The Disk Radiator for the Influence on the Gas Media

Vladimir KHMELEV
Biysk Technological Institute (branch) of Altay State Technical University named after I.I. Polzunov, Biysk, vnh@bti.secna.ru

Andrey SHALUNOV
Biysk Technological Institute (branch) of Altay State Technical University named after I.I. Polzunov, Biysk, shalunov@bti.secna.ru

Roman GOLYKH
Biysk Technological Institute (branch) of Altay State Technical University named after I.I. Polzunov, Biysk, grn@bti.secna.ru

Viktor NESTEROV
Biysk Technological Institute (branch) of Altay State Technical University named after I.I. Polzunov, Biysk, nva@bti.secna.ru

Abstract: - The article is devoted to the development of efficient ultrasonic disk radiators for the intensification of the processes in the gas media. It is found out, that the main reason of insufficient efficiency of the existing radiators is the reciprocal compensation of vibrations of the opposite phases. The parameters of the radiators developed on the base of the theoretical procedure are revealed experimentally with the help of designed special-purpose test-bench. It is determined, that in near field developed ultrasonic radiators provide maximum level of acoustic pressure of 145 dB, and in far field the level of acoustic pressure is no less than 130 dB.

Keywords: - Ultrasonic, radiator, optimization, phase-shift, direction diagram

1. INTRODUCTION

The ultrasonic action is well-known method of initiation and intensification of the technological processes in the different branches of industry [1–4]. At that as a rule the piezoelectric ultrasonic vibrating systems are applied as a source of ultrasonic action [2–3]. The principle of their action is based on the transformation of electric oscillations of the ultrasonic frequency into the mechanical vibrations of the radiators in the form of the flat plates and the disks [2–3, 5].

For the influence on liquid and solid media the size of the radiator can be limited by the diameter of up to 40…50 mm (the area is ~ $2 \cdot 10^{-3}$ m^2) [6–7], however for the action on gas media the area of the radiator should be essentially enlarged. The reason of this fact is the difference of acoustic resistance of the radiator material and the air medium providing small energy output from the unit of the radiator area. The possible method of the enlargement of the radiating surface area is the use of the radiators made with the form of the flexural-vibrating disks. As the disk area making flexural vibrations is practically unlimited, there is a possibility for the development of the radiators with the surface area of 1 m^2 and more [5].

However at the design [8] of such radiators there is a problem concerning with the presence of the zones vibrating both in-phase with the disk center and in anti-phase with it, Figure 1.

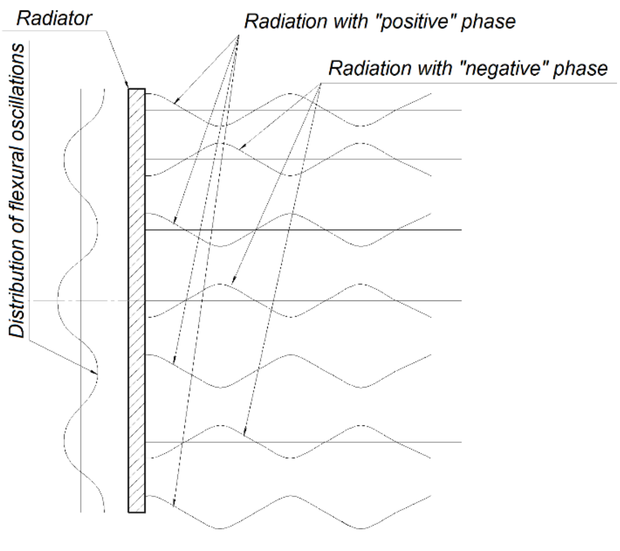


Figure 1. Diagram of the generation of the ultrasonic vibrations by flexural vibrating radiator

As a result of it the wave aggregation with the same phase and the wave aggregation with the opposite phase compensate each other partially or completely at some distance from the radiator. To eliminate undesirable reciprocal compensation of the ultrasonic vibrations it is necessary to decrease the vibration amplitude of the disk radiator zones with one phase and increase the amplitude of the zones with the opposite phase, for instance, by changing the thickness of the corresponding zones [2–4]. At that the task is to determine the disk form providing the generation of the ultrasonic vibrations with maximum possible level of the acoustic pressure.

2. CALCULATION PROCEDURE

To determine optimum form of the disk radiator it is necessary to define the number and the limits of the zones of the disk radiator vibrating in one phase. These zones will be divided by the "nodal circles" (shown in black), in which the vibration amplitude equals zero (Figure 2).

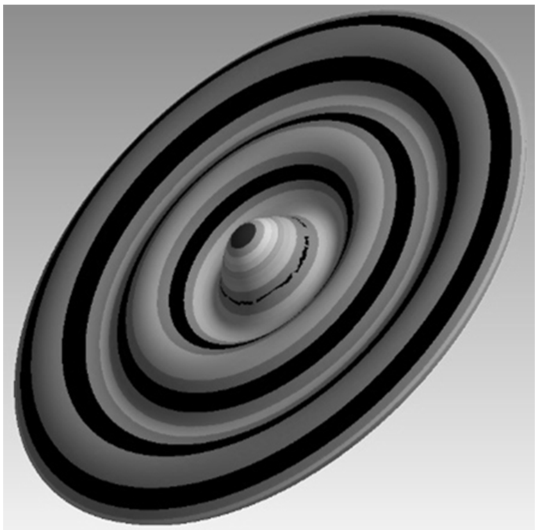


Figure 2. Form of flat disk vibrations

To determine the number of the nodal circles it can be used the following expression [9]:

$$\lambda^2 = 2\pi f r^2 \sqrt{\frac{12\rho(1-\sigma^2)}{Eh^2}}, \qquad (1)$$

where r [m] is the radius of the radiator; h [m] is the thickness of the radiator; E [Pa] is Young modulus of the radiator material; ρ [kg/m^3] is the density of the material; σ [dimensionless quantity] is Poisson ratio.

Further from Table 1 it is chosen the coefficient λ^2, which is the closest to the calculated value, and it is determined the number of the nodal circles corresponding to it. By the substitution of obtained table coefficient into the expression (1) it is defined new improved value of the frequency of the vibration of the disk radiator. If obtained value of the frequency does not satisfy the requirements of the technological process, it is necessary to change the diameter of the radiator. As to decrease frequency of natural vibrations at this mode it is necessary to enlarge the diameter of the disk.

Table 1. The value of λ^2 for the free plate at different numbers of the nodal circles n; σ = 0.33 (for titanium alloys)

	λ^2									
	9.084	38.55	87.80	157.0	245.9	354.6	483.1	631.0	798.6	986.0
n	1	2	3	4	5	6	7	8	9	10

The number of the zones of the disk partition is defined as $n+1$, where n is the number of the nodal circles.

After changing the thickness of the alternating zones of the disk the form of its vibrations and consequently the location of the nodal circles also will change. There appears a task to determine optimum width and thickness of the disk zones.

For this purpose the disk of the stepped variable form is presented as a set of flat zones (rings) of simple form, vibrations of each have been already studied and described in the form of the mathematical equations. Knowing the equations and defining the boundary conditions in the places of junction of the zones with different thicknesses it is possible to describe analytically the vibrations of all stepped variable disk without using the resource-intensive methods, such as finite-element method.

The cross-section of the disk radiator of the stepped variable form is shown in Figure 3. Dashed-dot lines show the places of the disk partition into the ring elements. It can be seen, that the section of the disk radiator is made at every change of its thickness in the way that each ring has permanent thickness along its radius.

The value R_i is determined by the division of the radius value of the disk radiator by the number of the zones of the radiator partition calculated earlier. Natural vibrations of each obtained ring element of constant thickness h are described by the equation [9, 10]:

$$W(r,\theta) = (AJ_n(\frac{kr}{\sqrt{h}}) + BY_n(\frac{kr}{\sqrt{h}}) + CI_n(\frac{kr}{\sqrt{h}}) + DK_n(\frac{kr}{\sqrt{h}}))\cos(n\theta); \qquad (2)$$

where n is the number of the nodal diameters; J_n is Bessel function of the first kind of n- order; Y_n is Bessel function of the second kind of n- order; I_n is modified Bessel function of the first kind of n- order; K_n is modified Bessel function of the second kind of n- order; k is the wave number, m^{-1}; W is the vibration amplitude of the zone with the polar coordinates (r, θ); h [m] is the thickness of the ring element.

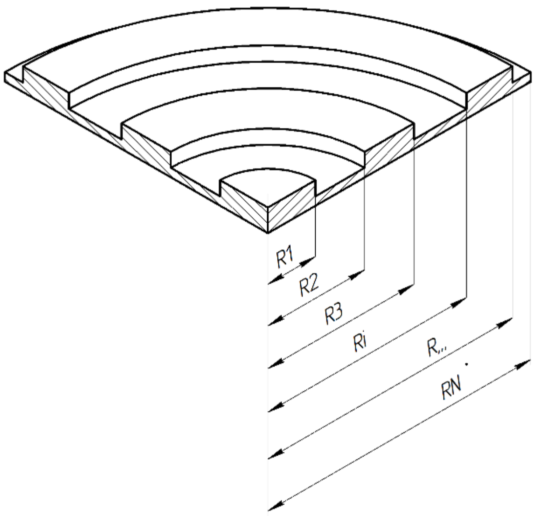


Figure 3. Method of the section of stepped variable disk radiator into the ring elements

Each point of the ring element will make vibrations determined by the following expression:

$$z(r,\theta,t) = W(r,\theta)\sin(2\pi f t); \qquad (3)$$

where t [s] is the time; f [s^{-1}] is the frequency of vibrations; z [m] is the value of the point shift along the axis z, m.

At that the specific form of vibrations depends on the coefficients A, B, C, D being a part of the equation (2), which in turn depends on the conditions on internal and external boundaries of the ring element.

From the practical viewpoint for the radiation of ultrasonic vibrations with maximum efficiency it is necessary to provide ring vibration modes of the disk radiator, i.e. inside obtained ring elements, for which the disk radiator is divided, there should be no points, in which cos(nθ) becomes zero.

That is why further it will be considered only the modes of vibrations, for which $n = 0$, then $\cos(n\theta) = 1$, and the equation (2) takes the form:

$$W(r)=AJ_0\left(\frac{kr}{\sqrt{h}}\right)+BY_0\left(\frac{kr}{\sqrt{h}}\right)+CI_0\left(\frac{kr}{\sqrt{h}}\right)+DK_0\left(\frac{kr}{\sqrt{h}}\right). \qquad (4)$$

By the obtained expressions it can be described vibrations of all ring elements except the central element. The central ring element has internal radius, which equals zero and so it is transformed into the disk of the constant thickness. At that in the center of the disk the values of Bessel functions Y_0 and K_0 turn into infinity, that has no physical sense. That is why these coefficients should be excluded from the equation of vibrations of the ring element by nulling of the coefficients B and D. Then the equation of vibrations of the central elements takes the form:

$$W_1(r)=A_1J_0\left(\frac{kr}{\sqrt{h_1}}\right)+C_1I_0\left(\frac{kr}{\sqrt{h_1}}\right). \qquad (5)$$

Thus, the vibrations of all disk radiator of stepped variable form can be described by the piecewise-defined function (defined for each ring element):

$$W(r)=\begin{cases} A_1J_0\left(\frac{kr}{\sqrt{h_1}}\right)+C_1I_0\left(\frac{kr}{\sqrt{h_{1+}}}\right), \quad 0<r<r_1 \\ A_2J_0\left(\frac{kr}{\sqrt{h_2}}\right)+B_2Y_0\left(\frac{kr}{\sqrt{h_2}}\right)+ \\ +C_2I_0\left(\frac{kr}{\sqrt{h_2}}\right)+D_2K_0\left(\frac{kr}{\sqrt{h_2}}\right), r_1<r<r_2 \\ \dots \\ A_nJ_0\left(\frac{kr}{\sqrt{h_n}}\right)+B_nY_0\left(\frac{kr}{\sqrt{h_n}}\right)+ \\ +C_nI_0\left(\frac{kr}{\sqrt{h_n}}\right)+D_nK_0\left(\frac{kr}{\sqrt{h_n}}\right), r_{n-1}<r<r_n \end{cases}; \qquad (6)$$

Depending on the initial phase φ of vibrations $W(r)$ can have both positive (at φ = 0) and negative (at φ = π) values.

The coefficient k depends on the material, from which the disk is made and it is equal for all ring elements. Thus, to define the form of the natural disk vibrations in each point of the surface it is necessary to determine the values of all coefficients in the equation above. The number of these coefficients equals: $4\cdot(n-1)+2$, where n is the number of the ring elements, which equal to the number of the equations.

In order to determine the coefficients in the equation (6) it is necessary to enter additional ratios. They can be obtained analyzing the conditions of vibrations in the places of the ring elements junction and at free boundary of the last ring element. As in real construction of the disk radiator all ring elements make a whole, they cannot make free vibrations. First of all, the junction means the continuity of the stepped variable disk in this place. For this purpose it is necessary, the values of vibrating shift of the ring elements in the place of junction to be congruent. This condition gives the system of the equations:

$$\begin{cases} W_1(r_1) = W_2(r_1) \\ \dots \\ W_{n-1}(r_{n-1}) = W_n(r_{n-1}) \end{cases}. \tag{7}$$

Due to the rigidity of the material, from which the disk is made, at its vibrations there should be no fractures on its surface:

$$\begin{cases} \frac{dW_1(r)}{dr} = \frac{dW_2(r)}{dr}, \quad r = r_1 \\ \dots \\ \frac{dW_{n-1}(r)}{dr} = \frac{dW_n(r)}{dr}, \quad r = r_{n-1} \end{cases}. \tag{8}$$

According to the third Newton's law at the interaction of two bodies the forces, with which the bodies act at each other, equal in value and opposite in sign. The fact also relates to the moments of forces. Writing two conditions in the form of the formulae, we obtain:

– for the forces:

$$\begin{cases} \left(\frac{d^2W_1(r)}{dr^2} + \frac{\sigma}{r_1}\frac{dW_1(r)}{dr}\right)h_1^3 = \\ -\left(\frac{d^2W_2(r)}{dr^2} + \frac{\sigma}{r_1}\frac{dW_2(r)}{dr}\right)h_2^3, \quad r = r_1 \\ \dots \\ \left(\frac{d^2W_{n-1}(r)}{dr^2} + \frac{\sigma}{r_{n-1}}\frac{dW_{n-1}(r)}{dr}\right)h_{n-1}^3 = \\ -\left(\frac{d^2W_n(r)}{dr^2} + \frac{\sigma}{r_{n-1}}\frac{dW_n(r)}{dr}\right)h_n^3, \quad r = r_{n-1} \end{cases} \tag{9}$$

– for the moments of forces:

$$\begin{cases} \frac{d}{dr}\left(\frac{d^2W_1(r)}{dr^2} + \frac{1}{r_1}\frac{dW_1(r)}{dr}\right)h_1^3 = \\ -\frac{d}{dr}\left(\frac{d^2W_2(r)}{dr^2} + \frac{1}{r_1}\frac{dW_2(r)}{dr}\right)h_2^3, \quad r = r_1 \\ \dots \\ \frac{d}{dr}\left(\frac{d^2W_{n-1}(r)}{dr^2} + \frac{1}{r_{n-1}}\frac{dW_{n-1}(r)}{dr}\right)h_{n-1}^3 = \\ -\frac{d}{dr}\left(\frac{d^2W_n(r)}{dr^2} + \frac{1}{r_{n-1}}\frac{dW_n(r)}{dr}\right)h_n^3, \quad r = r_{n-1} \end{cases} \tag{10}$$

The external boundary of the last ring elements is not connected with anything. It means that no forces act upon it:

$$\left(\frac{d^2W_n(r)}{dr^2} + \frac{\sigma}{r_n}\frac{dW_n(r)}{dr}\right)h_n^3 = 0, \quad r = r_n. \tag{11}$$

$$\frac{d}{dr}\left(\frac{d^2W_n(r)}{dr^2} + \frac{1}{r_n}\frac{dW_n(r)}{dr}\right)h_n^3 = 0, \quad r = r_n. \tag{12}$$

Combining all obtained equations, we get the system of equations (13), by solving it is possible to obtain the values of all coefficients in the initial equation of vibrations of the disk with the stepped variable form. After determining the values of all coefficients, the value of vibrational displacement for any point of the disk surface can be calculated.

$$\begin{aligned} & W_1(r_1) = W_2(r_1) \\ & \dots \\ & W_{n-1}(r_{n-1}) = W_n(r_{n-1}) \\ & \frac{dW_1(r)}{dr} = \frac{dW_2(r)}{dr}, \quad r = r_1 \\ & \dots \\ & \frac{dW_{n-1}(r)}{dr} = \frac{dW_n(r)}{dr}, \quad r = r_{n-1} \\ & \left(\frac{d^2W_1(r)}{dr^2} + \frac{\sigma}{r_1}\frac{dW_1(r)}{dr}\right)h_1^3 = \\ & = -\left(\frac{d^2W_2(r)}{dr^2} + \frac{\sigma}{r_1}\frac{dW_2(r)}{dr}\right)h_2^3, \quad r = r_1 \\ & \dots \\ & \left(\frac{d^2W_{n-1}(r)}{dr^2} + \frac{\sigma}{r_{n-1}}\frac{dW_{n-1}(r)}{dr}\right)h_{n-1}^3 = \\ & = -\left(\frac{d^2W_n(r)}{dr^2} + \frac{\sigma}{r_{n-1}}\frac{dW_n(r)}{dr}\right)h_n^3, \quad r = r_{n-1} \\ & \frac{d}{dr}\left(\frac{d^2W_1(r)}{dr^2} + \frac{1}{r_1}\frac{dW_1(r)}{dr}\right)h_1^3 = \\ & -\frac{d}{dr}\left(\frac{d^2W_2(r)}{dr^2} + \frac{1}{r_1}\frac{dW_2(r)}{dr}\right)h_2^3, \quad r = r_1 \\ & \dots \\ & \frac{d}{dr}\left(\frac{d^2W_{n-1}(r)}{dr^2} + \frac{1}{r_{n-1}}\frac{dW_{n-1}(r)}{dr}\right)h_{n-1}^3 = \\ & = -\frac{d}{dr}\left(\frac{d^2W_n(r)}{dr^2} + \frac{1}{r_{n-1}}\frac{dW_n(r)}{dr}\right)h_n^3, \quad r = r_{n-1} \\ & \left(\frac{d^2W_n(r)}{dr^2} + \frac{\sigma}{r_n}\frac{dW_n(r)}{dr}\right)h_n^3 = 0, \quad r = r_n \\ & \frac{d}{dr}\left(\frac{d^2W_n(r)}{dr^2} + \frac{1}{r_n}\frac{dW_n(r)}{dr}\right)h_n^3 = 0, \quad r = r_n \end{aligned} \tag{13}$$

As an efficiency criterion of calculated disk radiator it is used integral amplitude of the radiator U (m^3):

$$U = \oiint W(r)ds. \tag{14}$$

Given expression takes into account possible reciprocal compensation of vibrations.

Thus, the task of the optimization of the disk radiator is the definition of sizes of the plate zones corresponding to maximum value of the function U

and, consequently, maximum level of the acoustic pressure provided by the radiator.

3. SEQUENCE OF CALCULATION OF THE DISK RADIATOR

Presentation of the integral amplitude value U as a function from the sizes of the plate zones makes possible the use of the numerical methods of the optimization, such methods of gradient descent which are realized in a following way:

1. At the first stage it is chosen the initial value of the parameters, R_1, ..., R_n, (see Figure 3) of required accuracy ε, the values of the gradient step γ.
2. Further it is calculated the form of vibrations $W(r)$ for the current values of the parameters R_1, …, R_n.
3. The value of the function U_0 is calculated.
4. For each parameter R_i:

4.1. It is determined new form of vibrations $W(r)$ for the parameters R_1, ..., R_i+d, ..., R_n, where d is some rather small increment of the parameter.

4.2. The calculation of the value of the function U_i for newly determined form of vibrations $W(r)$.

4.3. It is calculated the value of the partial derivative of the function U by the parameter R_i:

$$\frac{\partial U}{\partial R_i} = \frac{U_i - U_0}{d}. \qquad (15)$$

5. It is calculated the modulus of the parameter changes and compared with the required accuracy:

$$dR = \sqrt{\sum_{i=1}^{n} \left(\frac{\partial U}{\partial R_i} \right)^2}. \qquad (16)$$

If $dR < \varepsilon$, the calculation stops.

6. Each parameter R_i is changed in a following way:

$$R_i = R_i - \gamma \frac{\partial U}{\partial R_i}. \qquad (17)$$

7. It is carried out the transition to the paragraph 2.

On the base of developed procedure, it was analyzed the influence of the width of each ring element on the function of the disk radiation efficiency. For the disk radiator with 8 ring elements the dependence of the relative integral amplitude on the relative value of the change of each element is shown in Figure 4. It was determined, that the widths of the second and the last zones influence mostly on the efficiency. The change of these parameters in 15 % changes the integral amplitude approximately in 10 %.

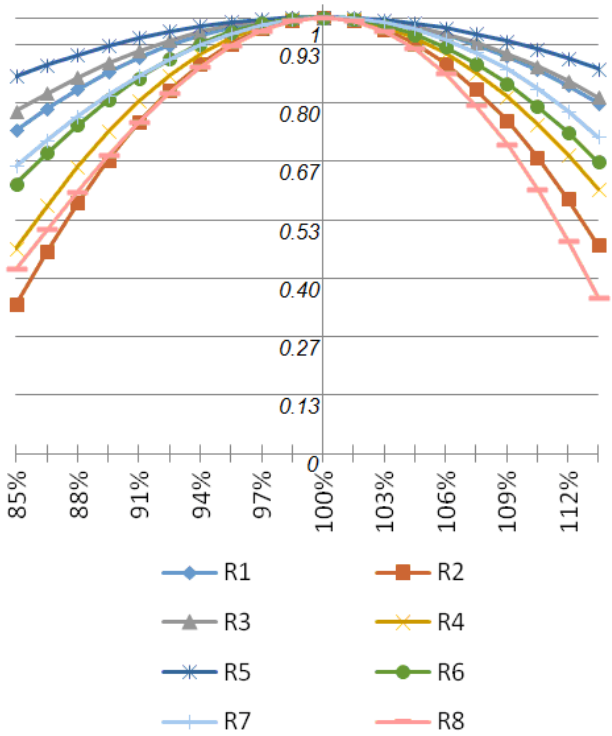


Figure 4. Dependence of radiation efficiency U on the relative deviation of the values of the sizes of the plate zones from the optimum ones

Further studies are aimed at the determination of the optimum ratio of the ring element thicknesses. The efficiency is evaluated by the expression:

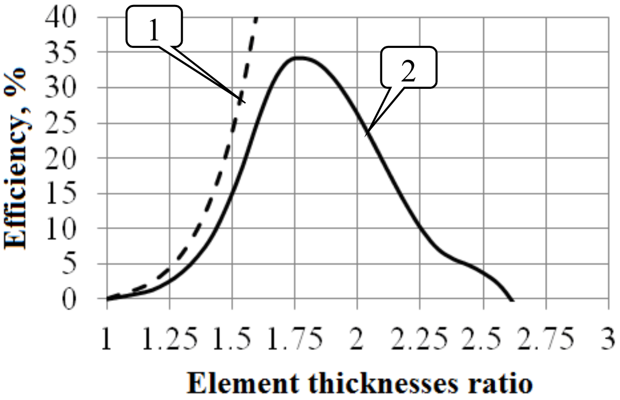
$$k_{eff} = \left(\frac{U_{disk}}{U_{plate}} - 1\right) \cdot 100\%, \qquad (18)$$

where k_{eff} is the relative radiation efficiency; U_{disk} is the integral vibration amplitude of the disk surface with stepped variable thickness; U_{plate} is the integral vibration amplitude of the flat disk surface.

The results of the calculations are shown in Figure 5 (1 – the result obtained by gradient descent method; 2 – the result obtained by finite element method).

By the calculation with the help of developed procedure (curve 1 in Figure 5) it was determined, that the more the ratio between the thicknesses of neighboring zones is, the higher the amplitude of the thin zones and the less the amplitude of the thick zones, at that the efficiency value rises unlimitedly.

However, previous experience of the production of the disk radiators showed that the design of the ring elements of considerable thickness led to the decrease of vibration amplitude of the radiator surface, the increase of loss and in some cases to the mechanical failure of the disk.



1 – result of calculation with the help of gradient descent method
2 – result of calculation with the help of finite-element method

Figure 5. Dependence of radiation efficiency of the disk radiator on the ratio of the thicknesses of neighboring ring elements

The reason of the difference in the theoretical and experimental results is the absence of consideration of vibration re-reflection from the boundaries of the radiator in the direction, which is perpendicular to the radiating surface of the disk.

At the enlargement of the thickness of the radiator zones the re-reflections essentially influence on the distribution of vibrations and consequently lead to additional reciprocal compensation of the vibrations causing the fall of the dependence with the rise of the element thickness ratio in Figure 5 after passing through maximum.

At that the width of the ring elements of the disk radiator was calculated by the procedure described previously. The results of the calculations are given in the form of the dependence 2 in Figure 5. From obtained dependence it is evident, that there is a maximum of the radiation efficiency achieved at the ratio of the thicknesses of "thin" and "thick" zones of 1.75. Given ratio keeps the same for all studied operating frequencies (from 20 kHz to 44 kHz).

4. DEVELOPED DISK RADIATORS

With the help of developed procedure, it was calculated 4 disk radiators of the different diameters and frequencies, which main characteristics are given in Table 2 and 3 [11–13].

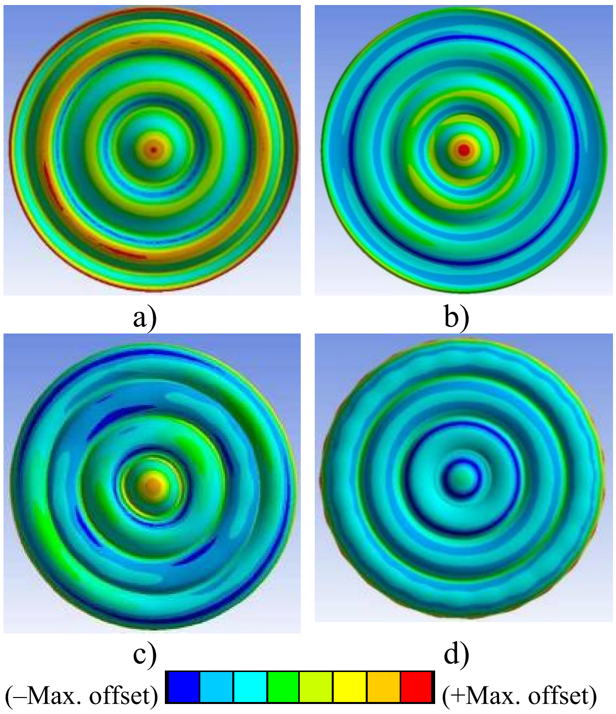
Table 2. Parameters of developed disk radiators

№	Radiator diameter (mm)	Frequency (Hz)	Number of vibration	Thickness of thin zone,	Thickness of thick zone,
1	250	21020	6	4	7
2	320	34372	8	6	11
3	360	27158	8	6	11
4	420	25273	8	7	13

Table 3. Width of the ring elements of developed radiators, mm

№	R_1	R_2	R_3	R_4	R_5	R_6	R_7	R_8
1	14.6	23.3	22.5	26.2	24.6	13.8		
2	13.5	21.4	20.5	23.3	21.4	24.3	23.2	12.4
3	15.2	24.1	23.1	26.3	24.2	27.4	26.3	13.4
4	17.7	27.9	26.8	30.4	28.0	31.7	30.5	17.0

The forms of vibrations (obtained with the use of the finite-element method) of developed disk radiators are shown in Figure 6.



a) – the diameter of 250 mm; b) – the diameter of 320 mm; c) – the diameter of 360 mm; d) – the diameter of 420 mm

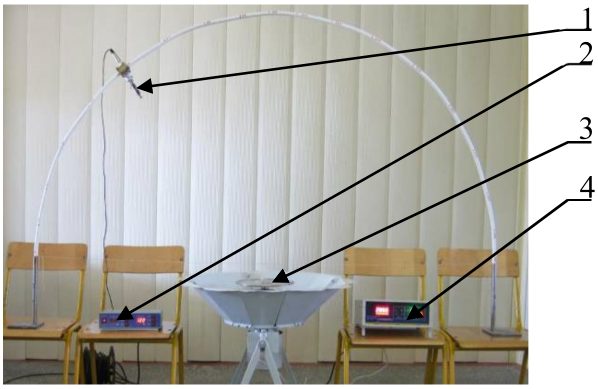
Figure 6. Distribution of vibration amplitude along the surface of the disk radiator

It is evident, that the distribution of the amplitudes along the surface of the disk radiator of stepped variable form has ringed character manifesting in alternation of minimums and maximums of vibrations. It proves the possibility of the practical application of developed calculation procedure of the disk radiators.

5. EXPERIMENTAL STUDY OF THE DIRECTION DIAGRAM OF THE DEVELOPED DISK RADIATORS

On the base of carried out calculations it was made of the titanium alloy the disk radiators with the diameters of from 250 mm to 420 mm [14–17].

To determine characteristics of the ultrasonic field generated by the developed radiators it was designed measuring test-bench [17] intended for the measurement of the acoustic pressure generated by the transducer 3 connected to ultrasonic generator 4 in arbitrary point in space at the distance of 1 m above the surface of the transducer (Figure 7).



1 – measuring microphone of the sound-level meter; 2 – sound-level meter; 3 – disk radiator; 4 – power generator for the disk radiator

Figure 7. Test-bench for the measurement of the acoustic pressure

The measurement of the acoustic pressure was carried out by the noise and vibration analyzer "Assistant" 2 (made by NTM-Security, Ltd., Russian Federation) with microphone 1. The results of the measurements for the developed radiators (at the distance of 1 m) of stepped variable form are shown in Figures 8–11.

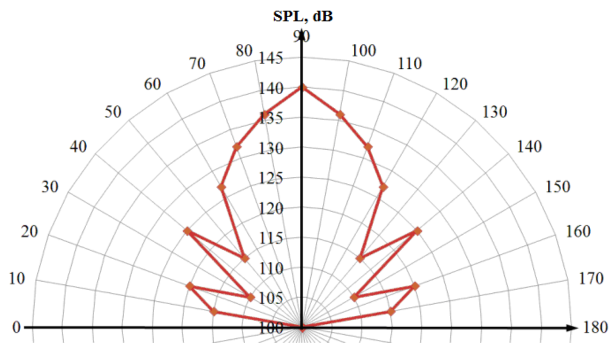


Figure 8. Direction diagram for the disk radiators of 250 mm diameter

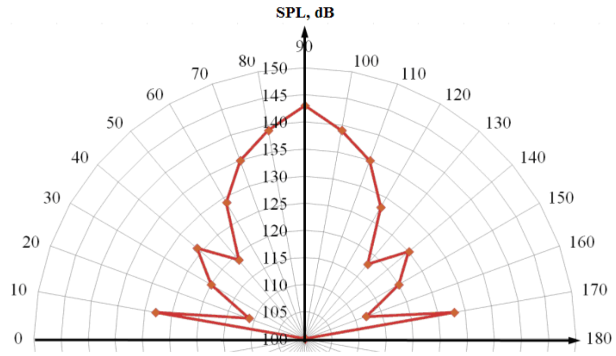


Figure 9. Direction diagram for the disk radiators of 320 mm diameter

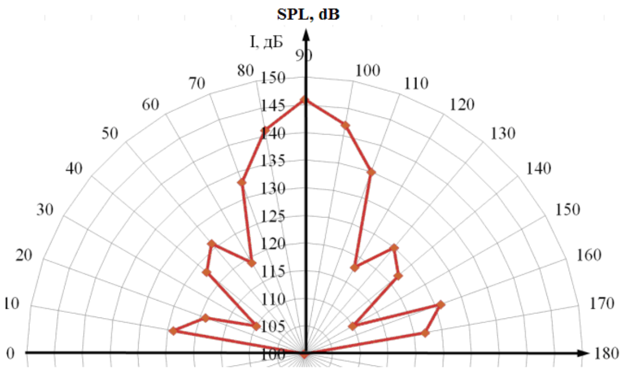


Figure 10. Direction diagram for the disk radiators of 360 mm diameter

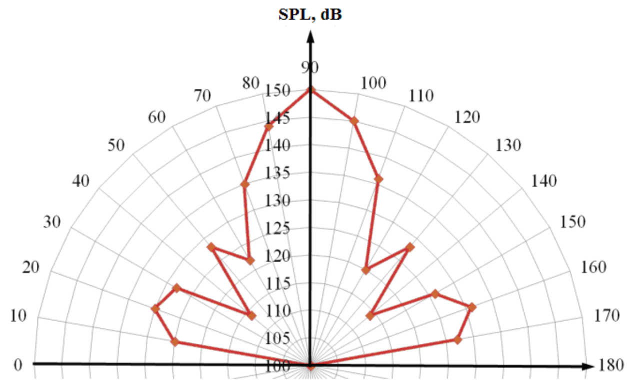


Figure 11. Direction diagram for the disk radiators of 420 mm diameter

The value of the acoustic pressure decreases monotonically with the rise of the distance from the flat of the radiating surface from 140–145 dB (at the distance of 0.5 m) up to 120–125 dB (at the distance of 4.5 m). The analysis of the direction diagram showed, about 95 % of acoustic energy was concentrated within the limits of the main lobe of the direction diagram, which aperture angle was from 30º to 60º depending on the disk diameter.

Thus, the intensification of the processes, for which it is required higher level of the acoustic pressure (no less than 135–140 dB) – defoaming, drying, is recommended to carry out in the near field of the disk radiator; the intensification of the processes, for which it is required lower level of the acoustic pressure (from 130 dB) – coagulation of smokes and aerosols, is advantageous to realize in the far field.

6. CONCLUSION

It is shown, that the main problem at the development of the radiators is the reciprocal compensation of vibrations generated by the zones of the radiators vibrating with the opposite phase. To solve the problem, it is proposed to decrease the vibration amplitude of the zones of the disk radiator with one phase and to increase the amplitude of the zones with the opposite phase due to the change of the thickness of corresponding zones.

For the determination of the optimum forms and sizes of the radiator it is developed the procedure of the definition of the geometric dimensions of the radiator zones vibrating with the opposite phase. With the use of developed procedure, it was calculated four radiators in the form of the disk of 250 mm to 420 mm diameter.

Carried out experimental studies allow determining the main characteristics of the disk radiators. It is found out, that in a near field generated designed ultrasonic radiators provide maximum level of the acoustic pressure of 145 dB, in a far field the level of the acoustic pressure is no less than 130 dB.

ACKNOLEDGEMENTS

This study is carried out under the financial support of the Grant of the President of Russian Federation No. MD-424.2018.8, Russian Foundation for Basic Research and Administration of Altai Region (Project No. 18-48-220006 r_a "Development of scientific foundations of ultrasonic technology of gaseous mediums cleaning from high-dispersed particles of natural and technogenic origin").

REFERENCES

[1] Babakov I.M., *Theory of vibrations. 4 edition*, cor., Moscow, Russian Federaton: Drofa, 2004.

[2] Corral G., Riera E., Gallego-Juarez J., Acosta Aparicio V., Pinto A., Martinez I., Blanco A., Experimental study of defoaming by air-borne power ultrasonic technology, *Physics Procedia*, 3, 2010, pp. 135-139.

[3] Dorovskikh R.S. et. al., Efficiency increase of wet gas cleaning from dispersed admixtures by the application of ultrasonic fields, *Archives of Acoustics*, 2016, Vol. 41, No. 4, pp. 757-771.

[4] Gallego-Juarez J.A., High-power ultrasonic processing: recent developments and prospective advances, *Physics Procedia*, Vol. 3, Issue 1, 2010, pp. 35–47.

[5] Gallego-Juarez J. A., Advances in the development of power ultrasonic technologies based on the stepped plate, *36th Annual Symposium*, Teddington, Middlesex, England, 2007.

[6] Gallego-Juárez J.A. Rodríguez G., Acosta-Aparicio V.M., Riera E., Cardoni A., Power ultrasonic transducers with vibrating plate radiators, *Power Ultrasonics*, *Woodhead Publishing, Cambridge*, 2014, pp. 159-194.

[7] Khmelev V.N. et. al., Study of the influence of secondary modes of vibrations on the uniformity of the distribution of working ring disk of ultrasonic disk, *18th International Conference of Young Specialists on Micro/Nanotechnologies and Electron Devices, EDM'2017*, 2017, pp. 290–293.

[8] Iliev I., Zhivomirov H., On the Spatial Characteristics of a Circular Piston, *Romanian Journal of Acoustics and Vibration*, Vol. 12, No. 1, 2015, pp. 29-34.

[9] Khmelev V.N. et. al., Study of the influence of the anisotropy of the mechanical properties of the material on the distribution of ultrasonic vibrations disk, *18th International Conference of Young Specialists on Micro/Nanotechnologies and Electron Devices, EDM'2017*, 2017, pp. 260–263.

[10] Khmelev V.N. et. al., Ultrasonic radiators for the action on gaseous media at high, *16th International Conference of Young Specialists on Micro/Nanotechnologies and Electron Devices EDM'2015*, 2015, pp. 224–228.

[11] Khmelev V.N. et. al., The measurements of acoustic power introduced into gas medium by the ultrasonic apparatuses with the disk–type, *17th International Conference of Young Specialists on Micro/Nanotechnologies and Electron Devices EDM 2016*, 2016, 251–256.

[12] Khmelev V.N., Shalunov A.V., Galakhov A.N., Romashkin A.A., *International Conference and Seminar on Micro/ Nanotechnologies and Electron Devices. EDM'2011: Conference Proceedings*, 2011, 230-235.

[13] Leissa Arthur W., *Vibration of plates*, NASA, 1969.

[14] Khmelev V.N. et. al., Efficiency increase of the dust-extraction plant by high-intensity ultrasonic vibrations, *16th International Conference of Young Specialists on Micro/Nanotechnologies and Electron Devices EDM'2015*, 2015. – P 213 – 217

[15] Sheng C., Shen X., Simulation of acoustic agglomeration processes of poly-disperse solid particles, *Journal of Aerosol Science and Techology*, 2007, Issue 41, 1–13.

[16] Patent US7156201B2, *Ultrasonic rod waveguide-radiator*.

[17] Khmelev V.N., Shalunov A.V., Tsyganok S.N., Golykh R.N., Shalunova K.V., Experimental Investigations of the Effectiveness of Acoustic Vibration Influence of Ultrasonic Frequency on Fogs, *Romanian Journal of Acoustics and Vibration*, Vol. 14, No. 2, 2017, pp. 69-74.