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ACOUSTIC FIELD IN THE MECHANICAL WORKSHOP

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Either wave methods or geometric ones are used to describe an acoustic field in industrial halls. The acoustic field distribution in such halls is not uniform, thus the assumption of field diffusivity applied in an acoustics of interiors is often not fulfilled. The room forms amplitude and phase characteristics of the signal, introducing additionally effects related to space development, which are decisive for the acoustic pattern in the observation point. The paper presents an analysis of an acoustic field inside an industrial room, when one machine of not large dimensions is operating. The geometric model of the workshop was implemented and Raynoise software was used to determine the distribution of the sound level for selected frequencies. The measurement results and computer simulations related to the acoustic field distribution are given.

Keywords: industrial noise, acoustic field.

1. Introduction

Two factors decide on the planning of an interior of an industrial hall: the way of machining and the kind of transport. The size and shape of a hall is determined by what the hall is to be used for. Sound propagation inside a hall is a physical process that mainly depends on the shape and dimensions of the enclosed space, the interior as well as characteristics of the sound absorptive materials the surrounding surfaces are made of. The hall creates the amplitude and phase characteristics of the signal, additionally introducing effects related to its space that influence the sound picture reception by a person or by a microphone.

If we are dealing with halls many times larger than the wave-length radiated by a given source, no resonance vibrations are formed and the distribution of reflections can be analysed with geometric methods. However, those methods do not take into account either real deflections of sound waves or dissipation at reflections from surfaces which are not planes. Thus, they are not accurate especially when low frequencies constitute most of the acoustic characteristics of the source. Another limitation of geometric methods is that they do not take into account any wave attenuations caused by the material of the surrounding surfaces or attenuation caused by air. However, due to the often low value of the average absorption coefficient in the majority of industrial halls, those methods can be used in their acoustic analysis. The experimental part of this work was performed in a mechanical workshop, a typical industrial room.

2. The acoustic field in an industrial room

According to the Helmholtz equation, in a model of the acoustic field radiated by a monopole sound source, we can write:

$$\nabla^2 p(\mathbf{x}) + k^2 p(\mathbf{x}) = 0 \tag{1}$$

for boundary conditions determined for a locally influencing surface:

$$\nabla p(\mathbf{y})\mathbf{n} = jk\beta(\mathbf{y})p(\mathbf{y}), \tag{2}$$

where $p(\mathbf{y})$ – complex value of acoustic pressure [Pa], k – wave number, [rad/m], $\mathbf{x}$ – position of the receiving point inside the room [m], $\mathbf{y}$ – position on the wall in the room [m], β – dimensionless coefficient of the acoustic admittance of the wall, $\mathbf{n}$ – unit vector normal to the surface of the wall.

The acoustic field, both for a free field and inside a room, is the sum of acoustic pressures of L single sound sources [3].

$$p(\mathbf{x}) = \sum_{i=1}^{L} G(\mathbf{x} \,|\, \mathbf{x}_i)(-j\omega\rho q_i), \tag{3}$$

where q_i – complex value of a strength of the i-th source localized in point x_i, ρ – density of the surrounding medium, [kg/m^3], $G(\mathbf{x} \,|\, \mathbf{x}_i)$ – Green's function for the free field:

$$G(\mathbf{x} \,|\, \mathbf{x}_i) = \frac{e^{jk|\mathbf{x}-\mathbf{x}_i|}}{4\pi|\mathbf{x}-\mathbf{x}_i|} \,. \tag{4}$$

However, when the source is inside the room, Green's function, taking into account the eigenfunctions of room Ψ_n, can be written as:

$$G(\mathbf{x} \,|\, \mathbf{x}_i) = \sum_{n=0}^{N} \frac{\Psi_n(\mathbf{x})\,\Psi_n(\mathbf{x}_i)}{V|k_n^2 - k^2|} \,, \tag{5}$$

where Ψ_n – n-th eigenfunction of the room for the boundary conditions determined with Eq. (2), k_n – wave number related to the n-th eigenvalue, V – volume of the room, m^3.

The eigenvalues of the room are normalized [4], i.e.:

$$\int\limits_V \Psi_n\,\Psi_m\,\mathrm{d}V = \delta_{nm} \,,$$

where $\delta_{mn} = 1$ for $m = n$ and $\delta_{mn} = 0$ for $m \neq n$.

In the case of several observation points M we can write Eq. (3) with the application of a matrix notation:

$$\mathbf{p} = [p(\mathbf{x}_1) \ldots p(\mathbf{x}_m) \ldots p(\mathbf{x}_M)]^{\mathrm{T}} \tag{6}$$

for a finite number of observation points M.

When considering the acoustic field distribution on a certain plane above the floor, we can determine it as originating from a plane wave of the amplitude A dissipating in that plane:

$$p(\mathbf{x}) = Ae^{-j\mathbf{kx}}, \tag{7}$$

k – wave number vector.

However, when determining the wave in the room as a spherical wave we have to apply the following formula:

$$p(\mathbf{x}) = A\frac{e^{-jk|\mathbf{x}-\mathbf{x}_0|}}{|\mathbf{x}-\mathbf{x}_0|}\,. \tag{8}$$

The position of the source in the acoustic field inside the room is defined by $\mathbf{x}_0$. The discrepancy between the calculated and measured values can be defined with the error function:

$$e(\mathbf{x}) = p^{\mathrm{cal}}(\mathbf{x}) - p^{m}(\mathbf{x}). \tag{9}$$

If the acoustic parameters of partial sources of sound **q** are determined correctly, the error function value $e(\mathbf{x})$ approaches 0. For M observation points an error vector **e** can be introduced:

$$\mathbf{e} = [e_1,\, \ldots,\, e_m,\, \ldots,\, e_M]^{\mathrm{T}}.$$

Thus, the error vector can be minimised by controlling the source strength values **q**. The error criterion J between the expected and the estimated solutions can be expressed with the least squares' method [1–3, 6]:

$$J = \mathbf{e}^H\,\mathbf{e}. \tag{10}$$

This problem can be also solved with Tikchonov's regularization method:

$$J_M = \mathbf{e}^H\,\mathbf{e} + \gamma\mathbf{q}^H\mathbf{q}, \tag{11}$$

where γ is the regularisation parameter. The application of Tikchonov's method is given in [3].

3. Experimental tests inside the workshop

When a sound source inside a room radiates constant-intensity sound energy, energy losses are covered by the source and after some time there is equilibrium: the energy absorbed by the walls equals the energy supplied by the source.

The acoustic field inside industrial rooms can be described with a statistic theory for two different cases: an acoustic field in an empty room and an acoustic field in a room with a finite density of dissipating surfaces.

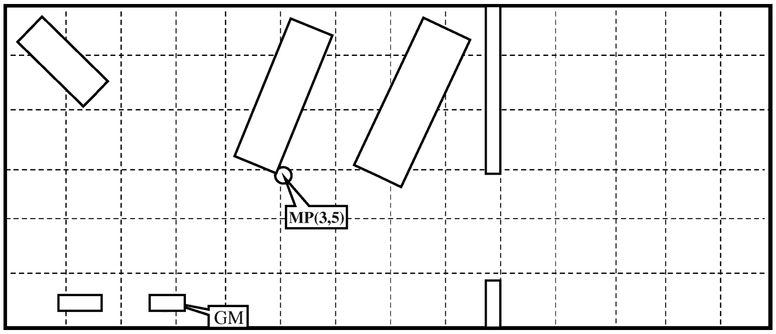

Fig. 1. The mechanical workshop schematic plan.

In experimental tests a 12-microphone system for multi-channel data collection was used. Reference microphones were placed directly at the machine, and the measuring microphone array was shifted along the 1 m by 1 m measuring grid nodes in a mechanical workshop (Fig. 1). The dimensions of the workshop were as follows: length – 14.2 m; width – 6.1 m; height – 6.6 m, volume $V = 581$ m^3. The effective values of acoustic pressure were recorded in 10-Hz bands in the 10 Hz to 12 kHz range. At the same time the phase shifting of the acoustic signal between the reference microphone and the remaining microphones was measured. With the measured values it was possible to plot the distribution of acoustic pressure level in the selected frequency bands in the steady state (Fig. 2). Then the geometric model of the workshop was implemented and Raynoise software was used to determine the distribution of the sound level for selected frequencies (Fig. 3). The next stage consisted in an experimental determination of the influence function for the investigated room for the point source located at the machine under testing (a grinding machine) [5, 7]. An influence function quantifies the dependence of an estimated parameter of the model as a function of the data. In this case the influence function quantifies the dependence of an estimated of the sound pressure level inside the work shop as a properties of the geometrical model. The actual position of the sound source (the grinding machine) and the geometrical model used are assumed. The values of the influence function for all octave band frequencies are obtained through calculation. A distribution maps of this function for selected frequencies is presented in Fig. 4.

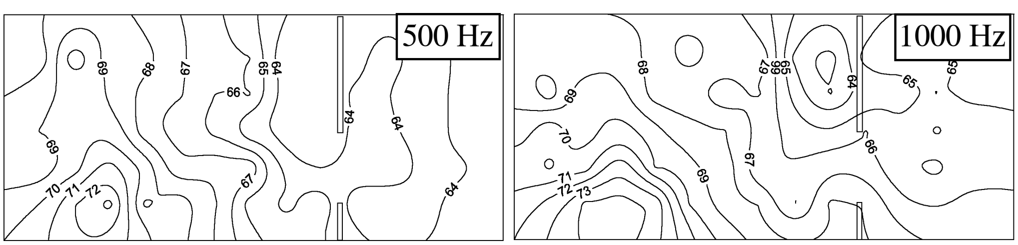

Fig. 2. The distribution of sound pressure levels, determined experimentally, for 500 Hz and 1000 Hz.

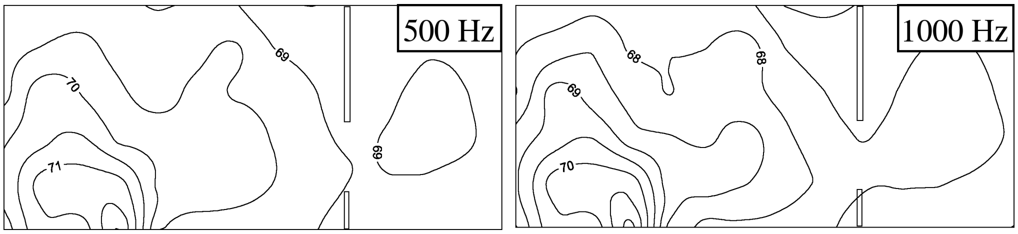

Fig. 3. The distribution of sound pressure levels, estimated in a computer simulation, for 500 Hz and 1000 Hz.

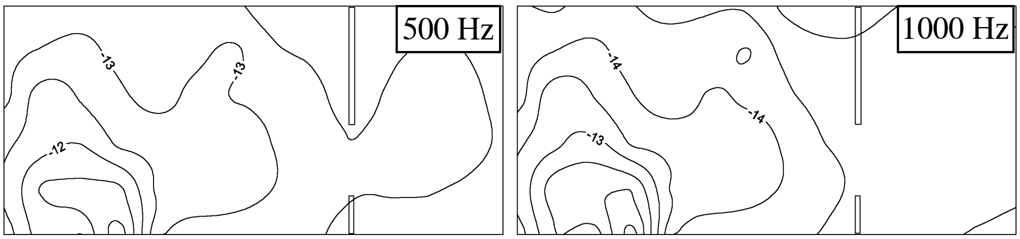

Fig. 4. A distribution map of the influence function for 500 Hz and 1000 Hz for a mechanical workshop.

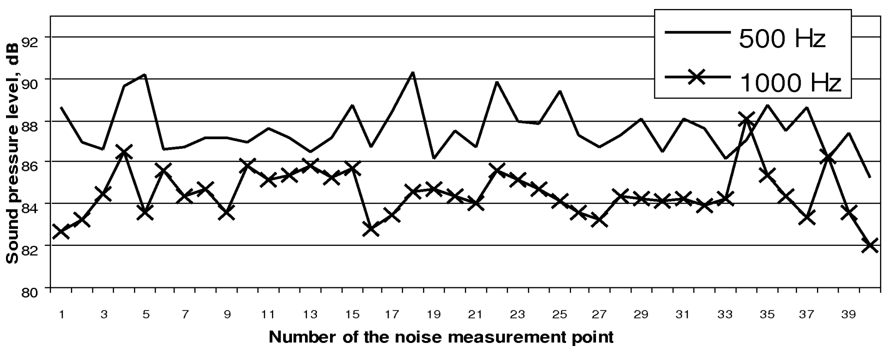

Fig. 5. Fluctuation of the sound pressure level at the source (the grinding machine).

There are some differences between the experimental and computed results. The computed values are 2–3 dB higher than the experimental data. One of the reasons for this is most likely the insufficient accuracy of the geometric model of the workshop. Another reason is the irregular sound level of the machine being tested. Those fluctuations of the sound pressure level for the frequencies 500 Hz and 1000 Hz are shown in Fig. 5.

4. Conclusion

The accurate estimation of the distribution sound pressure level is the basis for the determination of the sound power of a source in industrial room. Using modelling to investigate acoustic fields is especially applicable in correcting the acoustic climate of rooms. The geometric modelling of rooms, sound sources, selection of appropriate building and finishing materials, an analysis of the way a room has been planned and filled make it possible to modify already existing rooms or ones that are being designed. This paper discusses the experimental stage of research leading to an analysis of the acoustic phenomena in a mechanical workshop with one small machine. The results make it possible to extend investigations to larger machines, whose modelling will require substituting the machine with a larger number of partial sources of the sound. Further investigations intend to use a modernized measurement system for multi-channel data acquisition. This system uses a combination of PC-based measurement hardware and software to provide a flexible, user-defined measurement system. It allows for simultaneous multipoint measurements of the sound pressure level and phase shift angle inside the industrial testing hall. This makes it possible to minimize some kinds of errors. The next step will consist of determining localisation and acoustic parameters of machines on the basis of the acoustic field distribution in a room.

Acknowledgments

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