Paper No.337/2021

OPTIMIZATION OF COMPLEXLY SHAPED DISSIPATIVE SILENCERS

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***Abstract**: The article is about a comprehensive study of dissipative silencers of complex shapes. Analyzed the acoustic efficiency of such silencers. The sound absorption coefficient determined from experimental data. The acoustic properties of plate silencers take into account that coefficient. Presented the results of verification of models with plates of constant thickness. Considered the plates of complex shapes: uniform thickness change, convex and concave. Obtained results of modeling the acoustic efficiency of silencers with complex shape plates. These results are for different values of the mean integral thickness for the octave bands frequencies. Determined the dependence of the length of complex shapes silencers on their mean integral thickness. Presented the results of mathematical modeling of the airflow in silencers to determine their aerodynamic drag. The calculations showed that there is an optimal shape and mean integral thickness of the plates. Under which conditions the aerodynamic resistance of the silencer is minimal.*

***Keywords:** Dissipative silencer, acoustic modeling, aerodynamic modeling, silencer's optimal shape*

***DOI:** 10.36336/akustika20213994*

1. INTRODUCTION

The physical impact of energy equipment in the form of noise has a negative impact on human health. Today in Russia thermal power plants (TPPs) are the main source of electricity. Also, in large cities they are main source of heat. Thermal power plants make a significant contribution to noise pollution in cities. Noise from heat power facilities can have different frequency and time characteristics [1]. The acoustic impact of TPP facilities is the reason for exceeding sanitary standards in the surrounding area [1].

An effective way to reduce the noise of power equipment is the installation of dissipative silencers [2]. The ability to reduce noise in such silencer bases on the conversion energy. The sound energy converts into thermal energy when sound waves interact with sound-absorbing material [3]. Dissipative silencers use to reduce noise from blowing machines, ventilation ducts, steam emissions and other sources of intense noise.

Dissipative plate silencers use for noise reduction in power equipment. The main elements of the plate are a sound-absorbing material, a perforated sheet designed to protect the sound-absorbing material from blowing out, inlet and outlet fairings that reduce the aerodynamic drag of the plate and elements that impart rigidity to the structure: frames of plates and cassettes [4].

The effectiveness of silencers determines not only by the required noise reduction, but also by the least aerodynamic drag [2]. Various methods apply to reduce the aerodynamic drag of silencers. The work [5] shows that to reduce the drag of the plates, the installation of output wedge-shaped fairings (instead of cylindrical ones) uses. It makes possible to reduce the aerodynamic drag by 10-17%. It is also possible to use special guide vanes located in a turn up to the installation site of the plate dissipative silencers to reduce drag [6]. This leads to a decrease in the silencer's aerodynamic drag by 14-20%.

The purpose of this article is to analyze the possibility of using silencers of complex shapes to reduce the noise of power equipment. The use of such plates leads to a decrease in both the aerodynamic resistance of the silencer due to the appearance of a diffuser section. Also, the acoustic efficiency of the silencer due to a decrease in the average integral thickness of the plates. To reduce noise by the required amount, it is necessary to increase the length of the plates. But it leads to an increase in friction losses [4]. So, there is an optimal value of the mean integral thickness of the plate, at which the aerodynamic drag of the silencer is minimal.

But it is not possible to test the acoustic efficiency of silencers of complex shapes using empirical formulas. In this case, there is a need for mathematical modeling to determine the acoustic efficiency of such plates.

The works [7-9] present the results of modeling silencers in the Ansys ACT Acoustic program. Authors of these works set the acoustic characteristics of sound-absorbing materials. The Johnson-Champoux-Allard model uses to describe the properties of the sound-absorbing material. According to this model the sound attenuation in the silencer determines by the visco-inertial properties of porous sound-absorbing materials.

The articles [10-12] provide an analysis of various methods for determining the acoustic efficiency of silencers. The acoustic characteristics of various sound-absorbing materials investigates. Also, the influence of material properties on the efficiency of silencers determines.

In this article, to describe the acoustic properties of plate dissipative silencers, the sound absorption coefficient uses [3]. The sound absorption coefficient depends on the porosity of the sound-absorbing material, density and thickness.

2. RESULTS OF ACOUSTIC MODELING

2.1. Modeling silencers with constant thickness plates

The main task of modeling silencers is to determine the losses in the transmission of sound energy, which calculates according to the following formula:

$$\mathbf{TL = 10 \cdot lg\left(\frac{W_{in}}{W_{tr}}\right)} \tag{1}$$

where
W_{in} – incoming sound energy, dB;
W_{tr} – transmitted sound energy, dB.

The purpose of modeling silencers of constant cross-section is to verify acoustic models with experimental data [3]. It is necessary for the possibility of their use in calculating silencers of complex shapes. The acoustic efficiency of plate silencers of constant thickness 100, 150, 200 mm determined. The length of the plates in all cases is 1000 mm. The calculation simulates the propagation of airborne noise through the channel at the location of the plate dissipative silencers.

The main quantity characterizing the acoustic properties of the sound-absorbing material is the sound absorption coefficient. In practice, two types of sound absorption coefficient are: real and equivalent. Calculations perform in Ansys ACT Acoustic. The real absorption coefficient uses when specifying the Absorption surface boundary condition. According to [3], the real sound absorption coefficient for the octave band frequencies for plates of constant thickness 100, 150, 200 mm determines. These values use when setting the Absorption surface boundary condition.

In the example there is no reflection of sound waves from the input and output sections of the channel. Thus, in order to perform the simulation, a wave absorption condition specifies. This condition assumes the absence of reflection of incoming and outgoing waves from the walls.

The boundary condition for the normal surface velocity of a plane wave sets as the sound source. The condition sets at the inlet surface of the channel, where plate dissipative silencers installed (Fig.1).

Based on the results of mathematical modeling, a decrease in sound pressure levels in silencers with a plate thickness of 100, 150, 200 mm obtained for octave band frequencies of 31.5-8000 Hz. Fig. 1 and Fig. 2 show a comparison of the results of harmonic analysis and experimental data [2] for plates with a thickness of 100 and 150 mm.

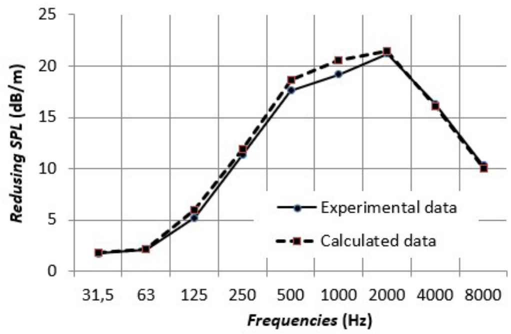

Fig. 1: Reducing the sound pressure level in a silencer with a plate thickness of 100 mm

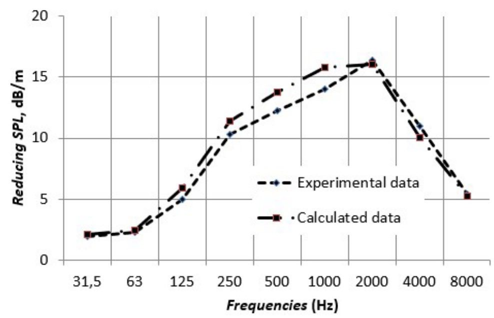

Fig. 2: Reducing the sound pressure level in a silencer with a plate thickness of 150 mm

The results show good agreement between the experimental and calculated data. For plates with a thickness of 100 and 150 mm, the maximum difference in the decrease in the sound pressure level in the channels of the plate silencer does not exceed 1.6-1.8 dB at octave band frequencies 500-1000 Hz. The results confirm the possibility of using the mathematical models to determine the attenuation of noise in silencers of complex shapes.

2.2. Complex shapes silencers modeling

Determined the loss of sound energy in plate dissipative silencers using mathematical modeling. Modeled silencers of complex shapes during the propagation of air noise from a sound source.

Modeled three forms of plates: concave, convex, and plates with a uniform change of thickness from the inlet section to the outlet (Fig. 3). The thickness of the plates in the inlet is 200 mm. The thickness of the convex and concave plates in the outlet is 100 mm. To compare the results for plates of different shapes, we chose a single criterion. This criterion is the average integral thickness. With the same average integral thickness and the thickness of the inlet, the thickness of the outlet will vary for different plate shapes. To determine the specific noise attenuation per length of the silencer, the length of the plates is 1000 mm. In a rectangular channel with dimensions of 800x800 mm were installed two plates.

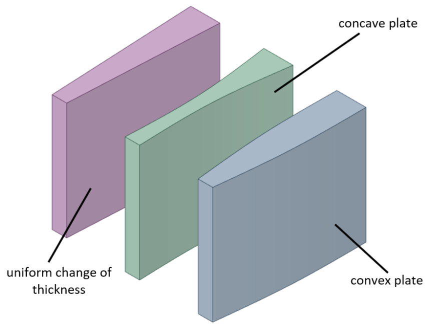

Fig. 3: Models of plates of complex shapes

In the case of uniform change of plate thickness, the actual sound absorption coefficient is the average in the silencer at the inlet and outlet. The value of the actual sound absorption coefficient is the arithmetic mean:

$$\alpha_r = 0{,}5 \cdot (\alpha_{inp} + \alpha_{out}) \quad (2)$$

where

α_{inp} – sound absorption coefficient in the inlet,
α_{out} – sound absorption coefficient in the outlet.

For concave and convex plates formula 2 is incorrect. In this case, the average integral thickness of the plates has different values. For this reason, the change of plate thickness from inlet to outlet of such plates we described by function.

The average integral thickness of concave and convex plates is:

$$\bar{\delta} = \frac{1}{b-a} \cdot \int_a^b \delta(x) \cdot dx \quad (3)$$

where

b = 1000 mm;
a = 0 mm – the limits of integration,
$\delta(x)$ – functional dependence of plate thickness changes.

The values of the average integral thickness of convex and concave plates are in Tab. 1.

Mean integral concave plates thickness, mm	Mean integral convex plates thickness, mm
99	163
112	175
125	188
137	201

Tab. 1: Values of mean integral thickness

The values of the actual sound absorption coefficient determined by calculated data [3]. The boundary condition „Absorption surface“ used these values.

Based on the results of mathematical modeling, plotted a graph of the reduction of sound pressure levels in silencers. This graph is for plates of complex shapes for the octave band frequencies 31.5-8000 Hz and different values of the mean integral thickness (Fig. 4).

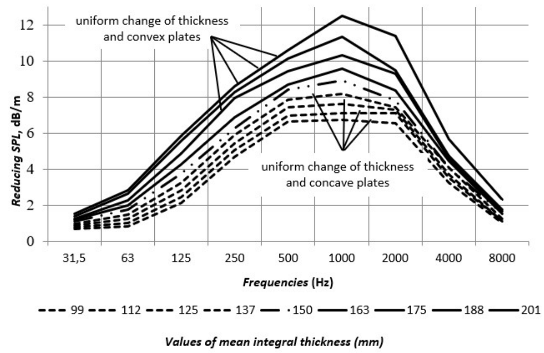

Fig. 4: Reducing the sound pressure level in silencers with a different integral thickness

The acoustic efficiency of plates with the same mean integral thickness has the same value and does not depend on the shape of the plate. The largest value of noise attenuation is at a frequency of 1000 Hz. With an increase in the mean integral thickness of the plates, the acoustic efficiency of the silencer increases. This is due to a decrease in the cross-section ratio.

Fig. 5 visualizes the processes of sound absorption in the channels of plate silencers. On it, there is the field of sound pressure reduction for convex plates with a mean integral thickness of 175 mm at a frequency of 1000 Hz.

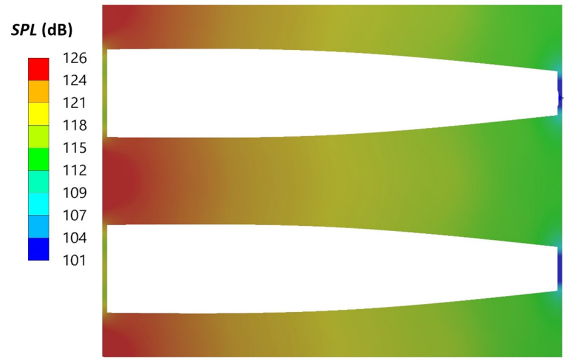

Fig. 5: Reducing the sound pressure at a frequency of 1000 Hz

3. PLATE LENGTHS CALCULATION

In this chapter, we define the required lengths of the plates considered in chapter 2. The calculation of the power plant model in the Predictor program gives the required noise reduction of fans [1]. The values of the required noise reduction for the octave frequency band are in Tab. 2.

octave band frequency, Hz	31,5	63	125	250	500	1000	2000	4000	8000
required noise reduction, dB	0	0	6,2	15,0	19,9	22,2	12,2	0	0

Tab. 2: Required noise reduction

When calculating the length of the plates, it is necessary to take into account regional climatic factors. The authors [1] showed that due to changes in temperature and humidity during the year, the change in excess can be 1.7 dBA.

The length of the silencer, taking into account the correction, is:

$$l_{sil} = \frac{\Delta L_{req}+c}{\Delta L_{spec}} \quad (4)$$

where

ΔL_{req} – required noise reduction (Tab. 2);
$c = 1.7$ **dB** – takes into account the influence of regional climate factors;
ΔL_{spec} – the specific noise reduction per length of the silencer (defined in chapter 2).

The lengths of the plates of silencers of complex shapes are equal to the greatest values. For all the considered plate shapes, the greatest length values are at low (250 Hz) and medium (500-1000 Hz) frequencies.

A curve (Fig. 6) is a result of the calculations of the change in the length of the plates from the mean integral thickness.

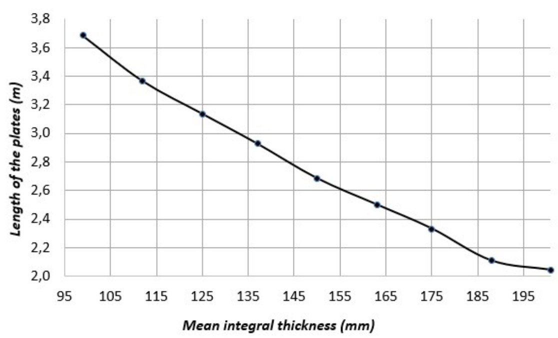

Fig. 6: Changing the length of the plates

With an increase in the mean integral plate thickness, the length of decreases due to an increase in noise attenuation. When changing the thickness from 99 mm to 201 mm, the length reduces by 1.7 m.

4. AERODYNAMIC MODELING RESULTS

By means of SolidWorks software, we determined the aerodynamic drag of plates of complex shapes. To perform the simulation, the values of the airflow and ambient pressure are set at the inlet and outlet of the air channel (Fig. 7).

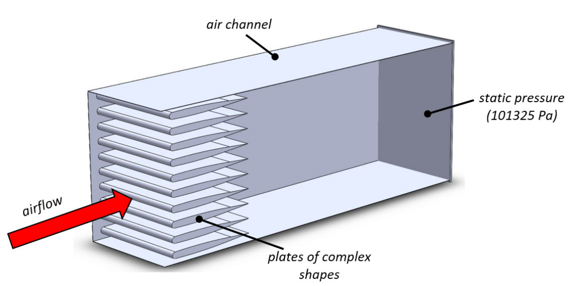

Fig. 7: Modeled section of the air channel

Plot at Fig. 8 shows the dependence of the plate's aerodynamic drag on the mean integral thickness.

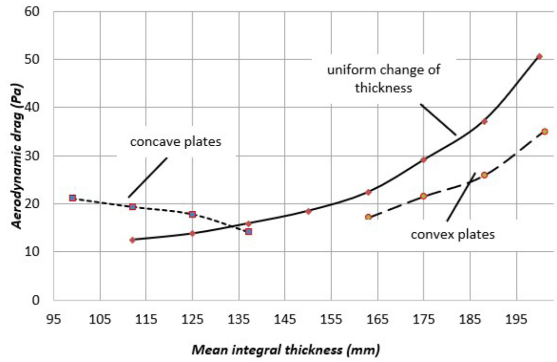

Fig. 8: Change in aerodynamic drag

The aerodynamic drag of uniform thickness change plates and convex plates decreases with a decrease of the mean integral thickness. This is due to an increase in the length of the diffuser section. The curve of change in the aerodynamic drag of concave plates is the opposite. With an increase in the mean integral thickness, the aerodynamic drag of such plates increases. Despite the presence of a diffuser section, due to the profile and the longer length of the concave plates, friction losses in this case prevail.

The plates with a uniform thickness change (12.5 Pa) have the lowest aerodynamic drag at an average integral thickness of 112 mm. Compared to concave plates, the outlet edge of these plates has a smaller thickness (24 mm). Between the channels of the silencer, there is a stronger flow deceleration (a large length of the diffuser section) and a decrease in aerodynamic drag. There is takes place transformation of the dynamic pressure converted into a static one [13].

Due to the profile of convex plates, their aerodynamic resistance is lower than that of plates with a uniform thickness change at mean integral thicknesses of 163-201 mm. The convex profile of the plate provides a large amount of diffuser area and less aerodynamic drag.

That it is most appropriate to use plates with a uniform thickness change with a mean integral thickness of 112 mm. Such plates provide the least aerodynamic drag (12.5 Pa).

5. CONCLUSION

The authors have shown that there are optimal designs of plate dissipative silencers of complex shapes. The design of modern silencers should provide not only the greatest acoustic efficiency. Also, the least impact on the equipment is important. The following conclusions based on the results of the study:

1. The complex shapes silencers acoustic efficiency calculation by empirical formulas is not possible. To determine the acoustic efficiency of complex plates, you should use mathematical modeling.
2. The authors have developed models of traditional silencers and new generation silencers. The simulation results show the convergence of the experimental and calculated data.

3. The mean integral thickness of the plates determines the acoustic efficiency of silencers. All other things being equal, silencers with the same mean integral thickness have the same efficiency. It does not depend on the shape of the plate.
4. Plates with a lower mean integral thickness have a larger required length. This is due to an increase in noise attenuation. For example, increasing the plate thickness from 99 mm to 201 mm leads to reducing the required length of the silencer by 1.7 m.
5. Different designs of silencers with the same efficiency differ in the values of aerodynamic drag.
6. The use of complex-shaped silencers developed by the authors reduces the aerodynamic drag by 75%.

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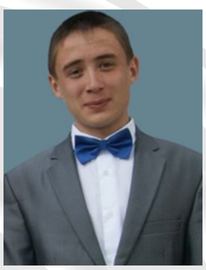
He presented the main results of scientific research at the international conferences in in Moscow, St. Petersburg, Samara, Harbin (China).



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