

2 **Prediction of High-frequency Vibro-acoustic Coupling in** 3 **Anechoic Chamber Using Energy Finite Element Method** 4 **and Energy Boundary Element Method**

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6 **Abstract:** Energy finite element method(EFEM) is a promising method to solve
7 high-frequency vibro-acoustic problem. Energy boundary element method (EBEM)
8 is an effective way to compute high-frequency sound radiation in the unbounded
9 medium. Vibro-acoustic coupling of cavity structure in anechoic chamber includes
10 both the interior acoustic field and unbounded exterior acoustic field. In order to
11 predict this kind of high-frequency vibro-acoustic coupling problem in anechoic
12 chamber, an approach combined EFEM and EBEM is developed in this paper. As
13 a numerical example, the approach is applied to solve the high-frequency vibro-
14 acoustic coupling response of a cubic cavity structure excited by a point sound
15 source in an anechoic chamber. The result shows the approach is of the feasibility.

16 **Keywords:** Energy finite element method; Energy boundary element method;
17 anechoic chamber

18 **1 Introduction**

19 A series of tests for an aircraft must be performed on ground to ensure aircraft reli-
20 ability before using. Testing of the acoustic environment is a very important part of
21 this verification. Aircraft during flight can be considered as being in a free sound
22 field, which can be simulated in an anechoic chamber on ground. For most ane-
23 choic chamber acoustic tests, test pieces are placed in the anechoic room, a sound
24 source is placed at another location away from the test pieces, and the internal and
25 external acoustic field characteristics and test piece vibration are measured. These
26 experiments often have to be repeated multiple times to obtain reproducible results.
27 Alternatively, testing time and cost can be reduced significantly with a proper nu-

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merical simulation test program. Therefore the study of anechoic chamber simulation methods is very meaningful. Finite element method (FEM) and boundary element method (BEM) [Brancati and Aliabadi (2012); Brancati et al. (2011)] is quite mature methods to deal with the vibro-acoustic problems in anechoic chamber. However, FEM and BEM are not always efficient and accurate, especially for high frequency ranges, since structural response is extremely sensitive to material and geometry details [Bitsie (1996)]. In order to capture the structural characteristic length, a much smaller mesh size is necessary for FEM or BEM models, which results in the procedure being computationally expensive or even prohibitive. Statistical energy analysis (SEA) is an alternative efficient method which is suitable for solving high-frequency problems and has been widely used as an analysis tool in practice. But local modeling details that concern designers are usually ignored in SEA due to the fact that SEA is based on the division of sub-structures.

The energy finite element method (EFEM) presented by Nefske *et al* [Nefske and Sung (1989)] is a new tool for high-frequency vibro-acoustic coupling analysis. In this method, governing differential equations are derived in terms of energy density variables which are solved by applying the finite element approach. Compared with SEA, EFEM can obtain the detail vibro-acoustic response information. It is a promising effective method to high-frequency vibro-acoustic prediction. This method can compute the high-frequency acoustic field generated from structural vibration. The radiation efficiency in EFEM determines the amount of acoustic power radiated in the fluid due to structural vibration and the total amount of the radiated acoustic power can be computed by integrating the acoustic intensity over all the structural elements which are in contact with the fluid. However, EFEM is not able to compute the acoustic field at unbound medium since the governing difference equation for acoustic space is based on plane wave approximation, and particularly valid for highly reverberant acoustical environments at high frequency.

Energy boundary element method (EBEM), which was developed along with EFEM, is an effective way to compute high-frequency sound radiation in the unbounded medium of an external structure. From EFEM analysis, the acoustic power radiated from each structural element is obtained. Since the vibration energy in the structural elements and the acoustic intensity radiating from structural elements in contact with the fluid are incoherent among different elements, the incoherent acoustic intensity on the interface between a structure and an acoustic cavity can be regarded as a boundary condition in high-frequency EFEM interior acoustic computations. Similarly, the incoherent acoustic intensities radiating from each element on the outer structure surface in the unbound medium comprise the boundary conditions for EBEM computations [Wang et al. (2004)]. Therefore, the acoustic field at specific field points around the structure can be computed by EBEM.

instead of
"are"

In 1996, an indirect boundary element method coupling structural vibration and acoustic field was presented by Bitsie [Bitsie (1996)], in which structural vibration was computed by EFEM and acoustic field was calculated with EBEM. In 2004, Wang [Wang et al. (2004)] presented an EBEM formulation for calculating sound radiation at high frequency from a radiator with an arbitrary shape. In 2004, Choi and Dong [6-8] evaluated the exterior acoustic field of a vehicle excited by force as ~~a way of~~ combining EFEM and EBEM. In 2006, Nicklos and Zhang [Vlahopoulos and Wang (2008)] studied the structural vibration and interior acoustic field excited by an exterior acoustic source. In 2007, Raveendra and Hong [Raveendra and Zhang (2007); Hong et al. (2007)] performed an analysis of interior acoustic and vibration characteristics of a simplified airplane cabin and van models using hybrid EFEM/EBEM. **instead of "and"**

For the cavity structure, the vibro-acoustic response includes both the exterior acoustic field and the interior acoustic field. If the exterior acoustic field is closed by absorbent material, means that it is a bounded space, then EFEA can be used to solve the interior and exterior acoustic field simultaneously by viewing the absorbent material as a boundary condition [Vlahopoulos and Wang (2008)]. However, in anechoic chamber, exterior acoustic field is in an unbounded medium, so the EBEA and EFEA must be combined to deal with this kind of problem.

In this paper, we study the interior and exterior acoustic fields of a cavity structure excited by an exterior acoustic source in an anechoic chamber using EFEM/EBEM.

~~In reference [Vlahopoulos and Wang (2008)], Coupling of the cavity structure vibration with interior acoustic field and exterior acoustic field. In this method, the anechoic chamber is taken as an unbounded acoustic medium, and the vibroacoustic system consist of interior medium, cavity structure and exterior medium. Coupling between the interior acoustic field and the cavity structure vibration is computed by EFEM. Excitation imposed on the structure come from a point sound source placed in exterior medium. After this EFEA coupling analysis, the energy radiated from each structural element is determined, which is took as the source of EBEA to calculate the exterior acoustic field. As an example, a cubic shell excited by a point sound source in an anechoic chamber is studied to illustrate the feasibility. "~~

~~lowed through. The energy radiated from each structure element becomes the boundary condition of a second EBEM, giving the exterior acoustic field generated by structural vibration. Then, the method solving vibro-acoustic coupling of the whole structural-acoustic system in anechoic chamber is presented. As an example, a cubic shell excited by a point sound source in an anechoic chamber is studied to illustrate the feasibility.~~

2 Basic theory of high-frequency structure-acoustic response

As stated in introduction, an approach combined EFEM and EBEM is developed to solve the anechoic chamber high-frequency vibro-acoustic coupling problem. ~~Then the basic theory of EFEM and EBEM is introduced.~~

as following

2.1 Basic theory of EFEM

The governing equations for the structural-acoustic coupling ~~a complex~~ system are

$$\begin{aligned}
 & -\frac{C_{gB}^2}{(\eta_{sB} + \eta_{rad}) \omega} \nabla^2 \langle \bar{e}_{sB} \rangle + (\eta_{sB} + \eta_{rad}) \omega \langle \bar{e}_{sB} \rangle = \langle \bar{\pi}_{sB} \rangle \\
 & -\frac{C_{gL}^2}{\eta_{sL} \omega} \nabla^2 \langle \bar{e}_{sL} \rangle + \eta_{sL} \omega \langle \bar{e}_{sL} \rangle = \langle \bar{\pi}_{sL} \rangle \\
 & -\frac{C_{gT}^2}{\eta_{sT} \omega} \nabla^2 \langle \bar{e}_{sT} \rangle + \eta_{sT} \omega \langle \bar{e}_{sT} \rangle = \langle \bar{\pi}_{sT} \rangle \\
 & -\frac{c_a^2}{\eta_a \omega} \nabla^2 \langle \bar{e}_a \rangle + \eta_a \omega \langle \bar{e}_a \rangle = \langle \bar{\pi}_a \rangle
 \end{aligned} \tag{1}$$

where C is the wave speed; subscripts B, L and T express the bending, longitudinal and transverse shear waves, respectively; subscript g means group speed; subscripts s and a are used to denote the structure domain and the acoustic domain; η is the damping loss factor; η_{rad} is the radiation damping; ω is the radian frequency of the harmonic excitation; $\langle \bar{e} \rangle$ is the ~~wavelength of energy density~~; and $\langle \bar{\pi} \rangle$ is the ~~time average over a period and space average over~~

The finite element method is used to numerically solve the governing equations. ~~At the junctions between elements, the energy density is discontinuous while the power flow is continuous and is represented by~~

$$\left(\begin{bmatrix} K_s & 0 \\ 0 & K_a \end{bmatrix} + J_{ss} + J_{sa} \right) \begin{pmatrix} E_s \\ E_a \end{pmatrix} = \begin{pmatrix} F_s \\ F_a \end{pmatrix} \tag{2}$$

K_s and K_a are the system matrices for the structural elements and acoustic elements, J_{ss} and J_{sa} are joint matrices between discontinuous junctions, $\{E\}$ is the vector of nodal energy density, and $\{F\}$ is the vector of input power.

2.2 Basic theory of EBEM

Energy density is defined as $e = \frac{1}{2} \left(\rho v^2 + \frac{p^2}{\rho c^2} \right)$ and the acoustic intensity as $I = \frac{1}{2} \text{Re}(pv^*)$ where p is acoustic pressure, v is vibration velocity of the particle, ρ is the acoustic medium's mass density, c is the transmitted velocity of the wave in the medium, Re is expressed as the real part of the number, and $*$ denotes conjugation.

The ensemble averaging operator is applied to the ~~equation for~~ ^{expression of} e and I , resulting in

$$\begin{aligned}\langle \bar{e} \rangle &= \int_S \sigma(p) \left(\frac{\rho}{64\pi^2 r^4} + \frac{k^2 \rho}{32\pi^2 r^2} \right) dS \\ \langle \bar{I} \rangle &= \int_S \sigma(p) \frac{k^2 \rho c}{32\pi^2 r^2} \vec{E}_r dS\end{aligned}\quad (3)$$

where $\sigma(p)$ denotes ~~the strength density of the energy source placed at point p on the surface of the structure model~~ ^{the source strength density at point p on the sturcture surface}, r is stated as the distance between the field point and point p , and k is ~~wavenumber~~ ^{wavenumber}. The surface ~~structure surface S~~ ^{structure surface S} is divided into n quadrilateral or triangular elements and the source strength density σ_j on each element is considered to be constant. Eq.(3) can be written in the discrete form

$$\begin{aligned}\langle \bar{e}_Y \rangle &= \sum_{j=1}^n \left[\sigma_j(p) \int_{S_j} \left(\frac{\rho}{64\pi^2 r(\xi, Y)^4} + \frac{k^2 \rho}{32\pi^2 r(\xi, Y)^2} \right) dS \right] \\ \langle \bar{I}_Y \rangle &= \sum_{j=1}^n \left[\sigma_j(p) \int_{S_j} \frac{k^2 \rho c}{32\pi^2 r(\xi, Y)^2} \vec{E}_r dS \right]\end{aligned}\quad (4)$$

where s_j is the surface of the j th element and ξ is a point of the j th element. Then the energy density expression in Eq.(4) can be written in matrix form

$$[K] \{\sigma\} = \{p\} \quad (5)$$

where ~~re~~ ^{where} $\{P\}_{n \times 1} = \{\bar{p}_1 \ \bar{p}_2 \ \cdots \ \bar{p}_n\}^T$ is ~~the specified bound~~ ^{acoustic power} vector, $\{\sigma\}_{n \times 1} = \{\sigma_1 \ \sigma_2 \ \cdots \ \sigma_n\}^T$ is an unknown variables vector, and ~~each term~~ ^{each term} K_{ij} of the matrix $[K]$ is solved by

$$K_{ij} = \begin{cases} \int_{S_i} \left(\int_{S_j} \frac{k^2 \rho c}{32\pi^2 r(\xi, \eta)^2} \vec{E}_r \cdot \vec{n}_i dS \right) dS & i \neq j \\ A_i \frac{k^2 \rho c}{16\pi} & i = j \end{cases} \quad (6)$$

122 The source strength density ~~on~~ ^{of} each element is ~~obtained~~ ^{available} by solving Eq.(5). Then,
123 ~~by or~~ ^{Substituting the source strength density into Eq.(4), the acoustic energy density and intensity in}
124 ~~inter~~ ^{inter} exterior acoustic medium can be determined
125 ~~structure.~~

The acoustic power ~~P come from vibration energy density of structure. The energy density is obtained by EFEM. The~~
~~with the structural~~ ^{relation between acoustic power and the energy density of sturcture can be expressed as follow}
~~The acoustic power radiated from a structure immersed in an acoustic medium is~~
~~calculated by using of~~

$$p_i = \int_{S_j} \eta_{rad} \omega e_{sB} h dS \quad (7)$$

where η_{rad} is radiation damping, ω is the circle frequency and h is the thickness of the structure. The energy density e_{sB} on the nodes of the structure's surface and the acoustic power radiating from each element of a vibrating structure p_i is expressed by

$$P = WE_{sB} \quad (8)$$

The matrix W is comprised of NE rows and N columns, where each value is written as

$$W_{ij} = \int_{S_i} \Delta_{ij} \eta_{rad}^i \omega h dS \quad (9)$$

where $\Delta_{ij} = 1$ if node j is located on the i th boundary element, and $\Delta_{ij} = 0$ if node j is not on the i th boundary element. Eq.(5) can now be written as

$$[K] \{\sigma\} = [W] \{e_{sB}\} \quad (10)$$

2.3 Input power from an exterior acoustic source

In the anechoic chamber, the cavity structure is excited usually by exterior acoustic source. The input power must be got if we predict the coupling response of the interior acoustic field and the vibration of cavity structure by using of EFEA. The input power at each structural element which is in contact with the exterior acoustic medium is provided by the information of energy density radiated by acoustic source and the transmission coefficient from acoustic field to structure.

The energy density of any acoustic field generated by m point acoustic sources is

$$e_{Y0} = \sum_{i=1}^m \left[\bar{\sigma}_i \left(\frac{\rho_0}{64\pi^2 r_i^4} + \frac{k^2 \rho_0}{32\pi^2 r_i^2} \right) \right] \quad (11)$$

σ_i is the source strength of the i th acoustic source. The incident power π_{inc} on each energy finite element is determined from the exterior acoustic energy density e_{Y0} using a plane wave approximation,

$$\pi_{inc} = e_{Y0} c_0 A \quad (12)$$

where A is the area of the structure EFEM element and c_0 is the wave speed in the acoustic medium. The power transmitted to a plate due to an impinging plane wave is evaluated as

$$\tau_{as} = \frac{\rho_0 c_0^3}{\rho_s c_B^2 h f} \sigma_{rad} \quad (13)$$

By using Eq. (12) and (13), the input power applied on an EFEM element due to the external acoustic ~~energy density~~ **source** can be written as

$$\pi_{in} = \tau_{as} \pi_{inc} = \frac{\rho_0 c_0^4 A}{\rho_s c_B^2 h f} \sigma_{rad} e_{Y0} \quad (14)$$

3 Numerical example

3.1 Models

A cubic cavity structure model comprised of six plates, as shown in Fig.1, is constructed and studied. The dimensions of the cubic cavity structure are 1m x 1m x 2m and the thickness of the plates is 0.01m. The plates all have the same physical properties: the Young's modulus is 2.4e11Pa, Poisson's ratio is 0.30, Mass density is 7800Kg/m³ and the hysteresis damping factor is 0.01. The cubic cavity structure is immersed in air and the acoustic space enclosed by the plates is filled with air with mass density of 1.225Kg/m³, acoustic speed of 343m/s and hysteresis damping factor of 0.001. The distance between the center of the cubic cavity and point acoustic source is 5.5m. The location of the point acoustic source in relation to the cubic cavity structure is shown in Fig.1. Fig.2 shows the energy finite element model for the structure. The point acoustic source strength is unit strength. The response at several frequencies (2000Hz, 8000Hz 16000Hz and 32000Hz) is predicted.

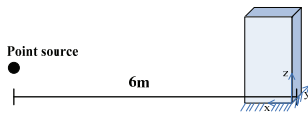


Figure 1: Schematic diagram of the point source and the structure

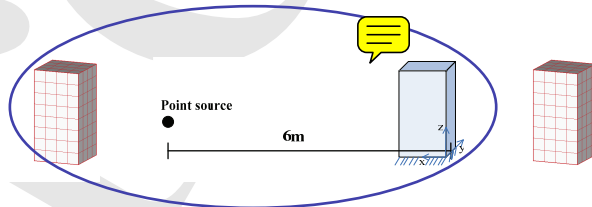


Figure 2: EFEA model for the structure

3.2 Results and discussion

Acoustic energy density is converted to the root mean square acoustic pressure with the approximation $p = c_0 \sqrt{e \rho_0}$ and expressed in terms of decibel level using the acoustic pressure reference of $2 \times 10^{-5} Pa$. In the case of 2000Hz, the exterior acoustic pressures generated by structural vibration excited by the point source are shown in Fig.3. The distribution of interior acoustic pressure in the acoustic medium, the distribution of interior acoustic pressure at the y=0.5 symmetric plane,

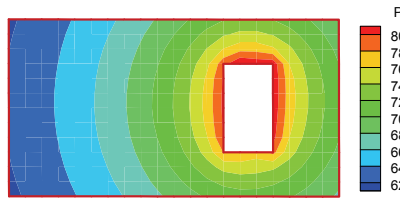


Figure 3: Exterior acoustic pressure generated by structural vibration

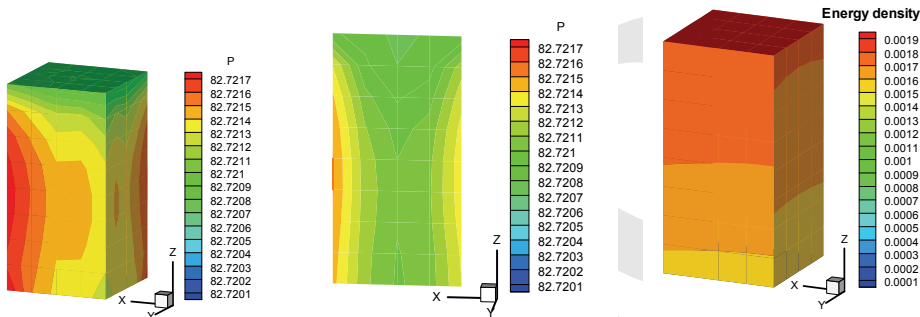


Figure 4: Distribution of interior acoustic pressure

Figure 5: Distribution of interior acoustic pressure at the $y=0.5$ symmetric plane

Figure 6: Distribution of energy density on the plate

and the computed energy density distribution on the shell are shown in Fig.4, Fig.5 and Fig.6, respectively.

In the case of 8000Hz, the corresponding results are shown as follow. in Fig.7-Fig.10

In the case of 16000Hz, the corresponding results are shown as follow. in Fig.11-Fig.14

In the case of 32000Hz, the corresponding results are shown as follow. in Fig.15-Fig.18

The results of the approach of combined EFEM and EBEM on the above model show that:

1. With the increasing of frequency, the difference between maximum and minimum pressure inside the cubic cavity structure become small. The exterior pressure and the vibration energy get smaller and smaller as the frequency increase.
2. The variation of acoustic pressure in the acoustic medium enclosed by the plate is very small as energy propagation and decay makes the , but the distribution trend is

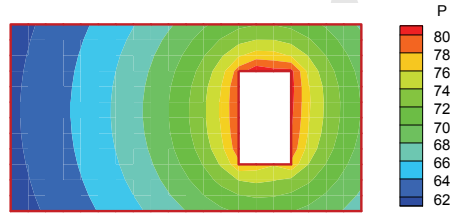


Figure 7: Exterior acoustic pressure generated by structural vibration

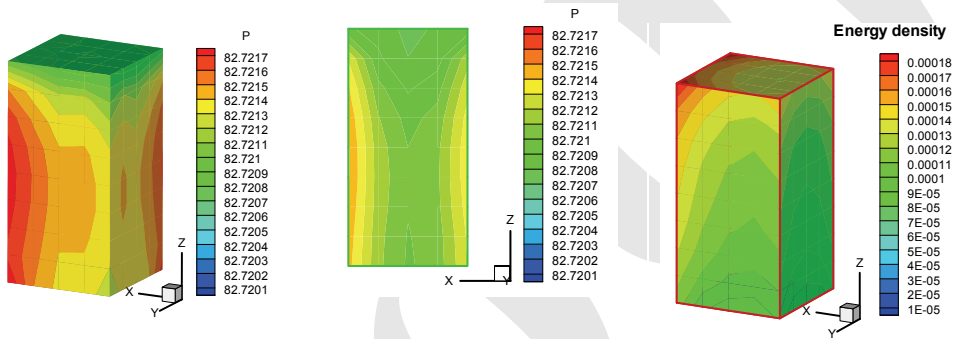


Figure 8: Distribution of interior acoustic pressure

Figure 9: Distribution of interior acoustic pressure at the $y=0.5$ symmetric plane

Figure 10: Distribution of energy density on the plate

sure diminish from the boundary to the centre. Explanation for this is that energy is transmitted from the shell to the centre of the acoustic medium since the acoustic pressure of the plate facing the point source is larger than the acoustic pressure on the plate on the far side of the cubic cavity structure from the point source.

3. Energy density on the four lateral plates increases from the bottom to the top of the cubic cavity structure because the bottom plate is fixed. However, the contour is not parallel to the bottom plate; the energy density of the plate which is closer to the load location is large.

4. If a sound absorbing material is utilized, EBEA don't suit any more. In this case, EFEA is adapted, and the absorbing material is taken as boundary element in EFEA which is similar with the method in the paper of Brancati, A.

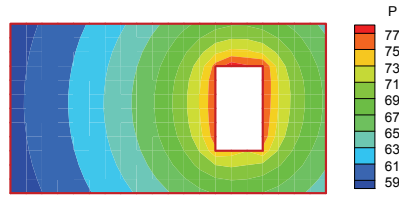


Figure 11: Exterior acoustic pressure generated by structural vibration

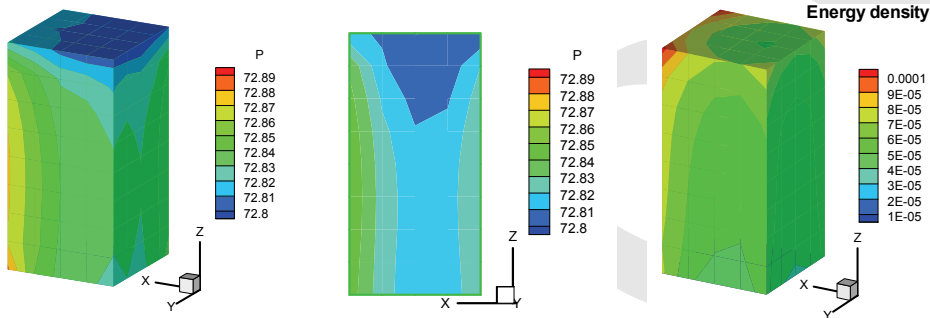


Figure 12: Distribution of interior acoustic pressure

Figure 13: Distribution of interior acoustic pressure at the $y=0.5$ symmetric plane

Figure 14: Distribution of energy density on the plate

3.3 Comparison of EFEA with SEA and FEA

In order to prove that the hybrid method of EFEA and EBEA is the more suitable method to deal with the high frequency vibro-acoustic in anechoic chamber, comparison of EFEA with SEA and FEA has been done in terms of accuracy and calculating time.

3.3.1 Comparison of EFEA with FEA

In FEA, the biggest factor affecting the computation time is the number of elements, so the number of elements can roughly determine the computation time. For the cuboid, according to the accuracy requirement, the required number of elements is shown as table 1. It can be seen that there need more and more elements as the frequency increasing.

However, in EFEA, the elastic wave propagation problem in EFEM is similar as the heat transfer problem in FEM. So, there is no need to change the mesh size with frequency. Frequency need not to be considered in EFEM mesh models. So more

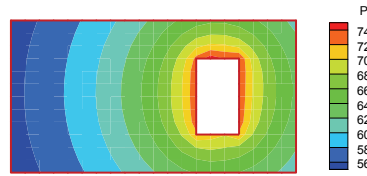


Figure 15: Exterior acoustic pressure generated by structural vibration

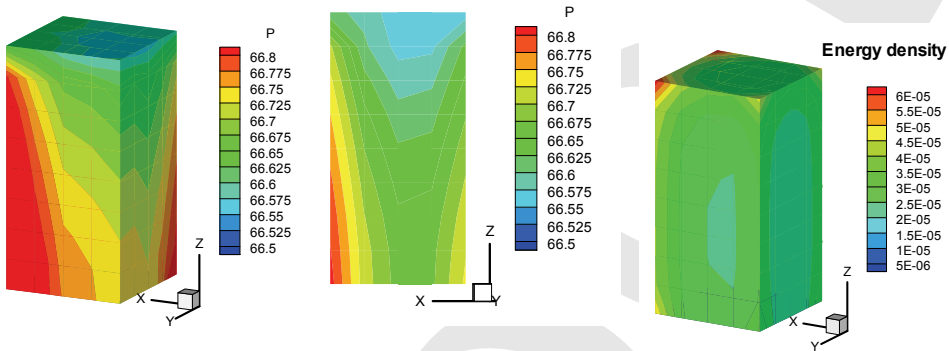


Figure 16: Distribution of interior acoustic pressure

Figure 17: Distribution of interior acoustic pressure at the $y=0.5$ symmetric plane

Figure 18: Distribution of energy density on the plate

Table 1: Required number of elements in FEA

Frequency	element of FEA
2000Hz	3240
4000Hz	6656
8000Hz	12816
16000Hz	25806

195 high frequency, more convenient energy finite element analysis is.

196 3.3.2 Comparison of EFEA with SEA

197 In order to make clear the advantage of EFEA over SEA, the cubic cavity structure
 198 excited by exterior acoustic source is calculated by using of SEA. At 2000Hz, the
 199 results are shown in Fig.19.

200 From the Fig.19, the vibration energy of the top plate and the front plate is $1.712 \times$

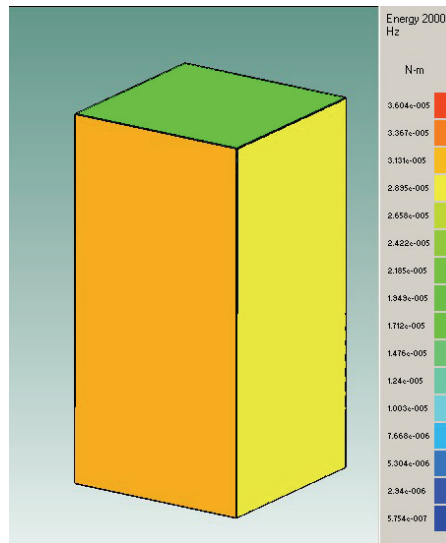


Figure 19: The distribution of energy calculated by SEA

201 $10^{-5}J$ and $3.131 \times 10^{-5}J$, respectively.

202 From the Fig.6, the energy density calculated by EFEA is obtained. In order to
 203 compare the result of EFEA to SEA, the energy density is converted to energy
 204 by the way of multiplying energy density by corresponding volume. Thus, the
 205 vibration energy of the top plate and the front plate calculated by EFEA is $1.700 \times$
 206 $10^{-5}J$ and $3.000 \times 10^{-5}J$, respectively. It can be seen that EFEA and SEA has
 207 approximate results, however, the distribution of energy density on structure can be
 208 obtained by EFEA.

209 4 Conclusion

210 Considering a structure excited by a point sound source in an anechoic chamber,
 211 the high frequency vibro-acoustic response is hard to reach with conventional finite
 212 element analysis because of the computational cost. EFEA based on energy conser-
 213 vation is effective. ~~Continued the method in the reference [Vlahopoulos and Wang~~
 214 ~~(2004) An~~ a approach solving the anechoic chamber high-frequency vibro-acoustic
 215 coupling problem using EFEA and EBEM is presented in this paper. The anechoic
 216 chamber is taken as an unbounded acoustic medium. The exterior/interior acoustic
 217 fields of the structure and the vibration energy density distribution of the structure
 218 within an anechoic chamber are obtained using an EFEM/EBEM developed in this
 219 paper. As an example, a cubic shell excited by a point sound source in an anechoic

chamber is studied. The result actually is reasonable and indicates that it is feasible and convenient to solving vibro-acoustic coupling of the whole structural-acoustic system in anechoic chamber using EFEM/EBEM proposed in this paper. It is to believe that the method described here can accurately simulate anechoic chamber high-frequency vibration-acoustic test results and provide an alternative to chamber testing of structural components.

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