NUMERICAL MODELING OF ACOUSTIC WAVE PROPAGATION IN UNLIMITED SPACE

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Abstract: *This article refers to the process of numerical modeling of acoustic wave propagation in unlimited space. Existing analytical methods are difficult to apply but present the main difficulty because they lead to some point values so that analysis of entire domain involves a large volume and is difficult to interpret. This paper put in evidence the influence of a sound absorption panel present. Numerical calculation provides post-processing of results a quick image of the distribution of acoustic pressure in the analyzed domain. The numerical analysis was carried out using Ansys program.*

Keywords: *acoustic pressure, acoustic elements, acoustic impedance, FEM.*

1. INTRODUCTION

Worldwide, the overall noise level is alarmingly high, we live in a noisy society mainly due to the technological environment in which we evolved.

The impact of noise on communities which are living in the proximity of airports represents a major importance for the aircraft manufacturers and airline operators for more than four decades. Knowing both aeronautics and acoustics is essential for a clear understanding of any aviation noise issue. Such an understanding is a necessary condition for controlling indoor and outdoor noise.

The main noise source of an aircraft is its propulsion system. Therefore, regarding the noise, airplanes can be classified in terms of their engine types.

Existing analytical methods are difficult to apply in this case and the main problem is that they lead to some point values so that analysis of entire domain involves a large volume and is difficult to interpret.

Numerical calculation provides post-processing results as a quick image of the distribution of acoustic pressure in analyzed domain. The finite element method is a numerical method that can be used for the accurate solution of complex engineering problems. Over the years, the finite element technique has been so well established, that today it is considered one of the best methods for solving a wide variety of practical problems efficiently.

2. MODELING WITH FINITE ACOUSTIC ELEMENTS

2.1. Fundamentals.

Finite Element Method is a general method of roughly solving differential equations with partial derivatives that describe or not a physical phenomenon.

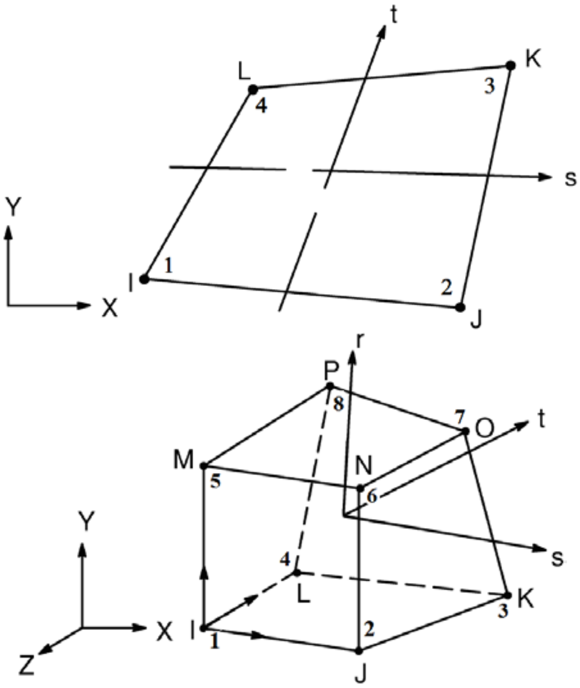
There are two kinds of models: 2D or plane models and 3D or spatial models. For each model type finite elements were created. As the Ansys finite element library is concerned 2D acoustic fluid and 3D acoustic fluid elements exist.

The most used acoustic finite elements are named FLUID29 for 2D modeling and respectively FLUID30 for 3D modeling, in the Ansys finite element library.

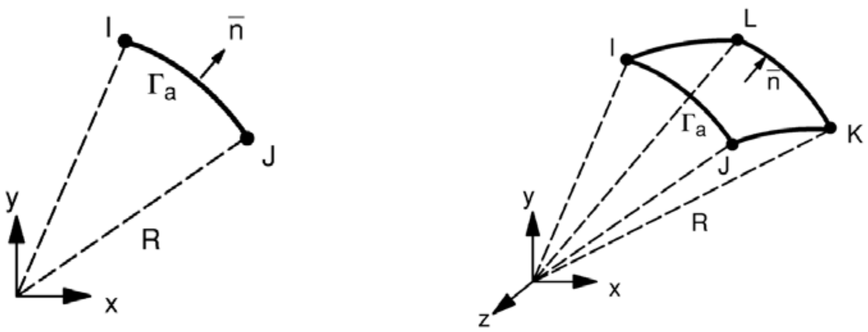
Next to these finite elements other kinds of finite elements were designed for infinitely modeling of the acoustic domain; practically, such finite elements are placed on the acoustic domain boundary and they work like a damping at the boundary by simulating the energy dissipation. So, no reflection wave occurs.

FLUID 129 and respectively FLUID130 finite elements are used for endless boundary modeling in the wave propagation direction.

In the case of the interaction between acoustic waves with a structure, like in the case of present of an acoustic absorbing panel, others proper structure finite elements can be used. A detailed description of all these finite elements can be found in Ansys theoretical manual and in main literature about finite element modeling [1], [6], [10] and [11]. Those two finite element types are presented in the Fig. 1.



a) FLUID29 b) FLUID30



c) FLUID129 d) FLUID130

FIG.1. Acoustic finite elements

The FLUID29 is a plane finite element, in the xOy plane, having two options: a planar or an axisymmetric finite element. All acoustic finite elements are isoparametric elements. Numerical analysis by FEM is made under following assumptions: the fluid is compressible (density changes due to pressure variations), the fluid is inviscid (no viscous dissipation exists), there is no mean flow of the fluid and the mean density and pressure are uniform throughout the fluid.

The fundamentals of acoustic finite element analysis start from the lossless acoustic wave equation [11],

$$\frac{1}{c^2}\frac{\partial^2 P}{\partial t^2}-\nabla^2 P=0 \tag{1}$$

where c is the speed of sound and P is the acoustic pressure.

The field parameters like acoustic pressure P and displacements u, inside the finite element can be calculated:

$$P=\{N\}^T\{P_e\} \tag{2}$$

$$u=\{N'\}^T\{u_e\} \tag{3}$$

where $\{N\}$ is the element shape function for pressure, $\{N'\}$ is element shape function for displacements, $\{P_e\}$ is the nodal pressure vector and $\{u_e\}=\{u_{ex}\},\{u_{ey}\},\{u_{ez}\}$ are the nodal displacement component vectors.

After some mathematical operations, from relation (1), the acoustic fluid matrices are obtained [11]:

$$\frac{1}{c^2}\int_V\{N\}\{N\}^T dV\{\ddot{P}_e\}+\int_V[B]^T[B]dV\{P_e\}+\rho_0\int_S\{N\}\{n\}^T\{N'\}^T dS\{\ddot{u}_e\}=\{0\} \tag{4}$$

or,

$$\left[M_e^P\right]\{\ddot{P}_e\}+\left[K_e^P\right]\{P_e\}+\rho_0[R_e]^T\{\ddot{u}_e\}=\{0\} \tag{5}$$

where the acoustic fluid matrices are:

$\left[M_e^P\right]=\frac{1}{c^2}\int_V\{N\}\{N\}^T dV$ is the fluid mass matrix; (6)

$\left[K_e^P\right]=\int_V[B]^T[B]dV$ is the fluid stiffness matrix; (7)

$\rho_0[R_e]=\rho_0\int_S\{N\}\{n\}^T\{N'\}^T dS$ is the coupling mass matrix (fluid-structure interface). (8)

$\{B\}=\{L\}\{N\}^T$ is a matrix resulting from applying of the matrix operator $\{L\}$ to element shape functions, and

$$\{L\}=\nabla(\ )=\begin{bmatrix}\frac{\partial}{\partial x}\\ \frac{\partial}{\partial y}\\ \frac{\partial}{\partial z}\end{bmatrix} \tag{9}$$

As the shape functions are concerned, these are those known from the general theory of finite element analysis, which can be expressed in local or general coordinates.

The modeling of the acoustic pressure field has some peculiarities dependind on acoustic domain which can be a closed one or an open one. In this case, a 2D model has to be an axisymmetric model; a 3D model can be used in any case, but it can be a complete one or a partial one, depending on the domain characteristics.

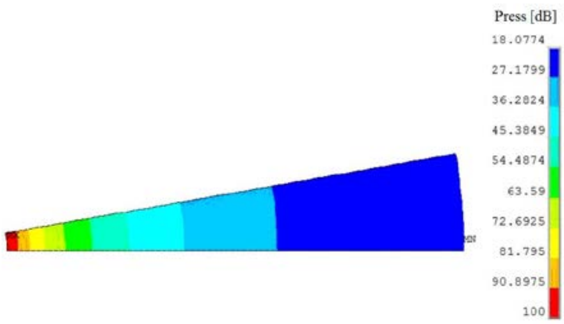


FIG. 2 Acoustic pressure field in 2D analysis

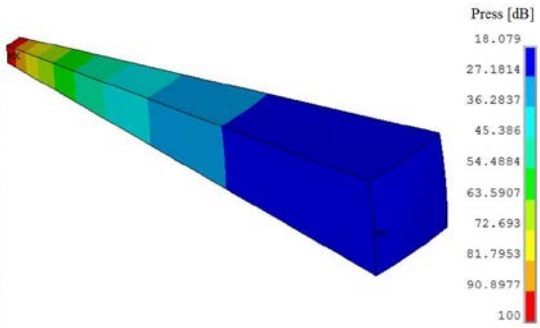


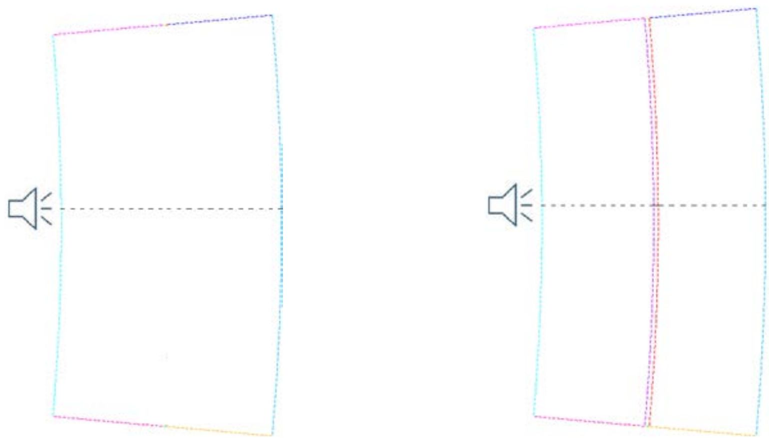
FIG. 3 Acoustic pressure field in 3D analysis

The Figures 2 and 3 present the calculus results for the same problem, but in two cases of modeling, 2D and respectively 3D.

Looking at the Fig. 2 and Fig. 3 we can see that the acoustic pressure values are practically the same no matter the model type 2D or 3D. Of course, a 2D model, when it is appropriate, is more efficient than a 3D model being easier to build and being solved in a shorter computer time.

2.2. Problem Formulation and Modeling.

Because we are interested not only in the acoustic field modeling, but in studding the influence of an acoustic absorbing panel, the problem domain is represented by a piece of acoustic field around the panel. So, the acoustic wave propagation range is defined in cylindrical coordinates with an angular symmetrical opening of 10 degrees and 1 m dimension on the propagation direction, as Fig. 4 shows.



a) Acoustic source and domain b) Geometrical model with absorbing panel

FIG. 4 Geometrical models

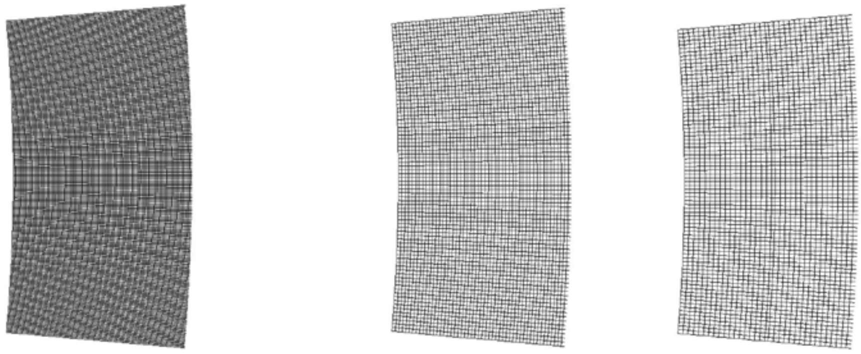
The authors have opted for a 2D axisymmetric model, this being most convenient (easy to build, shorter computer time).

Table 1. The properties of the materials

Details	*Density* (ρ)	*Sonic velocity* (c)	*Impedance* ($c \cdot \rho$)	Young's modulus (E)	Poisson ratio (υ)
	kg/m^3	m/s	$N \cdot s/m^3$	Pa	-
Air	1.225	340	416.50		
Aluminum	2700	5200	14040000	$0.7 \cdot 10^{11}$	0.33

The Fig. 4 presents those two cases taken into account: without absorbing panel (Fig. 4-a) and with the present of the absorbing panel (Fig.4-b). In the both cases a stationary nodal acoustic source with 100 dB was considered. The absorbing panel is made of aluminum having a thickness of 20 mm.

In the Table 1 the main properties of materials, used for solving the problem, are presented. The mesh used in developing of the acoustic numerical analysis was chosen after a short analysis of three versions (a, b and c) regarding the finite element size. Those three sizes of finite elements were 10 mm, 20 mm and 30 mm, for all those three element types: FLUID29, FLUID129 and PLANE42.

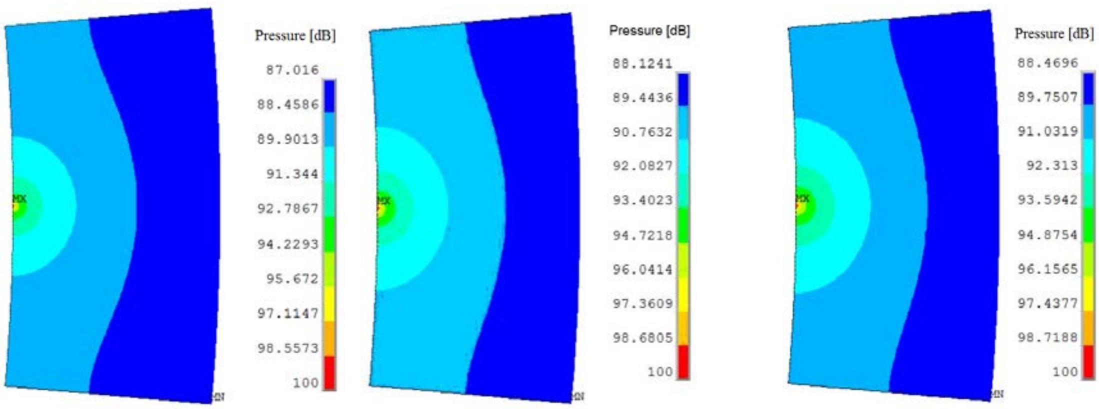


a) FE size = 10 mm b) FE size = 20 mm c) FE size = 30 mm

FIG. 5 Finite element models

The acoustic panel (screen) is placed in the middle of the field (Fig. 4-b), across the opening, each element having 4 nodes (the finite element PLANE42). The panel, as a mechanical structure, is clamped at the lower and upper parts. The right hand side of the domain boundary is also meshed with finite elements FLUID129.

For choosing the best mesh, those three mesh versions were used, running the program without any acoustic panel. Fig. 6 shows the acoustic pressure field for those three mesh versions considered.



a) *b)* *c)*

Fig. 6 The acoustic pressure field for three mesh versions

The Table 2 presents a synthetically analysis of the results obtained those three mesh version.

Table 2. Comparative analysis

F.E. size	Nodes	Elements	Max. value	Min. value	Differences
1 cm	16261	16160	88.4586	87.0160	1.4426
2 cm	4131	4080	89.4436	88.1241	1.3195
3 cm	2835	2800	89.7507	88.4696	1.2811
		Average values:	***89.2176***	***87.8699***	***1.3477***
Errors (%) towards average value:					
1 cm			***-0.85***	***-0.97***	***7.04***
2 cm			***0.25***	***0.29***	***-2.09***
3 cm			***0.60***	***0.68***	***-4.94***

As it results from the analysis of the values presented in the Table 2, the best mesh version is that having the finite element size of 20 mm.

3. RESULTS AND DISCUSSIONS

The results of numerical analysis with finite elements are presented synthetically and suggestively in the figures below, where these were obtained by graphically post-processing.

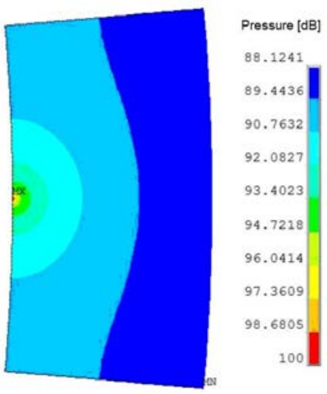


FIG. 7 Acoustic pressure field without panel

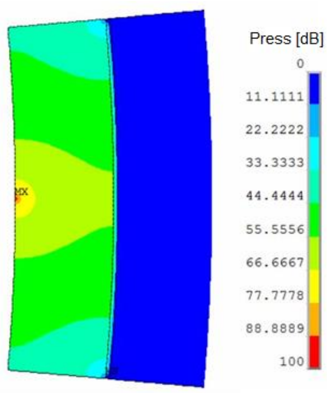


FIG. 8 Acoustic pressure field with panel

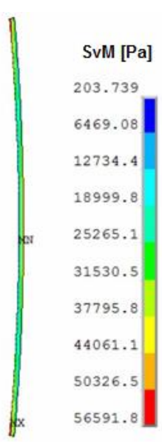


FIG. 9 Equivalent von Mises stress field of the panel

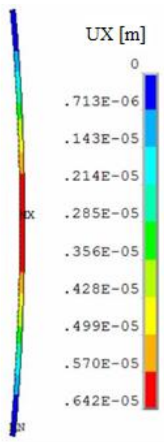


FIG. 10 UX-displacement field of the panel

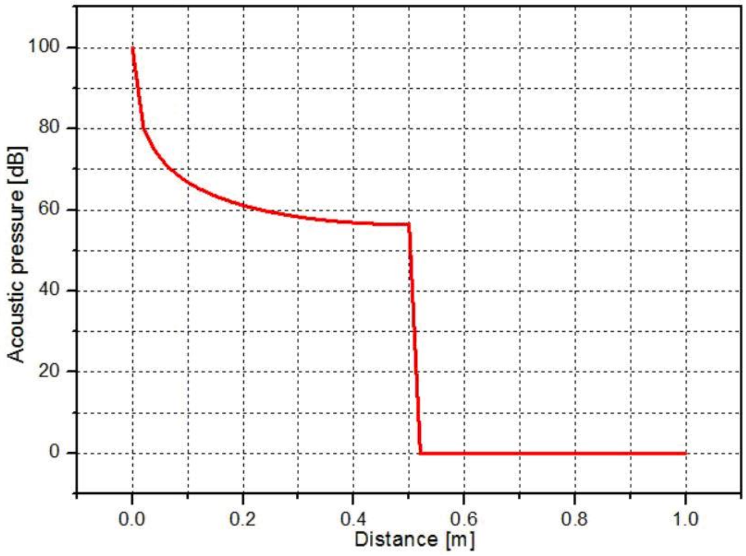


FIG. 11 Pressure-distance curve on median propagation direction

Fig. 7 and Fig. 8 demonstrate the significant effect of the acoustic absorbing panel considered, which causes the sound pressure level behind not to exceed 11.111 dB; the efficiency of the screen is proven by the comparative analysis of the Figures 7 and 8.

It is found that in the absence of the screen, the sound pressure level in the field does not fall below 88.1241 dB, thus remaining very high (the decrease is only 11.876% from the source level).

The acoustic pressure has the effect of an elastic stress loading on the screen, as shown in Figure 9 which shows the von Mises equivalent stress field in the screen, where the maximum value is 0.056592 MPa (low value, which does not raise problems from point in terms of mechanical strength).

Figure 10 shows the field of the nodal displacements of the acoustic absorbing panel in the direction of the acoustic wave propagation. As expected, the maximum displacement occurs at the middle of the screen, reaching the value of 0.00642 mm (this value is a small one, meaning the reduced mechanical load due to the acoustic pressure action).

Fig. 11 contains the variation curve with the distance from the source of the acoustic pressure in the direction of the acoustic wave propagation at the source level, in the conditions of acoustic panel existing.

As it seen in the Figure 11, a significant drop in the sound pressure level is produced by the panel presence. The considered acoustic absorbing panel produces a consistent variation, from 56.4 dB to zero dB.

The acoustic absorbing panel, under the given conditions, without significant mechanical stress, produces significant sound insulation.

A wider and more useful discussion can be done through similar simulations of several types of screens, different in size, material and shape.

4. CONCLUSIONS

Numerical analysis by finite element method allows a quantitative and qualitative evaluation of the effects of soundproof panels (screens). In this paper the methodology is presented first, for a given case, so that the conclusions and the observations are based on quantitative determinations.

The adopted 2D model allows fast calculation, which favors the study of a significant number of variants of the screen within a reasonable time.

The effect of soundproof screens can be studied by numerical simulation, both in the case of acoustic waves in transient mode and taking into account the influence of the frequency of sound waves. These issues are the subject of further studies, including a doctoral thesis.

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