tag{18}$$

Following the general practice in structural dynamics, solutions of Eq.(18) are sought by considering synchronous motion with harmonic frequency ω. Correspondingly, Eq. (17) reduces to:

$$\left[\mathbf{K}-i\omega\mathbf{C}+\omega^2\mathbf{M}\right]\{\overline{x}\}+[\mathbf{L}]\{\overline{p}_a\}=\{\mathbf{F}\} \tag{19}$$

where

$$x=\overline{x}e^{i\omega t}\ ;\ p_a=\overline{p}_a e^{i\omega t} \tag{20}$$

or, dropping the bar sign for convenience, but keeping the meaning in mind, Eq. (19) can be written as

$$\left[\mathbf{K}-i\omega\mathbf{C}+\omega^2\mathbf{M}\right]\{x\}+[\mathbf{L}]\{p_a\}=\{\mathbf{F}\} \tag{21}$$

Combining Eq. (16) and Eq. (21), the coupled BEM-FEM equation can then be written as:

$$\begin{bmatrix} \mathbf{K}-i\omega\mathbf{C}+\omega^2\mathbf{M} & \mathbf{L} & 0 \\ \rho_0\omega^2\mathbf{G}_{11}T & \mathbf{H}_{11} & \mathbf{H}_{12} \\ \rho_0\omega^2\mathbf{G}_{21}T & \mathbf{H}_{21} & \mathbf{H}_{22} \end{bmatrix} \begin{Bmatrix} x \\ p_a \\ p_b \end{Bmatrix} = \begin{Bmatrix} \mathbf{F} \\ i\rho_0\omega\mathbf{G}_{12}v_b \\ i\rho_0\omega\mathbf{G}_{22}v_b \end{Bmatrix} + \begin{Bmatrix} 0 \\ p_{inc_a} \\ p_{inc_b} \end{Bmatrix} \tag{22}$$

This equation forms the basis for the treatment of the fluid-structure interaction in a unified fashion. The solution vector consisting of the displacement vector of the structure and total acoustic pressure on the boundaries of the acoustic domain, including the acoustic-structure interface, is represented by $\{x\ p_a\ p_b\}^T$, while $\{0\ p_{inca}\ p_{incb}\}^T$ is the acoustic excitation vector. Due consideration should be given to **L**, where **L** is a coupling matrix of size $m \times n$, $m$ is the number of FE degrees of freedom and $n$ is the number of BE nodes on the coupled (interface) boundary $a$.

The detail of the Finite Element Formulation of the structure has been elaborated in previous work [4], where four node quadrilateral elements for shell modeling have been be utilized.

The boundary integral equation of equation (4) fails at frequencies coincident with the interior cavity frequencies of homogeneous Dirichlet boundary conditions [16]. For exterior problem, these frequencies correspond to the natural frequencies of acoustic resonances in the interior region. When the interior region resonates, the pressure field inside the interior region has non-trivial solution. Since the interior problem and the exterior problem shares similar integral operators, the exterior integral equation may also break down. The discretized equation of the [H] matrix in equation (14) becomes ill-conditioned when the exciting frequency is close to the interior frequencies, thus providing an erroneous acoustic loading matrix. This problem could be overcome by using the CHIEF[16][17], Burton Miller method[18][19], and a recent technique utilizing SVD and Fredholm alternative theorem[19] [20].

For the purpose of this study, such problem can be avoided by limiting the problem only for low frequency range; further treatment for this problem will be elaborated in a separate paper.

Several cases are considered to validate the program as well as to evaluate its performance.

## 4. Generic Problem I - Numerical Simulation of Flat Plate Subject to Acoustic Excitation in an Infinite Medium

As the first example, acoustic-structure interaction on a flat plate will be treated using the present method, which may typically be applicable to acoustic pressure effects on solar panel during launch. For the purpose of this study, the dimension of the flexible structure is 450 x 150 x 5 mm. The material properties for flat plate made of AISI 4130 Steel is as follows: Modulus of Elasticity, E = 29 x $10^6$ N/m$^2$, Shear Modulus, G = 11 x $10^6$ N/m$^2$, Poisson's Ratio $\upsilon$ = 0.32 and density, $\rho$ = 7.33145 x $10^{-4}$ Kg/m$^3$

The flexible structure is now modeled with Finite Element and the surrounding boundary is represented by a quarter space is modeled using Boundary Elements; typical boundary element discretization of the surface, and the finite element discretization of the plate, is exhibited in Fig. 4.

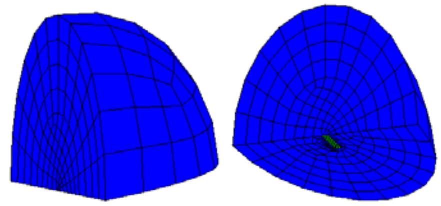


Fig.4 FEM-BEM discretization

The monopole acoustic source is placed at the center of the plate, at a distance 0.1 m above it.

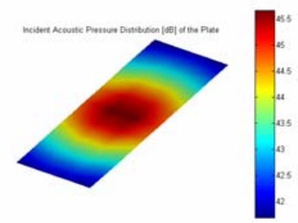



Fig.5 Incident pressure distribution [dB] due to monopole acoustic source as an acoustic excitation on flexible structure

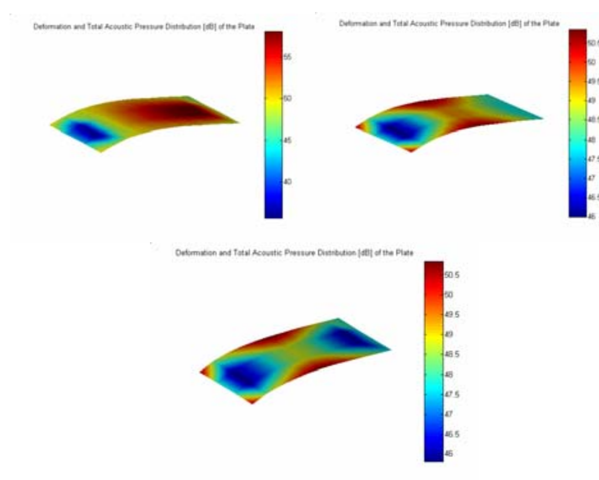



Fig. 6 Deformation and total acoustic pressure response [dB] on flexible structure for plate thickness 5, 10 & 20 mm (top to down)

For illustrative purposes, the frequency of the monopole source is assumed to be 10 Hz and the fluid medium is air with density $\rho$ = 1.21 kg/m$^3$ and sound velocity c =340 m/s. The incident pressure distributions are depicted in Fig.5 and the total acoustic pressure distribution is shown in Fig.6.

## 5. Generic Problem II - Submerged Spherical Shell Subject to External Forces

Numerical examples are presented by selecting constant thickness spherical shell under concentrated external forces. The material data for the spherical shell are listed as follows: the radius of the shell is 1 m, the thickness of the shell is 003 m, Young's modulus is 2.07 x $10^{11}$ Pa (N/m$^2$), the Poisson ratio is 0.3, the density of the shell and water are 7669 and 1000 $kg/m^3$, respectively, and the sound speed of the water is 1524 m/s. An external concentrated alternating force is exerted at one apex.

### *5.1. Normal Mode Analysis*

The modal analysis of the spherical shell to obtain the first ten normal modes using finite element program developed in MATLAB® has been carried out and the discretization is shown in Fig. 7. The first six eigen frequencies is the rigid body mode of the spherical shell and the results using the present routine are compared to those obtained using commercial package NASTRAN®. As exhibited in Table 1, good agreement is obtained.

Table.1 First ten eigen frequencies for spherical shell

| Mode | Natural Frequency (Hz) | |
|---|---|---|
| | MATLAB® | NASTRAN® |
| 1 | 1.5630e-05 | 2.7326e-05 |
| 2 | 8.0923e-05 | 1.9468e-05 |
| 3 | 1.3454e-05 | 9.8599e-05 |
| 4 | 2.1060e-05 | 2.0589e-05 |
| 5 | 2.5153e-05 | 1.8257e-05 |
| 6 | 3.9694e-05 | 3.8189e-05 |
| 7 | 605.44 | 604.69 |
| 8 | 607.48 | 606.80 |
| 9 | 609.08 | 608.91 |
| 10 | 610.30 | 610.33 |

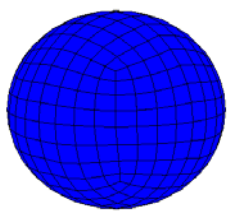


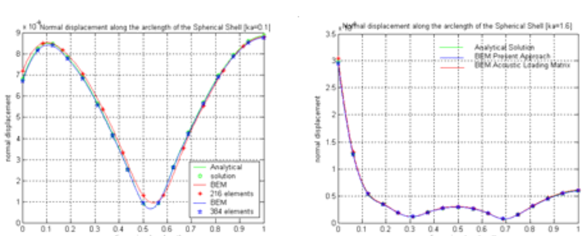
Fig. 7 Discretization of spherical shell

### *5.2. Coupled Analysis*

Because the response is axisymmetric, we plot the normal displacement along the arclength of a generator which generates the surface of the sphere by revolving around a symmetric axis and the results compare the present method with the analytical solutions.

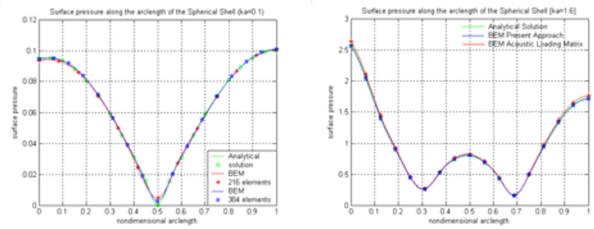
Fig. 8 compares the normal displacement for the present method for two different discretization with the analytical solutions [21] where the apex correspond to zero arclength for non dimensional frequencies ka 0.1 and 1.6.

Fig. 9 also compares the present method for two different discretization with the analytical solutions for non dimensional frequencies ka 0.1 and 1.6.



(a) (b)

Fig. 8 Normal displacement along the arclength of the spherical shell: a) ka = 0.1 and b) ka = 1.6

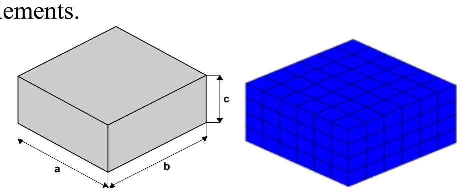


(a) (b)

Fig. 9 Surface pressure along the arclength of the spherical shell: a) ka = 0.1 and b) ka = 1.6

## 6. Generic Problem III - Flexible Structure Subject to Acoustic Excitation in a Confined Medium

Application of the method to another example is carried out for a box shown in Fig. 10a with a dimension of a × b x c = 450 × 450 × 270 mm. Each sides of the box are modeled as the flexible structure and assumed to be made of 1 mm aluminum plate, and is modeled as coupled boundary and finite elements.




(a) (b)

Fig. 10 Generic flexible structure typical of space-structure subjected to monopole acoustic source in an acoustic medium

This box structure is subjected to an acoustic monopole source at the center of the box and the acoustic medium is air with the following properties: density, $\rho$ = 1.21 $kg/m^3$ and sound velocity c =340 m/s. The discretization of the box is also depicted in Fig.10b.

### *6.1. Normal Mode Analysis*

The modal analysis of the box to obtain the first ten normal modes using finite element program developed in MATLAB® has been carried out. The first six eigen frequencies is the rigid body mode of the box and the results using the present routine are compared to those obtained using commercial package NASTRAN®. As exhibited in Table 2, good agreement is obtained.

Table.2 First ten eigen frequencies for box modeling with shell element

| Mode | Natural Frequency (Hz) | |
|---|---|---|
| | MATLAB® | NASTRAN® |
| 1 | 1.5630e-05 | 2.7326e-05 |
| 2 | 8.0923e-05 | 1.9468e-05 |
| 3 | 1.3454e-05 | 9.8599e-05 |
| 4 | 2.1060e-05 | 2.0589e-05 |
| 5 | 2.5153e-05 | 1.8257e-05 |
| 6 | 3.9694e-05 | 3.8189e-05 |
| 7 | 17.644 | 16.190 |
| 8 | 36.026 | 35.986 |
| 9 | 41.072 | 41.520 |
| 10 | 44.585 | 44.787 |

### *6.2. Acoustic Excitation*

Acoustic excitation due to an acoustic monopole source with initial frequency, f = 10 Hz, is applied at the center of the box. No other external forces are applied. The resulting distribution of the incident acoustic pressure is shown in Fig. 11a.

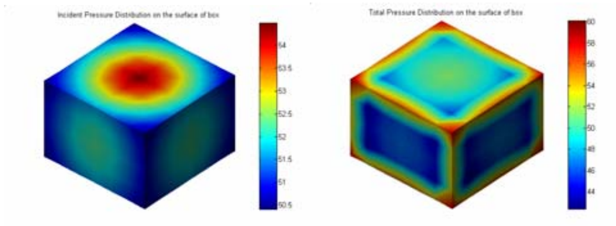



Fig. 11 (a) Incident acoustic pressure distribution on the surface of box due to monopole acoustic excitation and (b) Total acoustic pressure distribution on the surface of box due to monopole acoustic excitation

The total pressures on the surface of the box obtained from the computational results are shown in Fig.11b. The frequency response for the incident pressure and total pressure on the center top surface of the box computed using Eq. (22) is shown in Fig. 12.

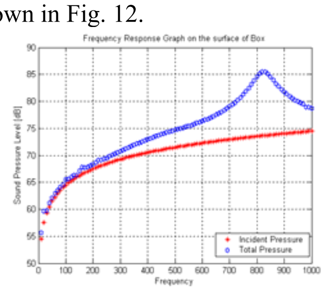



Fig. 12 Incident and total acoustic pressure distribution on the center top surface of box as a function of frequency

A few remarks are in order:

1. Although the flat plate problem is vary basic in nature, the validation of the computational procedure for solving combined excitation due to acoustic and external forces on structural problem formulated as coupled FEM-BEM equation has been validated by the application of the method to two generic cases: box and sphere.
2. The validation of the method for the generic problem II which deals with box structure, has earlier been validated by comparison of the results with those of ref.[6], for a simplified case, at which only the upper surface is a flexible structure. The results are elaborated in [5]. The present results may serve as benchmarking.
3. The application of the method to the acoustic-structure interaction on a sphere has been validated by comparison with analytical solution [21] as well as other numerical solution using acoustic loading matrix[7]. These results, in addition to serving to validate the present method, gives confidence in the applicability of the method. The result for the generic flat plate problem problem can be viewed with such perspective.

## 7. Conclusions

Method and computational procedure for BEM-FEM Acousto-Elasto-Mechanic Coupling which has been developed, has been validated by reference to classical examples or others available in the literature, and has been applied to generic problems typically found on spacecraft structures. The applicability of the method for analyzing typical problems encountered in space structure has thus been demonstrated. Refinements of the method for fast computation and for complex geometries are in progress. Such steps may be useful for acoustic load qualification tests as well as structural health monitoring.

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