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SOUND AND THERMAL INSULATING PROPERTIES OF MATERIALS ON MINERAL BASES

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Abstract

Results of the thermal and acoustic testing of selected mineral based materials used in building construction are presented. The sound absorption and reflection coefficient of the normal incidence sound wave on tested materials in the frequency range of 50 – 6500 Hz and thermal conductivity coefficient were determined. Obtained results were confronted with the theoretical numerical prediction models based on FEM BEM methods as obtained from the SAMCEF software. Obtained results allow optimization of the wall structure construction for household and building industry applications.

INTRODUCTION

The aim of this research was preparation and creation of the software module for simulation of material behaviour with perpendicular incidence of acoustic wave by means of numerical method – BEM/FEM.

1 PRACTICAL APPLICATION

In the practical part there were discussed possibilities of selected plate constructions for sound absorbing mats. In first part we solved the problem of vibrating plates and level of transmission loss with application on particular geometrical shape with engaged boundary conditions. Single method, which is fast and complex for effective using in solution of whole spectrum of problems with vibrating plates is the finite element method (FEM), eventually boundary element method (BEM).

1.1 Acoustic system – plate construction

Plate construction represents absorptive acoustic system with explicit frequency of resonance. The system is initialized to forceable oscillating with incidence of acoustic wave. Amplitude of oscillations extends of maximum for resonance frequency. In case, that acoustic wave will finish incidence on acoustic system, that way acoustic system will oscillate on natural frequency for some time. Period, after which resonance frequency fades, depends on internal damping of plate and called period of fading. In regard of total energy quantum, attends with incidence of resonator to inhibition every time. Because part of sound energy is changed into the other energy, generally to thermal energy.
On this accont the resonance system is effective only in case, that the system is damped down enough.
Absorbing of system is characterized with the coeficient of absorptivity. Dependence of coeficient of absorptivity on frequency shows with every resonance system maximum on natural frequency of system. Quantum of sound energy, changed into other energy, grows with size of amplitude of acoustic magnitude (generally acoustic speed) and the acoustic magnitude amounts maximum on resonance frequency. Critical frequency f_K is called coincident frequency in case, that natural frequency f of oscillation of flexion of plate and the frequency of acoustical wave are same.

The level of transmission loss of plate represents next equation

$$R(\Phi) = 10 . \log \left\{ 1 + \left[\frac{r_M \; . \; h_w \; . \cos \Phi}{2 \; . \; r_0 \; . \; c_0} \; . \left[1 - \left(\frac{f}{f_K} \right)^2 . \sin^4 \Phi \right] \right]^2 \right\} \tag{1}$$

Equation is true of homogenous isotropic plate.

1.2 Theoretical introduction to structural dynamics – basic equations of finite element method

When the elastic structure is subjected to the dynamic load, the displacement field within the structure varies with time and two types of distributed body forces must be taken into account. It his case the elasticity tensor is time dependent, because the loading of the structure may be a function of time.
Within a typical three-dimensional element we assume, that the displacement field is expressed as

$$\left\{\tilde{d}\right\}^e = \begin{Bmatrix} u(x,y,z,\,t) \\ v(x,y,z,\,t) \\ w(x,y,z,\,t) \end{Bmatrix}^e = \begin{Bmatrix} \sum_{i=1}^{r} N_i(x,y,z)u_i(t) \\ \sum_{i=1}^{r} N_i(x,y,z)v_i(t) \\ \sum_{i=1}^{r} N_i(x,y,z)w_i(t) \end{Bmatrix}^e \tag{2}$$

Minimization of the potential energy with respect to the nodal values of displacements leads to force-displacement relations for a typical node, except that all the parameters may be time dependent.
Hence at node q we have for the body force term the usual term plus two extra terms :

$$\left[F_B\right]^q = \int_{\Omega} \left[N_q \left\{F^*\right\}_q - c\,N_q \left\{\dot{d}\right\}^q - r\;\; N_q \left\{\ddot{d}\right\}^q \right] d\Omega^e \tag{3}$$

Assembly of the nodal equations then leads to element equations of the form

$$\left[M\right]^e \left\{\ddot{d}\right\}^e + \left[C\right]^e \left\{\dot{d}\right\}^e + \left[K\right]^e \left\{d\right\}^e = \left\{F(t)\right\}^e \tag{4}$$

where for an element with r nodes we have

$$\left[M\right]^e = \int_{\Omega^e} r\,\left[N\right]^T\;\left[N\right]\,d\Omega^e \tag{5}$$

and

$$\left[C\right]^e = \int_{\Omega^e} c\left[N\right]^T\;\left[N\right]\,d\Omega^e \tag{6}$$

in which we define the interpolation function matrix as

$$\left[N\right] = \left[\left[I\right]N_1, \left[I\right]N_2, \ldots\ldots\ldots\ldots\ldots\ldots\left[I\right]N_r\right] \tag{7}$$

with

$$\left[I\right] = \begin{bmatrix} 1 & 0 & 0 \\ 0 & 1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \tag{8}$$

Futher assembly gives the system equation of motion

$$[M]\{\ddot{d}\} + [C]\{\dot{d}\} + [K]\{d\} = \{F(t)\} \tag{9}$$

If we imagine the ideal case in which the system has no damping and no external forcing functions, the equations of motion reduce to

$$[M]\{\ddot{d}\} + [K]\{d\} = \{0\} \tag{10}$$

To find this natural modes and frequencies we assume that the diplacement vector may be expressed as

$$\{d\} = [\Phi]\, e^{iwt} = \{\Phi\}\ \{\cos w\,t + i \sin w\,t\} \tag{11}$$

where $\{\Phi\}$ is the vector of unknown amplitude at the nodes (modal vector) and w is one natural frequency. Noting that

$$\{\ddot{d}\} = w^2\ [\Phi]\, e^{iwt} \tag{12}$$

we find upon substitution that the equation (10) reduces to

$$\left[[K] - w^2 [M]\right]\{\Phi\} = \{0\} \tag{13}$$

This equation is recognized as an eingenvalue problem. The equation has a nontrivial solution only when the determinant $\left|[K] - w^2 [M]\right| = \{0\}$. This is equivalent to the polynomial

$$\left(w^2\right)^n + (\quad)\left(w^2\right)^{n-1} + ... + (\quad)\left(w^2\right) + (\quad) = \{0\} \tag{14}$$

For matrixes of dimensionen $n\ x\ n$ there will be n values of w_i^2 satisfying eguation (14) and hence n vectors $\{\Phi\}$ that satisfy equation (13)

1.3 Modal analysis of plate construction

1.3.1 Thin plate – shell element

The thin plate for calculation of natural frequencies and shapes of oscillations was studied. Material of plate was steel, plaster board and polystyrene. Size of plate was $a \times b$ = 500*mm x* 300*mm*. Solution of natural frequencies and natural oscillation was implemented with thin-walled plate (shell) elements. On figure no. 1 is finite element net and boundary conditions of plate mounting. Figure no. 2 represent first four natural shapes of oscillations and natural frequencies of plates maded from different materials. Calculations were implemented in FEM software “SAMCEFV10.1-03.“

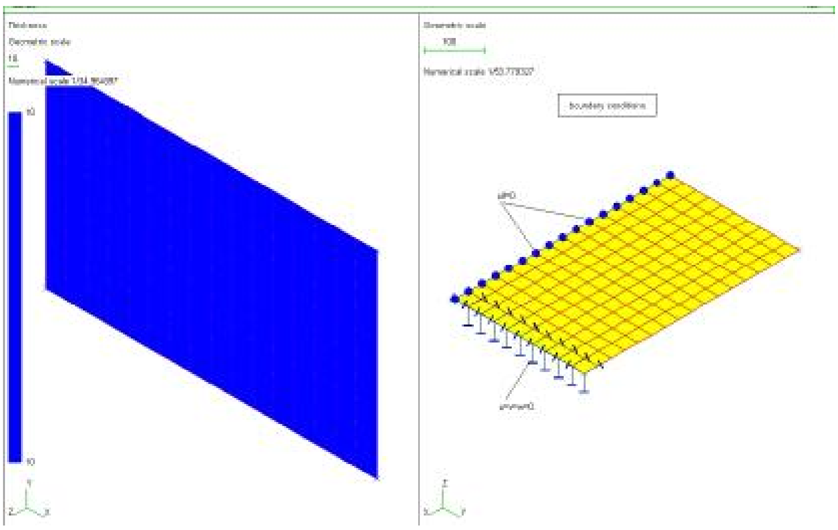

Figure no. 1 – Thickness of plate and boundary conditions

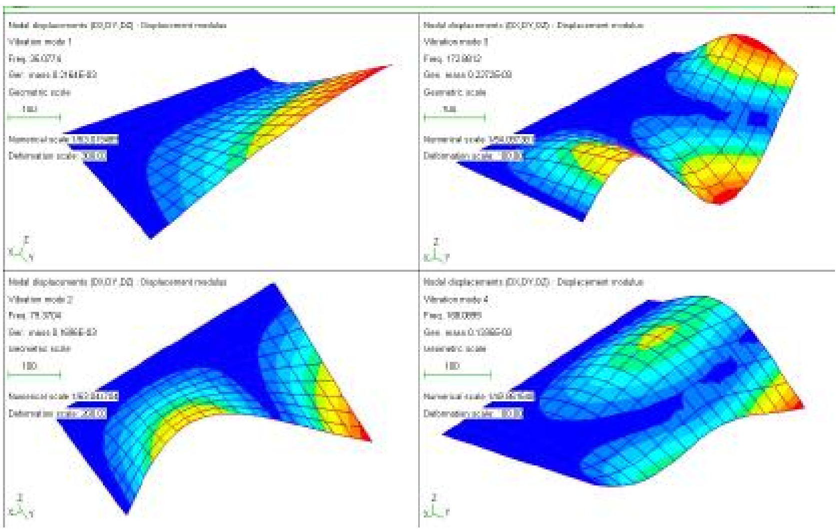

Figure no. 2 - Material - plasterboard

1.3.2 Plates with middle thickness – volume element

The plate for calculation of natural frequencies and shapes of oscillations was studied. Material of plate was steel, plaster board and polystyrene. Size of plate was $a \times b$ = 500*mm* x 300*mm*. Thickness of plate was t = 60*mm*. Solution of natural frequencies and natural oscillations was implemented with cubical (volume) elements. On figure no. 3 is finite element net and boundary conditions of plate mounting. Natural frequencies and natural shapes of oscillations for sandwich plate were computed. Sandwich plate was composed with three layers – figure no. 4. Outer layers were composed by plasterboard with thickness of t = 10*mm* and middle layer was composed by polystyrene with thickness of t = 40*mm*.

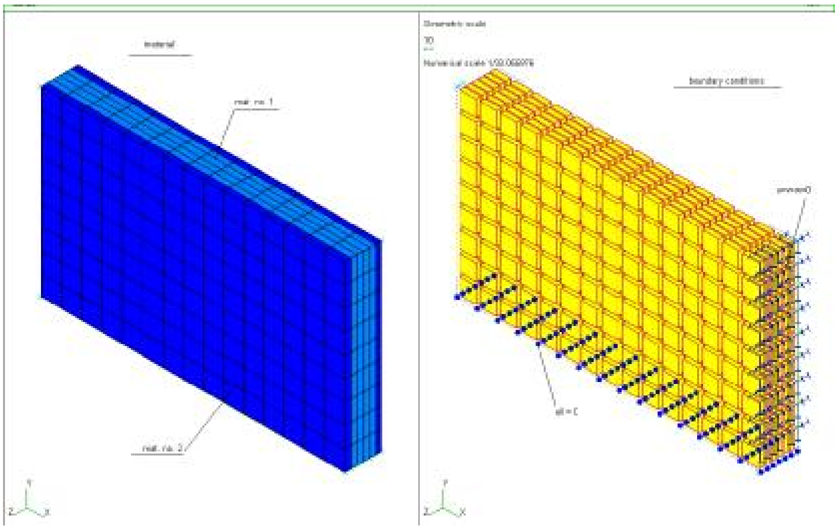

Figure no. 3 - Thickness of plates and boundary conditions

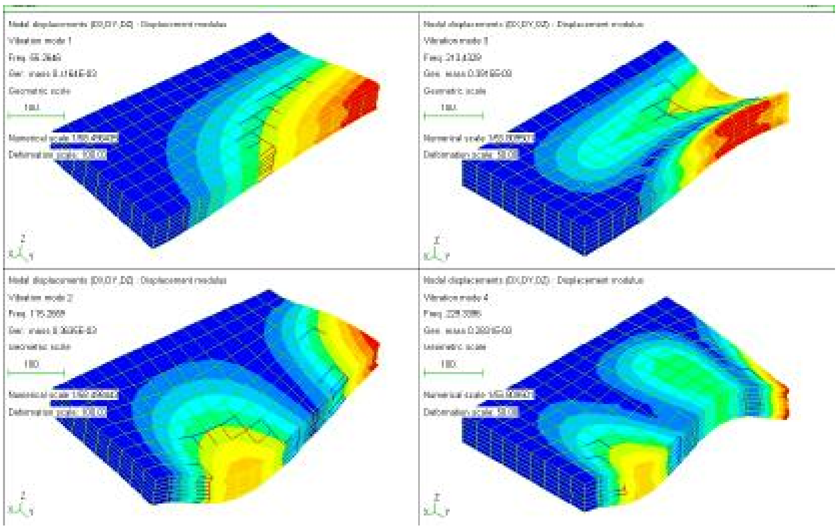

Figure no. 4 – Sandwich-plate plasterboard-polystyrene-plasterboard

On figures no. 5-6 is presented example of the solution of unstationary field of temperature in plate. The Plate is heated on the left side with the constant convection. The solution was created in FEM software “SAMCEFV10.1-03.“

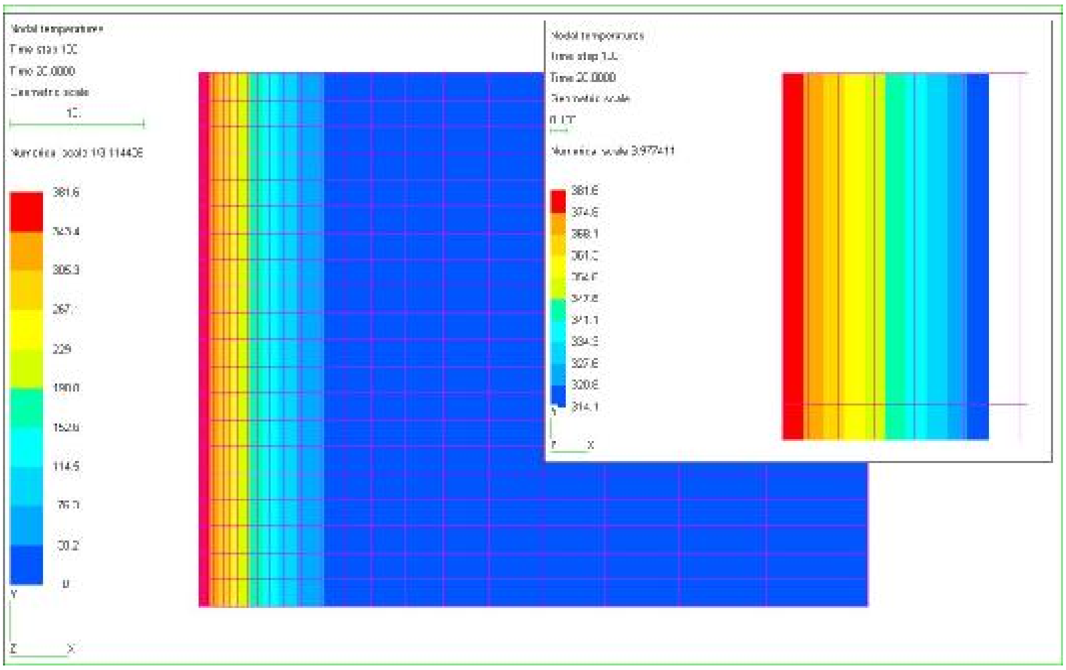

Figure no. 5 – Distribution of the field of temperature in plate.

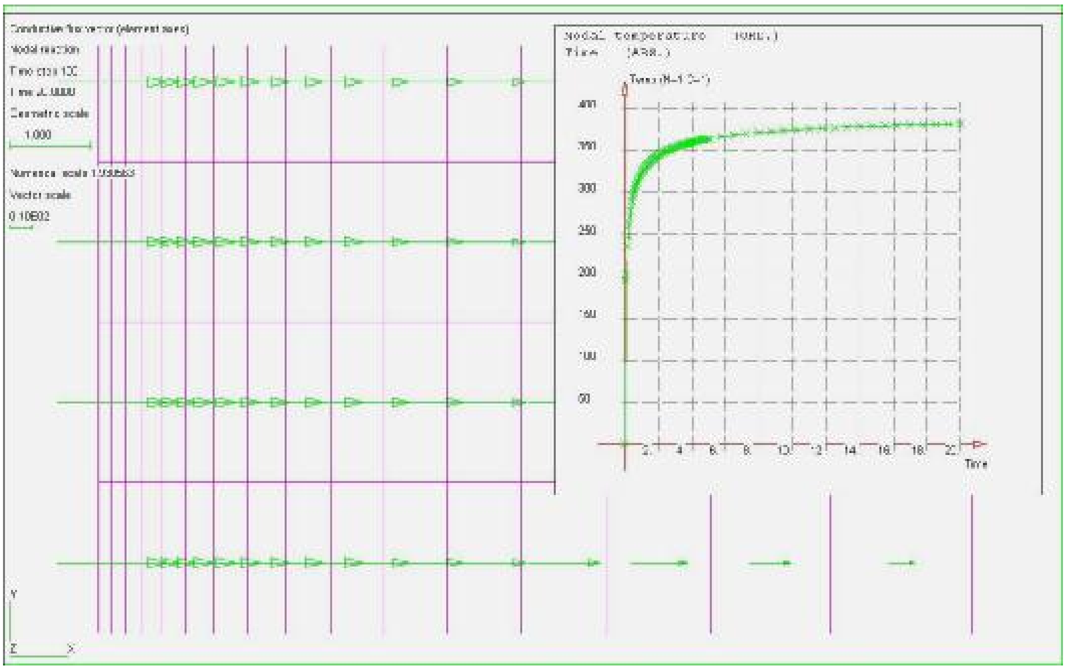

Figure no. 6 – Time computation of temperature on the left side of plate.

CONCLUSIONS

Study of acoustic properties of building materials was focused on simulation of chosen materials behaviour with incidence of acoustic wave with using of finite element method, namely BEM.
Until now large amount of the different material systems based on polymer blends with binders were studied in detail. Only limited number of experiments of materials on mineral base were published. Integrated technique for evaluation of building material behavior was offered in this paper. Technique is based in meassuring basic material properties (mechanical and acoustic properties), following by simulation of the materials behavior with incidence of acoustic wave via FEM Software Samcef or simulation of combination of more building materials.

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