Numerical study of the acoustic efficiency of a group of Helmholtz resonators of various configurations

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Abstract. In this work the physical and mathematical models of predicting the effective acoustic properties of sound absorbing structures (Helmholtz resonators) in joint operation were formulated. Numerical simulation has been performed on the modeling of an acoustic wave in a model channel with resonators of various configurations. Research was carried out to optimize the mutual arrangement of Helmholtz cells (resonators) in sound-absorbing structures of resonant type. According to the results of the research, the mutual influence of closely located prismatic resonators in the model channel of rectangular shape was revealed. The most effective combinations of prismatic resonators were determined. Schemes and recommendations for the placement of composite and base resonators in sound-absorbing structures were developed. Unique single-layer composite sound-absorbing structures, working at several resonant frequencies, were developed.

Introduction

Nowadays, the ecology of aviation transport has become the most urgent problem for civil aviation, according to the International Civil Aviation Organization (ICAO), second only to flight safety. One of the major components of the harmful impact of aviation on the environment is the noise created by aircraft. One of the most effective ways to reduce the noise of aircraft propulsion systems is to include sound-absorbing structures (SAS) in its composition [1]. SAS represent a set of Helmholtz resonators, allocated in the form of a honeycomb structure made of polymer composite materials. Such SASs are traditionally installed on the inside surface of the air intake of an aircraft engine to reduce noise propagating into the front hemisphere and on the walls of the channel of the outer contour of the engine to reduce noise propagating into the rear hemisphere [2].

Multilayer honeycomb panels of resonance type SAS allow to reduce the noise in a wide range of frequencies, thanks to the operation of resonators tuned to different frequencies [1]. At the first stage of the SAS developing process, the direct and inverse problems of determining the impedance by geometric parameters were solved. At the second stage, on the basis of the obtained values, the required geometric parameters of the SAS were chosen for a given impedance. At the same time, it should be noted that the approaches currently used in the design of the SAS are generally based on semi-empirical models that are obtained with the use of serious simplifications and take little or no account of many factors affecting sound attenuation [3]. Experimental studies of the developed SAS are also limited to testing only small-

sized samples. As a result, when creating a full-scale SAS design, its acoustic characteristics in practice significantly differ from those predicted in the design [4]. This requires a long and expensive modification of the design already at the stage of its introduction into serial production [5]. Therefore, it becomes necessary to carry out multifactor mathematical modeling of acoustic processes. It is of interest to identify the new ways to control the impedance of cells and SAS, including control over the shape and volume of cavity chambers, the design of the throat and perforations, the arrangement of resonators [6-9].

Several papers that deal with specific aspects of this problem have been published [10-11]. However, in general, the issue has not been studied enough.

The prototype of resonant SAS is a classic Helmholtz resonator [12], consisting of an air cavity connected by a hole or a neck with the surrounding air. If resonator size is small compared to the sound wavelength, then it can be considered as an oscillatory system with one degree of freedom (1-DOF), which mass is the mass of air in the neck together with the mass of attached air oscillating around its air cavity while the air in the cavity serves as an elastic element [13, 14].

Such a system has its own frequency:

$$f_0 = \frac{c}{2\pi} \sqrt{\frac{a^2\pi}{l_c V}}, \quad (1)$$

Where c is the speed of sound; a - the radius of neck cross-section; l is the neck length; V - cavity volume.

The Helmholtz resonator reduces the sound intensity at a particular oscillation frequency (resonant frequency) [12, 15, 16]. Active sound absorption of plating covers a wider frequency range due to the technological features of SAS manufacturing, however, the maximum noise

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reduction using SAS is also observed at a certain oscillation frequency, which is tuned to. The principle of SAS resonator operation is based on the absorption (dissipation) effects and incident sound wave reflection by the resonator. In the first case, the incident wave excites air oscillations in the resonator cavity, leading to absorption (transition of energy to translational degrees of freedom) [17, 18]. In the second case, the wave entering the resonator changes its phase to reverse due to the double passage in it, and, adding to the primary wave emitted by the source, dampens the latter [11].

In studies [19, 20] A. Selamet and others found the influence of flat and non-planar wave propagation on the resonant frequency and the attenuation of waves in concentric Helmholtz resonators when resonator size changes.

The study of the acoustic efficiency of closely spaced resonators has been research subject by many authors. An experimental study of the mutual influence of closely spaced resonators was carried out in [21, 22]. It was found that when the distance between resonators necks is smaller than the wavelength, the efficiency of resonators pair is less than that of a single one. In [22], mutual influence of 2 or more cylindrical resonators was studied. In [23], a numerical study of closely spaced cylindrical resonators interaction was performed. The optimal distances between the resonators, as well as the geometric characteristics of the neck were determined.

Usage of 1-DOF resonators group is a special interest. 2-DOF [24], as well as the study of their joint work presented in [25, 9]. The authors presented the optimal location system for two different Helmholtz 1-DOF resonators as a result of the research. A study on the relative distance, orientation and geometry of the resonators was conducted. Also in subsequent work it was revealed that two identical 2-DOF-resonator systems provide a relatively wider attenuation range for a bandwidth of 300 Hz, compared to previously published studies for single 2-DOF, single 1-DOF and two identical 1-DOF.

The purpose of this research is to develop mathematical models, computational algorithms for predicting effective acoustic properties, automated calculation and optimal design of honeycomb composite structures. The results of the research will make it possible to establish the patterns of distribution of the acoustic pressure fields in the model channel with various configurations in the arrangement and shapes of the aggregate cells and will allow to reveal significant factors influencing the acoustic efficiency of the SAS. The obtained results will allow developing new SAS for future aircraft engines.

1 Numerical model

Within the framework of the numerical simulation, the interaction of resonators of a prismatic shape with different volumes was investigated. The value of the damping effect produced by Helmholtz resonators in a joint operation was calculated in the operating frequency

range of 100-3000 Hz on the basis of a numerical solution of the Helmholtz equation [7].

For carrying out a number of numerical simulation, three geometric models were constructed (A 1-3). All models of the group contain the "base" prismatic resonator with the following characteristics: resonator height 10 mm, surface length of hexagon 6 mm, diameter and neck height 1 mm. As well as a second composite resonator, for which the following volume ratios (relative to the base volume) were taken: A1 1/3 for the upper and 2/3 for the lower one; A2 1/2 for the upper and 1/2 for the lower; A3 2/3 for the upper and 1/3 for the lower one. A general view of the geometric models of the group A 1-3 is shown in Figure 1.

Fig. 1. Geometric characteristics of models: a - A1; b - A2; c - A3.

To improve the convergence of the solution and reduce the errors in the results obtained, a computational grid where cells had a shape close to the shape of an equilateral tetrahedron was used. The maximum element size was defined as $N_{\max} = 343 \text{ [m/s]} / 6 \text{ [kHz]} / 10 = 0.0057 \text{ m}$, the minimum element size was assumed to be $N_{\min} = 0.001 \text{ m}$, the total number of elements was 700 thousand elements (Figure 2). In addition, when grinding the grid, sharp differences in the geometric dimensions of adjacent cells were avoided: the linear dimensions of neighboring cells do not differ by more than 2 times [11].

Fig. 2. Grid model: a - front view; b - isometry view from below.

It was taken for that the numerical experiment is considered to converge, in the case when the result of numerical simulation does not change with further refinement of the finite element grid [26].

The solution of this problem is performed by solving the Helmholtz equation (2):

$$-\nabla \cdot \left(\frac{\nabla P}{\rho_0} \right) - \frac{\omega^2}{\rho_0 c^2} P = 0, \quad (2)$$

where P is the initial pressure in the calculated region (1 atm.), $\omega = 2\pi\nu$ is the natural frequency of the resonator, ρ_0 is the air density, c is the speed of sound in the medium, and ν is the signal frequency at the input.

The scheme of the boundary conditions given in the numerical calculation is shown in Figure 3.

Fig. 3. Scheme of boundary conditions, where: 1-wave at the input; 2 - wave at the output; 3 - rigid wall.

The input wave was specified as a harmonic pressure wave with amplitude P_0 [27]:

$$n \cdot \left(\frac{\nabla P}{\rho_0} \right) = \frac{i\omega}{\rho_0 c} P - \frac{2i\omega}{\rho_0 c} P_0. \quad (3)$$

The output wave:

$$n \cdot \left(\frac{\nabla P}{\rho_0} \right) = \frac{i\omega}{\rho_0 c} P_0. \quad (4)$$

The boundary condition is the wall (rigid wall):

$$n \cdot \left(\frac{\nabla P_0}{\rho_0} \right) = 0. \quad (5)$$

The determination of loss factor [28, 29]:

$$TL = 10 \log \left(\frac{P_{in}}{P_{out}} \right), \quad (6)$$

where:

$$P_{in} = \int_{\partial\Omega} \frac{p_0^2}{2\rho_0 c_s} dA, \quad (7)$$

$$P_{out} = \int_{\partial\Omega} \frac{|p_c|^2}{2\rho_0 c_s} dA. \quad (8)$$

The verification of the presented numerical model was carried out on the example of a SAS with cellular cells and is given in [30].

2 Calculation of acoustic efficiency

Based on the results of the numerical simulation, the dependences of the acoustic transmission loss (TL) on the frequency were obtained, as well as the distribution of the acoustic pressure fields. When reviewing the results of numerical simulation obtained from the models of group A 1-3, it was found that the largest value of the acoustic transmission loss coefficient (TL) was shown by the geometry A1. $TL_{max} = 43$ dB at a joint resonant frequency of a composite resonator of 1023.5 Hz, $TL_{max} = 43$ dB at a resonant frequency of the base resonator of 1183 Hz and $TL_{max} = 13.4$ dB at the third resonant frequency of 3274 Hz (Figure 4).

Fig. 4. Field distribution of acoustic pressure along the longitudinal section C-C of the model channel (resonator region) for the design variant A 1: a - the resonance frequency of 1183 Hz; b - resonant frequency of 1023.5 Hz; c - resonance frequency 3274 Hz.

The lowest value of the coefficient of acoustic pressure loss is observed for the A2 geometry (Figure 5). $TL_{max} = 21.4$ dB at a joint resonant frequency of a composite resonator of 1097 Hz, $TL_{max} = 43$ dB at a resonant frequency of the base resonator of 1183 Hz and $TL_{max} = 14.8$ dB at the third resonant frequency of 2882 Hz.

a

b

c

Fig. 5. The acoustic pressure distribution field along the longitudinal section C-C of the model channel (resonator region) for the design variant A2: a - the resonance frequency of 1183 Hz; b - resonant frequency of 1097 Hz; c - resonant frequency of 2882 Hz.

When considering the results obtained for the A3 geometry (Figure 6), a significant decrease in the acoustic efficiency of the composite resonator at its joint (least) frequency was observed, in comparison with the A1 geometry, and also a negligible decrease at the third resonant frequency. The value of the acoustic pressure loss coefficient was for the base $Tl_{max} = 43$ dB at the resonant frequency 1183, for the combined frequency of the composite resonator $Tl_{max} = 32.9$ dB at the resonant frequency of 1172 Hz and $Tl_{max} = 10.2$ dB at the third resonant frequency of 3278 Hz.

a

b

c

Fig. 6. The acoustic pressure distribution field along the longitudinal section C-C of the model channel (resonator region) for the design variant A3: a - the resonance frequency of 1183 Hz; b - resonant frequency of 1172 Hz; c - resonant frequency 2856 Hz.

A comparative analysis of the results obtained for the resonators under consideration (Table 1) shows that the combination of resonators A1 is most preferable, at this combination the highest values of the acoustic pressure loss coefficient are observed both at the joint resonator frequency are observed, and at the frequency of the base resonator.

Table 1. Results of numerical simulation.

Calculated variation		Resonance frequency, Hz (calculation)	Resonance frequency, Hz (analytics)	TL, dB
A1		1023.5		43
		1183	1186.8	43
		3274		13.4
A2		1097		21.4
		1183	1186.8	43
		2882		14.8
A3		1172		32.9
		1183	1186.8	43
		2856		10.2

Thus, having reviewed the results obtained for the resonator groups under consideration, it can be concluded that the most effective combination for the "composite" and "base" resonators is the combination in which the resonator of a smaller volume is at the top. So the efficiency in the joint operation of resonators is affected by the volume and shape. It should be emphasized that the use of other forms, for example "biconical" as the "base" resonator, will increase the efficiency of the system. In addition, it should be noted that when the resonators of different volumes work together, not only the efficiency of the "base" resonator increases, but also the broadband of the group.

Conclusion

Unique single-layer compound SAS operating at several resonant frequencies have been developed as part of the research, as well as allocation layout of compound and base resonators in SAS. According to the results of numerical studies, it was found that the joint action of resonators of different volume increases the efficiency of the "base" resonator and the group bandwidth while the largest value of the acoustic pressure loss coefficient for compound resonators is observed at the joint frequency. The analysis of the obtained results revealed that the most effective combination for "compound" resonators is the one wherein a smaller-volume resonator is located on top. The maximum decrease in acoustic pressure in the case of resonators combination (single and composite) is observed. The maximum decrease in acoustic pressure was also revealed with a combination of a prismatic resonator and a "compound" (two-layer)

one. Based on the study, we can conclude that a combination of resonators will reduce the mutual influence of resonators at their common frequency, and also increase the bandwidth of resonators group, as well as for cylindrical resonators group. It is also determined that the most effective combination for "compound" resonators is one wherein resonator of smaller volume is located on top.

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