XX Polish Fluid Mechanics Conference,
Gliwice, 17-20 September 2012

NUMERICAL AND EXPERIMENTAL VISUALIZATION OF ACOUSTIC FLOW OVER FLAT OBSTACLES

Michał PYŁA, Stefan WEYNA, Witold MICKIEWICZ, Michał JABŁOŃSKI
West Pomeranian University of Technology, Szczecin Poland
E-mail: michal.pyla@zut.edu.pl

Abstract

In the paper authors have described the visualization methods in acoustic flow fields around and between a flat barriers and show how these methods may assist scientists to gain understanding of complex acoustic energy flow (vortisity and turbulent effects) in real-life acoustic field. Own proposals of the graphical form will be presented to determine the real acoustic wave distribution in 2D and 3D flow field. Visualization of research results are shown in the form of a *intensity streamlines* and as a *shape of floating acoustic wave* or *intensity isosurface*, which is unavailable by conventional acoustics metrology. Analysis of the results makes possible to obtain much new information about energetic and geometric distributions of the acoustic fields. The measurement technique described, as well as the method of graphical presentation of results, can enrich the knowledge of the mechanism of acoustic energy flux through the real partitions.

Key words: flow acoustic, sound intensity, numerical modeling

INTRODUCTION

Energy distribution images in acoustic fields, connected with the graphical presentation of the flow waves are a new element in acoustic metrology. Introduction of these possibilities has greatly changed the approach to examining many acoustic phenomena.

The new insight into the nature of acoustic field formation in real conditions of working sources may bring application of the sound intensity method in conjunction with the graphical presentation of space vector distribution of acoustic power. Acoustic conditions in these areas are much different from the theoretical assumptions ascribed to free or diffuse field. It is a frequent occurrence that the sound intensity measurements in real conditions may show great disparity between the theoretical assumptions of the acoustic fields distribution and the actual measurements. The disparity results mainly from the simplifications accompanying the analytical and numerical methods due to the lack of complete data concerning physical properties of an investigated object (de Rock at al. 2004).

In traditional acoustic metrology, the analysis of acoustic fields concerns only the distribution of pressure levels (scalar variable), however in real acoustic field both scalar and vector (the acoustic particle velocity) effects are closely related. Only when the acoustic field is described by both potential and kinetic energies may we understand the mechanisms of propagation, diffraction and scattering of acoustic waves on obstacles, as a form of energy. This attribute of intensity method is very important in any industrial acoustic investigations. Based on the research with intensity technique and using selected visualizations methods (Pyła, Weyna 2010), examples of vector space distribution of the real acoustic field are demonstrated in the publication.

SOUND INTENSITY MEASUREMENT

Sound intensity is the average rate at which sound energy is transmitted through a unit area perpendicular to the specified direction at the point considered (1). It is a vector quantity defined as the time averaged product of the sound pressure *p* (scalar) and the corresponding particle velocity *u* (vector) in the same position (Fahy 1989).

$$\vec{I} = \lim_{T\to\infty} \int_0^T \vec{u}(t) \cdot p(t) dt [W/m^2] \quad 1)$$

The frequency distribution of the mean intensity is

$$I(\omega) = Re\{G_{pu}(\omega)\} \quad 2)$$

Sound intensity amplitude may be determined by: a two-microphone method, cross correlation transform between pressures from two microphones and as a direct measurement of pressure and acoustic particle velocity (2). The aforementioned advantages of the SI technique may be used in acoustic metrology much more effectively if a new 3D-USP miniature intensity probe is applied (Fig. 1). The *Microflown Ultimate Sound Probe* - USP (made by Microflown Technologies B.V.) is a new type of sensor, as a practical SI transducer. It is a very compact and integrated sound probe that combines three orthogonally positioned particle velocity sensors and a miniature pressure microphone. The actual sensor configuration without its cap is less than 5 mm x 5 mm x 5 mm.

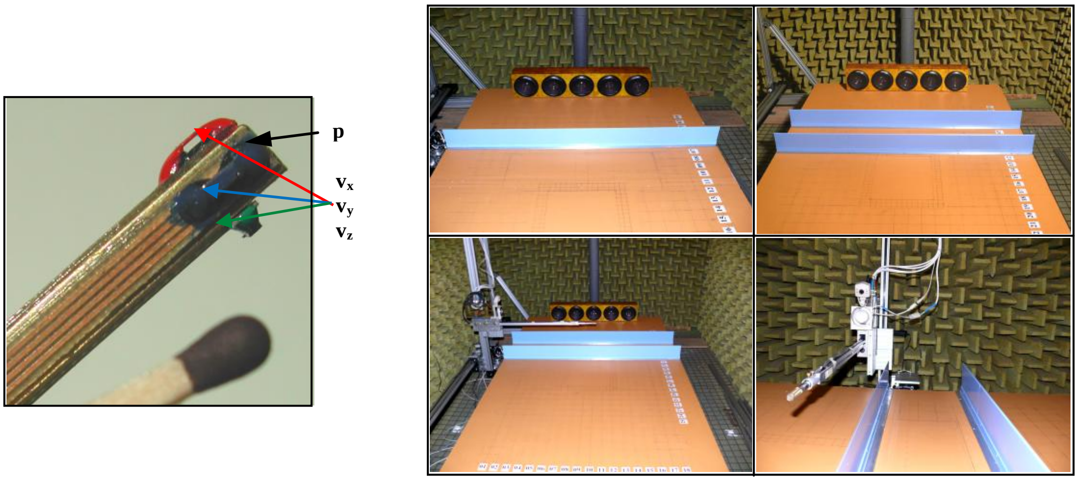

Figure 1. *The Microflown USP 3D sound intensity probe and the experimental set-up*

The Microflown 3D-USP, used as a scanning probe, was especially developed for measurements carried out very close to vibrating objects - the source of acoustic power. The USP effectively extends the traditional possibilities for complete sound intensity depictions of 3D energetic fields, by measuring three particle velocity vector components and the acoustic pressure (a scalar value). By minimizing the array distance to the sound source, we may investigate particle velocity levels in acoustic near field conditions, and the power acoustic flow may now be fully described in real-life experimental conditions.

ACOUSTIC WAVE STRIKING THE SINGLE AND DOUBLE THIN PLATE

Investigated models with palisade barriers (number of flat acoustic scatterers) should imagine noise-generating rotating machinery equipped with a propeller blade. The reduction of noise by active flow control is a method sought out for practical noise abatement in many industrial fields. For simplicity, in our model we assume that the blades are distributed linearly. In our research we are interested in the energy distribution of the acoustic field around obstacles. This is a comparative test, what sort of reaction can be observed while a running acoustic wave hits a single or two obstacles during the same excitation signal. Direct measurement of the acoustic power flow around plate explain diffraction and scattering phenomena occur in this region.

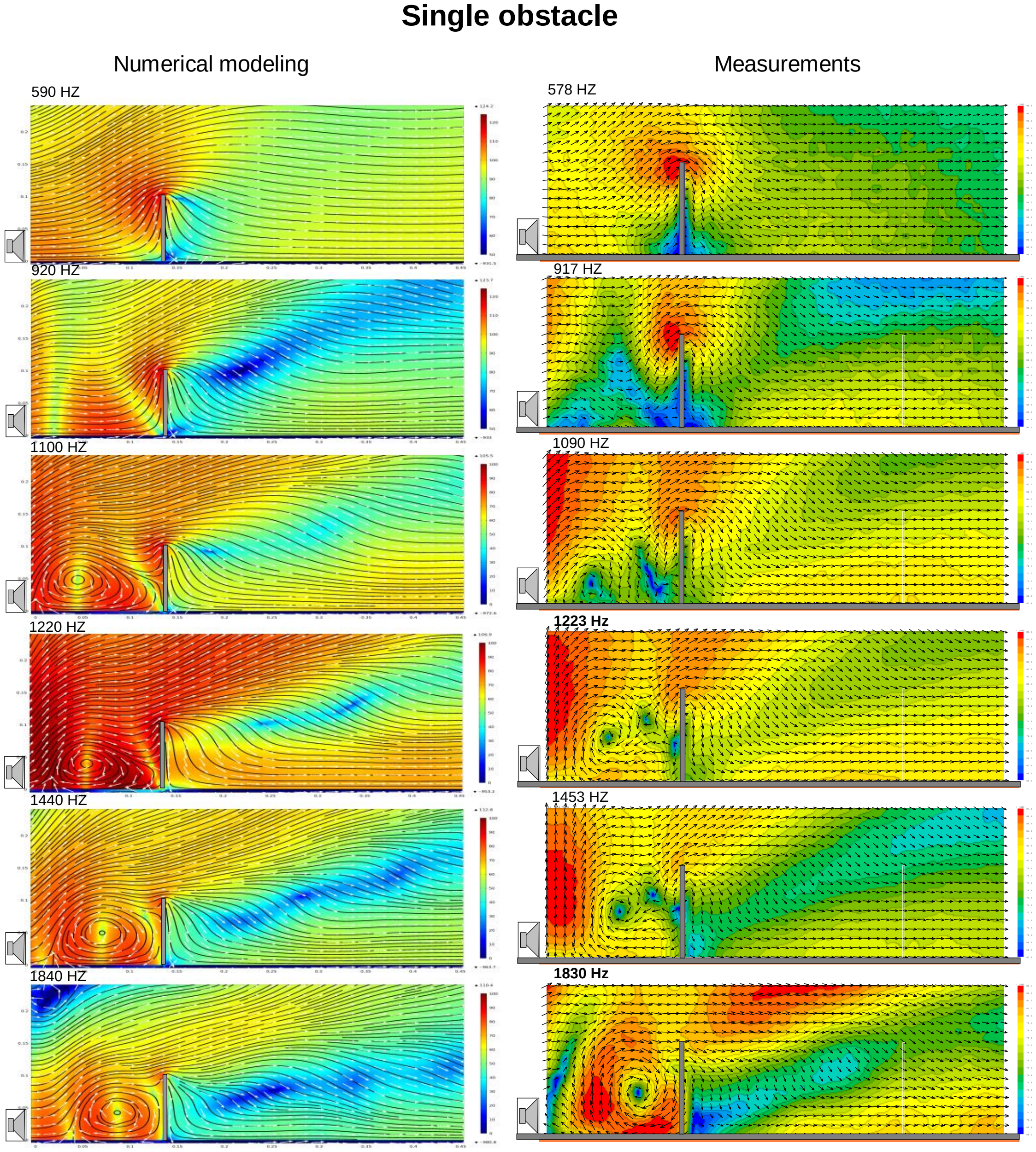

Figure 2. *Configuration of the model and distribution of computational grid used in the numerical model – single obstacle*

During the tests inside-semi anechoic chamber, barriers are fixed to a hard substrate dimension 2,0 m x 1,2 m. The line acoustic source is about 0.5 m before the first barrier. All the barriers are 100 mm high and 1.0 m long, and thickness 2 mm. Cavity between barriers are 250 mm long (dimension of cavity between barriers: L=250 mm, D=100 mm, L/D=2,5). For the sound intensity measurement the 3D-USP type miniature intensity probe is applied. It is a very compact and integrated sound intensity transducer combines three orthogonal positioned particle velocity sensors and a miniature (0,1 inch) pressure microphone. The actual 3D sensor configuration without its cap is less than 5 mm x 5 mm x 5 mm.

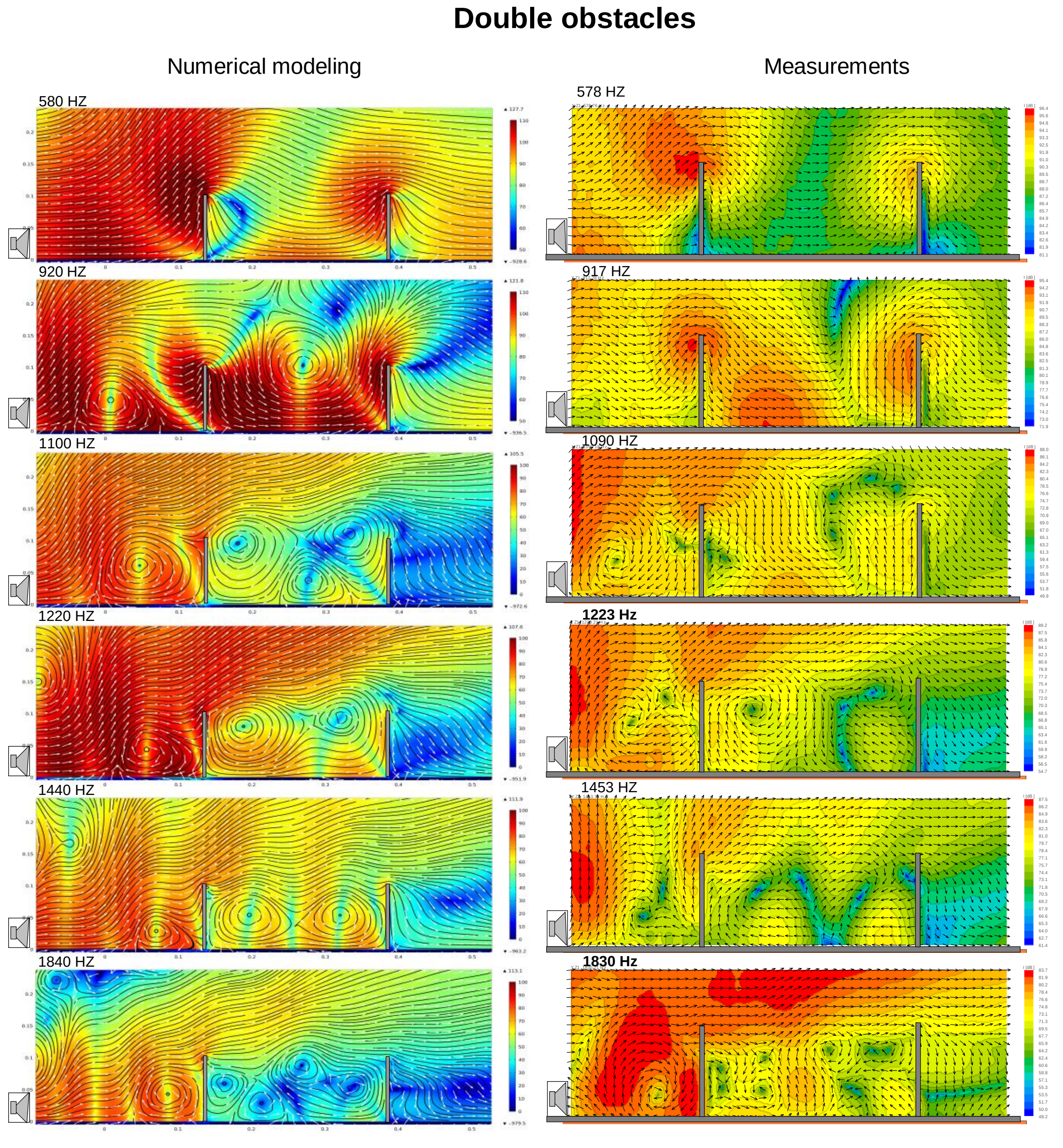

Figure 3. *Comparison of the results of numerical modeling with experimental results using the sound intensity technique – double obstacles*

Shapes of intensity streamlines describing the reaction of acoustic flow wave strike the thin single end double thin barriers shown in Fig. 2 and Fig. 3. This is a comparative test what sort of reaction can we observe while a running acoustic wave hit an single obstacle or a series of obstacles arranged in palisades. We can show in this picture that for the frequency band less than about 600 Hz we cannot see any big perturbation inside the acoustic flow field, but in highness frequency region there are some significant distortions of the sound intensity field in the form of vortex rotation.

NUMERICAL MODELING

Modeling the wave impingement noise is non-trivial exercise. Many times, sufficient resolution of the local turbulence, complicated by the varying gas temperature and often complex geometry is necessary. Modeling this phenomenon requires representation of both the fluid and structure elements as well as their interaction. The feasibility of modeling this phenomenon can be done using CFD (*Computational Fluid Dynamics*), a numerical simulation with FSI (*Fluid Structure Interaction*) methods and/or commercial code with both FSI and CAA (*Computational Aero-Acoustic*). Both fluid and structural dynamics need to be modeled and are described by different sets of equations. Once the flow and structure interaction was solved, the surface velocity at the structure can be post processed to calculate the sound pressure level at any predefined locations.

We must remember that the numerical modeling should be validate using experimental investigation on the properly model or the real structure. In our research, numerical simulation was performed in *Acoustic Module* of COMSOL Multiphysics 4.0a. The software employs *Finite Element Method* for solving acoustic wave propagation, described by linearized fluid and solid dynamics equations. The *Pressure Acoustic Interface* and *Frequency Domain Study* was used. For modeling of acoustic phenomena, Helmholtz's equation (3) for lossless, inhomogeneous medium was used:

$$\nabla\left[-\frac{1}{\rho_0}\cdot\left(\nabla p - q\right)\right] - \frac{\omega^2 p}{\rho_0 \cdot c_s^2} = Q \qquad 3)$$

where:

- ρ_0 - density of the medium, [kg/m^3]
- c_s - speed of sound, [m/s]
- q - dipole source
- Q - monopole source
- $\omega = 2\pi f$ - circular frequency, [rad/s]

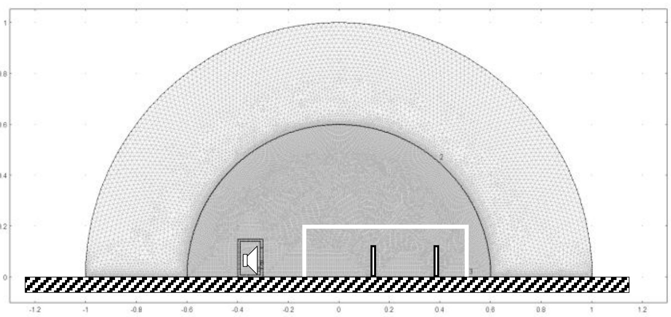

Figure 4. *Configuration of the model and distribution of computational grid used in the numerical model*

The frequency response was computed with parametric sweep over a frequency range using a harmonic load. The model was designed as a two-dimensional structure. On the rectangular plate dipole source was placed in the cabinet (to the modeled space radiates only one side of the source) and two flat obstacles. Numerical study area was modeled in form of a semi-circle whose base coincides with the surface of the plate (Fig. 3). To minimize the impact of the shape of the modeled space on the phenomena occurring inside that space, the semi-ring of the Perfectly Matched Layer (*PML*) elements was used. Model was meshed with the use of *Free Triangular* elements. The maximum and minimum size of PML elements defined respectively as 0.02 m and 0.01 m. The rest of model was meshed with the use of elements with size in range from 0.0008 m to 0.005 m. Complete mesh of model with one obstacle consists of 89430 elements and model with two obstacles consist 89824 elements.

CONCLUSION

The verifying tests using an intensity technique and numerical modeling may show how much the theoretical image of an acoustic field distribution is differing from the distributions obtained through the measurements in real conditions. The degree of discrepancy between the predicted numerical results and the real structure of the field formed over barriers, grows proportionally to the degree to which the simplified calculated assumptions differ from the conditions encountered in reality. The differences mainly result from either the fact that theoretical forecasting uses too sweeping simplifications or that it is impossible to obtain proper data on real physical features of the tested area.

Although the numerical model was not very accurate, comparing to the physical model for the same frequencies (see Fig. 3) we can conclude that the modeling results are encouraging. Further studies will be aimed at finding numerical modeling tools (*CAA*) which will be more similar to acoustic flows examined with experimental methods.

References

Fahy F. J., (1989) - *Sound Intensity*, London, Elsevier Applied Science.

Pyła M., Weyna S., (2010) - An Experimental Study of Sound Intensity Flow Fields, Acta Acustica united with Acustica, Vol. 96 Supl. 1, S19-6.

de Roeck W., Desmet W., Baelmans M., Sas P., (2004) - On the prediction near-field cavity flow noise using different CAA techniques, ISMA, 369-388.