ISSN 1392-2114 ULTRAGARSAS (ULTRASOUND), Vol.62, No.4, 2007

Development of the theoretical model of acoustic field on the basis of FEM and analysis of effectiveness*

R. Mikalauskas, V. Volkovas

Technological System Diagnostic Institute, Kaunas University of Technology

Address: Kestucio str. 27, Kaunas 3004, Lithuania; Tel/Fax:+370 7 323 720; e–mail: romika@ktu.lt

Abstract

This work presents the development model of the acoustic field imitating the real operation conditions of an industrial premise. The model is based on the FEM methodology. The effectiveness and possibilities of its application to a mobile system reducing the noise are analyzed and the received results of the theoretical experiment are compared with the experimental ones.

Key words: acoustics, FEM, modeling, passive control.

Introduction

The analysis of the suppression problem of acoustic noises shows that a number of experimental and theoretical investigations have been carried out with regard to this topic in the last decades. There are two main ways to control and reduce the acoustic noises – the active and the passive modes.

One of the tools in the passive control of the acoustic noise is the mobile noise absorption systems, which are essentially made of various noise suppression partition walls, curtains, shields, etc. These systems are used to reduce the noise in an open space and in premises. The reduction tools of the acoustic noise, used in closed spaces are effective in the case the direct sound field is prevalent with regard to the reflected field; however the same acoustic partition walls suppress the reflected sound field partly, as well. Generally the sound interacts with the obstacle in three ways: a) part of it is reflected from the obstacle, b) part of the sound penetrates through the obstacle, and c) its other part evades the obstacle, in other words, the diffraction appears.

The effectiveness of the tools used to reduce the noise depends on a number of factors, first of all including the sources, which create the acoustic field. There are a lot of sources in industrial premises, they have different on the field, and they can change when the environmental conditions are changing. Thus the task to create the mobile tools reducing the acoustic noise, to arrange them rationally and to manage them is becoming more and more topical. When the management systems of the acoustic noises are being created, the theoretical modeling of their interaction with the environment cannot be avoided. A lot of bibliographical resources models the appearance of acoustic fields, character of its distribution in space and interaction with the obstacles, which are described using the numericall – finite elements (FE), edge elements (EE), finite differences (BS) – and analytical models.

One of the widely used methods to create the acoustic models is the method of finite elements. When this method is used (as well as when the method of finite differences is used), the wave equation is being solved (with regard to the boundary conditions) by dividing the space (in certain cases the time, too) into the elements. Then the wave equation is expressed by the discrete set of the linear equations for these elements. The method of the finite elements also allows modeling the energy transmission between the separate surfaces (funk-beam tracing). The advantage of this method [1-3] is that it allows linking directly the structural and acoustic media and evaluating their interaction under changing conditions of the modeled environment, which is extremely important for development of the systems of acoustic partitions. The results received with the help of this method while solving the three-dimensional tasks of the acoustic medium reflect completely the character of the acoustic field in the analyzed space. With regard to the disadvantages of this method, it is possible to state that under changing conditions of the modeled environment and excitation, the model has to be created a new, and this usually needs a lot of time [4]. Besides, the solvable tasks are formed in the area of low frequencies, which very limits the application of FEM for the solution of control problems of the acoustic fields. According to the carried out review of the literature about modeling on the basis of FEM, various acoustic tasks are solved with the help of this method – analysis of the acoustic characteristics of various media, modeling of acoustic partition systems, test of the sound propagation in various media, analysis of the interaction between the structural and acoustic medium, etc. The main advantage of this method if compared to the others is that it allows modeling a heterogeneous acoustic medium, evaluation of several excitation sources, and obtaining the complete character of the acoustic field in the analyzed space. This makes a certain basis for the modeling of acoustic fields. However the modeling of the acoustic field characterized by the heterogeneity and generated by various sources in the closed room with different acoustic characteristics needs additional investigations.

In order to reduce the acoustic noise in the passive mode by the mobile screens, the following tasks have to be solved:

- Development the model of the acoustic field generated by several sources in closed space on the basis of FEM;

*This work was supported by Research and Study Found of Lithuania

- To analyze the adequacy of this theoretical model to the real fields and application possibilities while constructing the mobile or controlled noise reduction systems

Model of acoustic field on the basis of FEM

The discrete acoustic wave equation that evaluates the interaction with a structural medium and losses is expressed in the following way when the finite elements are used:

$$\left[M_e^P\right]\{\ddot{P}_e\}+\left[C_e^P\right]\{\dot{P}_e\}+\left[K_e^P\right]\{P_e\}+\rho_0\left[R_e\right]^T\{\ddot{u}_e\}=0 \quad (1)$$

In order to describe fully the interaction of the acoustic medium and structure, the affect of pressure on a structure has to be evaluated. In such a way the main structural dynamic equation is the following:

$$\left[M_e\right]\{\ddot{u}_e\}+\left[C_e\right]\{\dot{u}_e\}+\left[K_e\right]\{u_e\}-\left[R_e\right]\{P_e\}=\{F_e\} \quad (2)$$

where: $\left[M_e^P\right]$, $\left[M_e\right]$ are the matrixes of the mass of acoustic medium and structure, accordingly,

$\left[C_e^P\right]$, $\left[C_e\right]$ are the damping matrixes of the acoustic medium and structure,

$\left[K_e^P\right]$, $\left[K_e\right]$ are the stiffness matrixes of the acoustic medium and structure,

$\rho_0\left[R_e\right]^T$ are the matrix of the connection between acoustic and structural media,

$\{P_e\}$ are the vectors of the pressure in nodes and its derivatives with regard to the time $\{\dot{P}_e\}$, $\{\ddot{P}_e\}$,

$\{\ddot{u}_e\}$ is the vector of the derivative of the nodal displacement, $\{F_e\}$ is the vector of load, ρ_0 is the density of air medium.

Eq. 1 and Eq. 2 describe completely the interaction of the structure and acoustic medium and are expressed in the following way:

$$\begin{bmatrix} [M_e] & [0] \\ \rho_0[R_e]^T & [M_e^P] \end{bmatrix}\begin{Bmatrix} \{\ddot{u}_e\} \\ \{\ddot{P}_e\} \end{Bmatrix}+\begin{bmatrix} [C_e] & [0] \\ [0] & [C_e^P] \end{bmatrix}\begin{Bmatrix} \{\dot{u}_e\} \\ \{\dot{P}_e\} \end{Bmatrix}+\ldots$$
$$+\begin{bmatrix} [K_e] & -[R_e] \\ [0] & [K_e^P] \end{bmatrix}\begin{Bmatrix} \{u_e\} \\ \{P_e\} \end{Bmatrix}=\begin{Bmatrix} \{F_e\} \\ \{0\} \end{Bmatrix} \quad (3)$$

When the theoretical model is formed, the FEM software ANSYS 10 was used. The analyzed two-dimensional model consists of the acoustic and structural media. In order to model them the elements FLUID29 and PLANE42 were used. While modeling the harmonic analysis was made, when the system was harmonically excited by two sources of certain pressure.

The excitation frequency, acoustic partition walls, absorption coefficients of the premise's ceiling were changed during the analysis, and the acoustic homogeneous and non-homogeneous field were analyzed. The obtained results of the numerical experiment are presented below. The distance from the excitation source to the partition wall was 2 m. The physical characteristics of the separate model's elements were the following: air density ρ=1.2 kg/m^3; velocity of acoustic wave's spreading c=335 m/s; the absorption coefficient of air sound μ=0; the density of the partition ρ=950 kg/m^3; the elasticity modulus of the partition E=2.3e+9 Pa; the sound velocity in the partition material c_p=1700 m/s; the sound absorption coefficient of the partition μ=0.1-0.9.

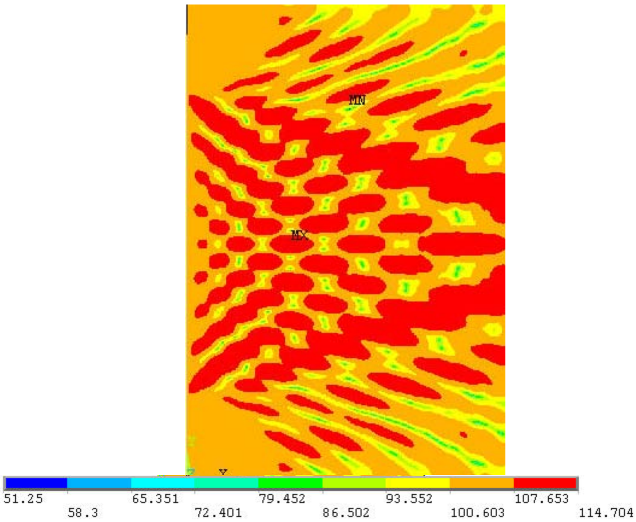

Fig. 1. **Level of acoustic pressure in a homogeneous field, when it is excited by L_p=107 dB, f=1000 Hz**.

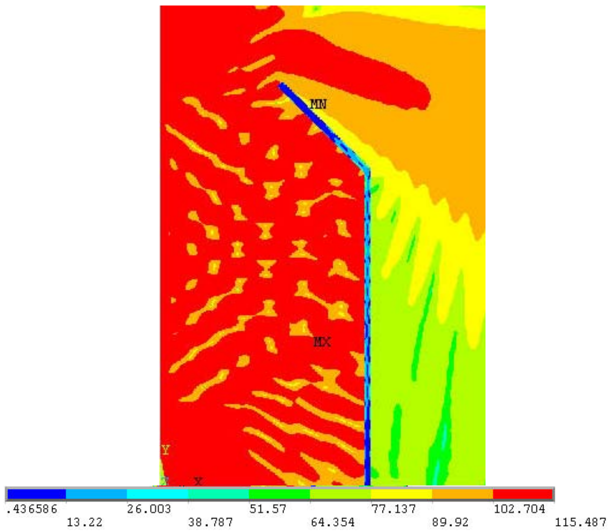

Fig. 2. **Level of acoustic pressure in a non-homogeneous field, when it is excited L_p=107 dB, f=1000 Hz, the sound absorption coefficient of the partition wall μ=0.9.**

The presented calculation results show that the level of the acoustic pressure in homogeneous and non-homogeneous acoustic fields under similar excitation conditions is different. In the case of acoustic partition, the level of the acoustic pressure decreases (Fig. 1 and 2). It is usual in practice to have more than one excitation source, frequencies of which are usually different. Fig. 3-5 present the results of the theoretical calculation in the case the excitation is done by two sources of different frequencies. According to the obtained results, the level of acoustic

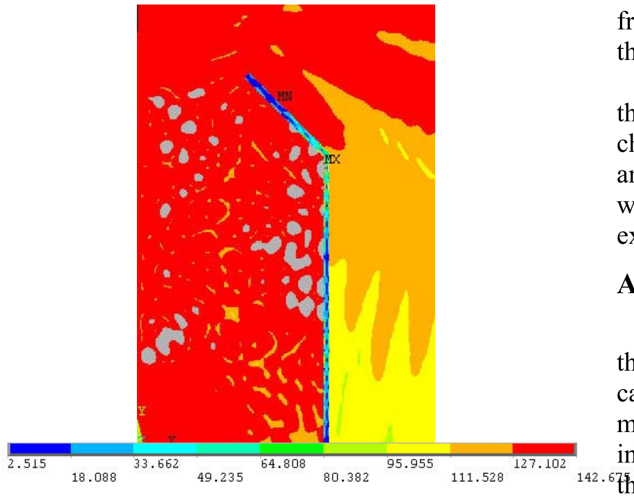

Fig. 3. **Level of acoustic pressure in the non-homogeneous field, when the excitation is done by two sources of different frequencies: L_{p1}=125 dB, f_1=1000 Hz, L_{p2}=97 dB, f_2=100 Hz, sound the absorption coefficient of the partition wall μ=0.2, ceiling's the absorption coefficient μ=0.1.**

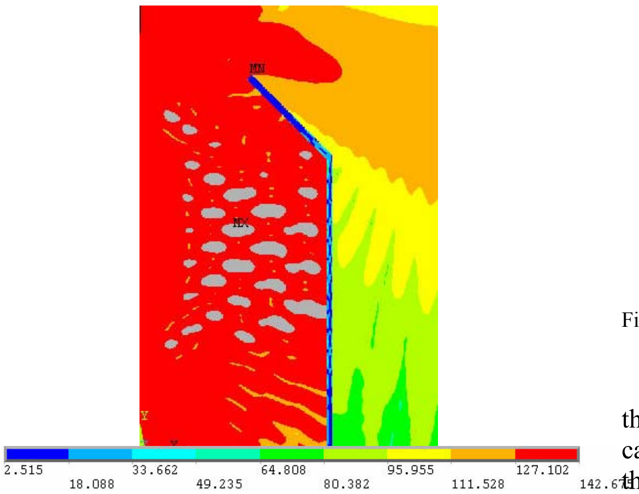

Fig. 4. **Level of acoustic pressure in the non-homogeneous field, when the excitation is done by two sources of different frequencies: L_{p1}=125 dB, f_1=1000 Hz, L_{p2}=97 dB, f_2=100 Hz, sound the absorption coefficient of the partition wall μ=0.2, ceiling's the absorption coefficient μ=0.9.**

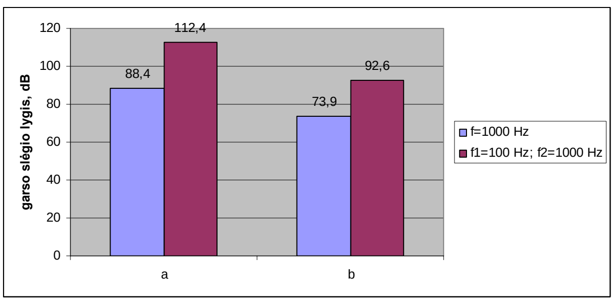

Fig. 5. **Level of acoustic pressure in the point 1. 2 m behind the partition wall and at the height of 1.5 m, in the case of different absorption coefficients by planes and excitation frequencies: *a* – partition wall's μ_1=0.2, ceiling's μ_2=0.1; *b* – partition wall's μ_1=0.9, ceiling's μ_2=0.9;**

field's pressure also changes in this case when the frequency of one source is low, and the other's is high, and the parameters of the reflection planes are changed.

To summarize, it is possible to state that the developed theoretical model on the basis of FEM defines the size and character of the pressure's level in the acoustic medium at any point of homogeneous and non-homogeneous field, as well as when the characteristics of reflection planes and excitation frequencies are changed.

Analysis of the theoretical model's efficiency

In order to analyze the efficiency of the development theoretical model, the experimental investigations were carried out and the obtained results of the theoretical modeling were compared to the experimental ones. The initial data of the theoretical investigation were selected through the imitation of the real experiment, which was done by company "SAPA Profiliai", where in order to reduce the acoustic noise in the workshop of the pipe cutting the acoustic partition walls were used. Fig. 6 shows the general view of this mounted partition wall.

Fig. 6. **General view of the acoustic partition wall in the workshop of aluminum pipe cutting**

In the real object the noise that appears during cutting the pipes was measured and its spectral analysis was carried out using the measurement equipment "Pulse" of the company Bruel & Kjaer. According to the range of acoustic pressure's level, the five dominant frequencies were distinguished, which values are presented in Table 1:

Table 1

Frequency, Hz	400	1250	3150	6300	8000
Level of acoustic pressure, dB	70,43	80,04	97,05	102,37	97,77

During the experiment the values of the acoustic pressure were measured behind the partition wall at the height of 1.5 m. The principal scheme of the measurement of acoustic pressure is presented in Fig. 7.

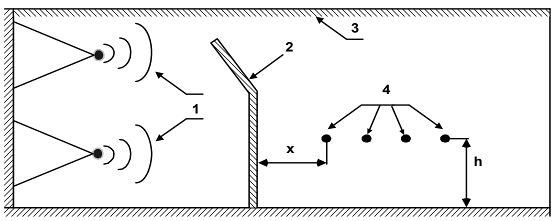

Fig. 7. **Principal scheme of the acoustic pressure's measurement: 1 – excitation sources, 2 – acoustic partition walls, 3 – reflection plane, 4 – measurement points**

In order to carry out the theoretical analysis of the acoustic noises in the pipe-cutting workshop, the above-described method on the basis of FEM was used together with the harmonic analysis, during which the excitation was performed harmonically by the determined values of the acoustic pressure corresponding to a certain excitation frequency. The obtained results of the theoretical and experimental tests are presented below.

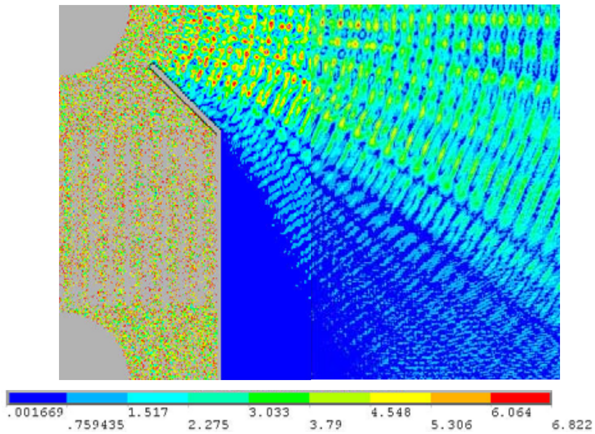

Fig. 8. **Acoustic pressure in one of the workshop's vertical sections in the area of the cutting machine in the presence of acoustic partition wall**

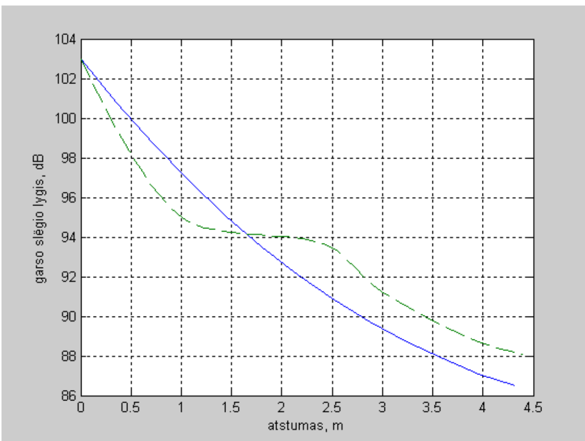

Fig. 9. **Dependency of the acoustic pressure's level on the distance to the acoustic partition wall: approximated experimental curve, — — the curve obtained with the help of theoretical model**

When the acoustic excitation, which appears in real operation conditions was imitated, the theoretical model helped to determine the distribution of the acoustic pressure in the area of cutting machine behind and in front of the partition wall (Fig. 8). The dependencies of the acoustic pressure on the distance to the partition wall (Fig. 9) obtained with the help of theoretical and experimental models show a quite good correspondence of the results.

To summarize, it is possible to state that using the theoretical model developed on the basis of FEM it is possible to model the acoustic excitation that appears in real conditions and to evaluate the effectiveness of the mobile system suppressing the noise.

Conclusions

The obtained results of the numeral experiment show that the suggested theoretical model developed on the basis of FEM is adequate to the real processes registered in industrial premises. The model allows modeling of mobile noise suppression systems and evaluation of their effectiveness with regard to the changes of sources and reflection planes.

When the passive method of noise suppression is implemented in the industrial or other premises, the theoretical model will allow supplementing the structural model of the analyzed premise with the acoustic noise reduction equipment – noise suppression screens, selection of their geometrical parameters, arrangement in space and materials, in order to improve the maximal noise reduction and to predict the values of the acoustic field's parameters at the analyzed point of the real object.

References:

1. **Everstine G. C.** Finite element formulation of structural acoustics problems, Comp. & Structures. 1997. Vol.65. P.307-321.
2. **Ihlenburg F.** Finite element analysis of acoustic scattering. Springer, New York. 1998.
3. **Morand H. J.-P. and Ohayon R.** Fluid structure interaction: Applied numerical methods. Wiley, Chichester, UK. 1995.
4. **Tsingos N.and Gascue**l **J.-D.** 'Soundtracks for computer animation: Sound rendering in dynamic environments with occlusions. GraphicsInterface'97. May 1997. P. 9–16.

R. Mikalauskas, V. Volkovas

Akustinio lauko teorinio BEM modelio sukūrimas ir efektyvumo analizė

Reziumė

Šiame darbe akustinis laukas modeliuojamas BEM pagrindu, imituojant realias gamybinės patalpos eksploatavimo sąlygas, analizuojamos mobilios, triukšmą mažinančios sistemos efektyvumo ir taikymo galimybės, lyginami tiriamųjų bandymų rezultatai. Daroma išvada, kad modelis leidžia kurti mobilias garso slopinimo sistemas, įvertinti jų efektyvumą, taip pat atsižvelgiant į šaltinių ir atspindėjimo plokštumų pasikeitimus tiriamosios patalpos struktūrinį modelį papildyti akustinio triukšmo mažinimo priemonėmis – triukšmo slopinimo ekranais, parinkti jų geometrinius matmenis, išdėstymą erdvėje ir medžiagas, prognozuoti akustinio lauko parametrų vertes nagrinėjamame realaus objekto taške.

Pateikta spaudai 2007 12 12

DOI: 10.5755/j01.u.62.4.17041