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SOUND POWER AND RADIATION EFFICIENCY OF A CIRCULAR PLATE

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This paper considers the problem of estimating the sound radiation by vibrating surfaces in the case of a circular plate clamped on the circumference. The aim of this paper is to verify the values of the equivalent surfaces of the plate, which were obtained theoretically, with the experimental values obtained in the free field and reverberant field conditions. The results obtained show that there is a good agreement of results for $k_{mn}a < 5$. For $k_{mn}a > 5$ an increase in the sound radiation of the plate can be observed, which results both from a complex character of the vibration of the plate and from the effect of the acoustic field on the vibration conditions of the plate.

1. Introduction

There are many methods of decreasing the noise and vibration level in the working environment. The most efficient method of noise control is decreasing the emission of sources. Decreasing or limiting the emission of sources is related to the development of effective methods of location of noise sources in machinery or devices. In the Institute of Mechanics and Vibroacoustics, Academy of Mining and Metallurgy, and in the Institute of Fundamental Technological Research Polish Academy of Sciences, of the research has long been done on the identification of the sources of vibroacoustic energy. This research is concerned with developing methods of identification of sound and vibration sources in machinery, in order to estimate the sound power radiated by the particular parts. This estimation permits the dominating sources and their frequency response to be determined. The early investigations were concerned with the use of a correlation method in the near field conditions to estimate the sound power of surface sources [5, 7].

The practical investigations were related to the estimation of the sound power of complex mechanical systems in the chamber airless shotblasting machines.

Another problem related to the methods of investigation of the emission of sources and to the assessment of the character of the emission is the investigation of relations between the sound power radiated and the structure-borne vibration.

From the practical point of view this problem is particularly important in the case of plates used as enclosures of machinery, which are often themselves the sources of noise. In the real conditions, the plate elements have a complex boundary condition as well as a complex shape with holes, ribbing or other irregularities which make their mathematical description impossible. However, in all the cases the resonance vibrations of these plates have a dominant significance. The classical theory of plate vibration permits the determination of the resonance vibration frequency for different boundary conditions. It is, however, much more difficult to estimate the sound power emitted or the radiation efficiency of the vibrating plates. These quantities are usually determined experimentally.

There are a number of methods of measurement of the sound power of plates, of which the most significance has been recently gained by the two-microphone method based on the determination of the imaginary part of the mutual spectral power density of the sound pressures in the near field conditions. This method is very precise but requires, however, complex and expensive instrumentation.

The need, therefore, arises for a method which can preliminarily give an approximate estimation of the sound power of a plate by means of simple measurements. This estimation is possible if a piston model of a plate is assumed, for which the values of the equivalent surface of the plate were previously determined. The values of the amplitude of the vibration velocity should, however, be determined from measurement at a chosen point of the plate.

Thus on the basis of simple measurements of the plate vibration, it could be possible to estimate an approximate characteristic of its sound power as a function of vibration frequency. The method of equivalent surface determination was given by MORSE in [11] and concerned the first vibration mode of a circular membrane.

The aim of this paper is to verify through measurements the relation mentioned above and to extend it to include the higher vibration modes. Since calculations of the sound power radiated by plates of complex shape and different clamping are rather difficult, this paper is restricted to the analysis of a circular plate clamped on the circumference. Using the classical fundamentals of plate vibration theory, calculations can be made in this case, and verified experimentally later on. This approach can contribute to deeper knowledge of plate radiation over a wide frequency range, which can provide the basis for expanding the investigations to include the more complex systems.

2. Theoretical approach

It is known that flexural vibrations of plates are characterized by resonance frequencies, the so-called vibration modes, which result from the solution of the differential equation

$$\nabla^4 \xi + \frac{12\varrho(1-\nu^2)}{Eh^2}\frac{\partial^2 \xi}{\partial t^2} = 0. \tag{1}$$

where ξ is the displacement of the vibrations of the plate, ϱ is the density of the plate material, ν is the Poisson ratio, E is the modulus of the longitudinal elasticity, and h is the plate thickness. For a circular plate excited uniformly over the whole surface clamped on the circumference, i.e. satisfying the boundary conditions $\xi = 0$ and $\partial\xi/\partial r = 0$ for $r = a$ (where r is the current radius of the plate and a is the value of the radius on the circumference of the plate), the characteristic equation can be given as

$$I_m(\gamma_a)\left(\frac{d}{dr}\right)J_m(\gamma_r) - J_m(\gamma_a)\frac{d}{dr}I_m(\gamma_r) = 0. \tag{2}$$

Solution of this equation gives a series of the values γ_{mn}, which define the eigenfrequencies f_{mn}

$$f_{mn} = \frac{\pi h}{4a^2}\sqrt{\frac{E}{3\varrho(1-\nu^2)}}\,(\beta_{mn})^2, \quad \beta_{mn} = \gamma_{mn}\frac{a}{\pi}, \tag{3}$$

where the indices m and n denote the order of the axially nonsymmetrical and axially symmetrical modes, respectively.

Since the sound power of vibrating plates is determined mainly by the resonance vibrations, the further considerations will concern the sound power measurements for the particular vibration modes dependent on the product $k_{mn}a$, where $k_{mn} = 2\pi f_{mn}/c$ is the wave number, while c is the wave propagation velocity in the air.

One of the methods of calculating the sound power of circular plates for the particular vibration modes is the method based on the measurement of the vibration velocity v on the surface of the plate.

Considering that for higher vibration modes a complex vibration distribution occurs on the surface of the plate, it is necessary to determine the mean value of the velocity

$$\bar{v}^2 = \frac{1}{S_r} \iint_{S_r} v^2(r, \varphi)\, dr\, d\varphi, \tag{4}$$

where S_r is the surface of the plate.

The sound power radiated by the plate is thus

$$W_{\bar{v}} = \bar{v}^2 R_p S_r, \tag{5}$$

where R_p is the radiation resistance of the plate, which for a circular plate in an infinite baffle is

$$R_p = \varrho_0 c \left[1 - \frac{J_1(2k_{mn} a)}{k_{mn} a} \right], \tag{6}$$

where $J_1(k_{mn} a)$ is a Bessel function of the first kind of the first order, and ϱ_0 is the air density.

The sound power of a circular plate can also be obtained on the basis of the determination of its vibration velocity at a given point and of the equivalent surface S_{eq} so that the sound power thus determined is equal to the power expressed by formula (5).

For axially symmetrical vibrations, the best point for which the vibration velocity of the plate can be determined is its centre. Designating as v_0 the *r.m.s.* vibration velocity of the plate centre, the following expression can be written for the sound power radiated by the plate

$$W_v = v_0^2 R_p S_{eq}. \tag{7}$$

The ratio of the equivalent surface of the plate, S_{eq} and its real surface S_r can be called the coefficient of the vibration distribution of the plate

$$\varkappa = \frac{S_{eq}}{S_r}. \tag{8}$$

Hence, considering realtions (5) and (6), the expression for the equivalent surface of the plate takes the form

$$S_{eq} = \frac{W_v}{v_0^2 \left[1 - \dfrac{J_1(2k_{mn} a)}{k_{mn} a} \right]}. \tag{9}$$

The equivalent surface can also be estimated theoretically from the relation

$$S_{eq} = 2\pi \int_0^a R(r)\, r\, dr, \tag{10}$$

where the function $R(r)$ can be defined from the solution of equation (1).

In the case of the flexural vibrations of a thin circular plate clamped on the circumference of the radius $r = a$, the function $R(r)$ takes the form

$$R(r) = CJ_m(\gamma r) + DI_m(\varrho r), \tag{11}$$

where C and D are constant, and the quantity γ is defined by the solutions of equation (2).

Insertion of equation (11) into relation (10) gives after integration the following expression for the equivalent surface of the plate

$$S_{eq} = \frac{2\pi a}{\sqrt{k_{mn}}\,cD}\left[J_1(B\sqrt{k_{mn}}\,ca) - \frac{J(B\sqrt{k_{mn}}\,ca)}{I_0(B\sqrt{k_{mn}}\,ca)}\,I_1(B\sqrt{k_{mn}}\,ca)\right], \tag{12}$$

where

$$B = \sqrt[4]{\frac{12\varrho}{Eh^2}(1-\nu^2)}. \tag{13}$$

The equivalent surface can also be determined experimentally by the measurements of the vibration velocity at the centre of the plate (for the axially symmetrical modes) and by the determination of the sound power by one of the known methods.

3. Experimental set-up and procedure

An experimental set-up was constructed for the investigation of the radiation of acoustic energy of thin circular plates (Fig. 1). It included two steel rings which permitted the plate tested to be clamped. The lower ring *1* was clamped to the base *2*, while the upper ring *3* was axially pressed down by a system of six pneumatic servos *4*. The vibration of the plate *5* was forced by the piston excited, in turn, by the head (BK 4813) of the exciter *6* (BK 4801). The plate vibration was forced by a vibrating air layer between the plate and the circular piston. As a result of this, according to the theoretical assumptions, the whole plate was excited to uniform vibration.

In order to assure the conditions of a plate vibrating in an infinite baffle, the system included an acoustic baffle and the encasing of the lower part containing the exciter.

The plate tested was clamped in annular jaws which pressed it down uniformly over the circumference. The mean value of the unitary pressure on the circumference of the plate was 1570 Nm^{-1}. The value of the mean unitary pressure on the circumference of the plate could be adjusted by changing the value of the air pressure in the adjusting chamber which fed the system of 6 servos.

The exciting system used (Fig. 2) permitted the flexural vibrations of the plate to be forced. The exciting source was an electrodynamic vibration exciter fed with a sinusoidal signal from a generator over the frequency range 20-1200 Hz.

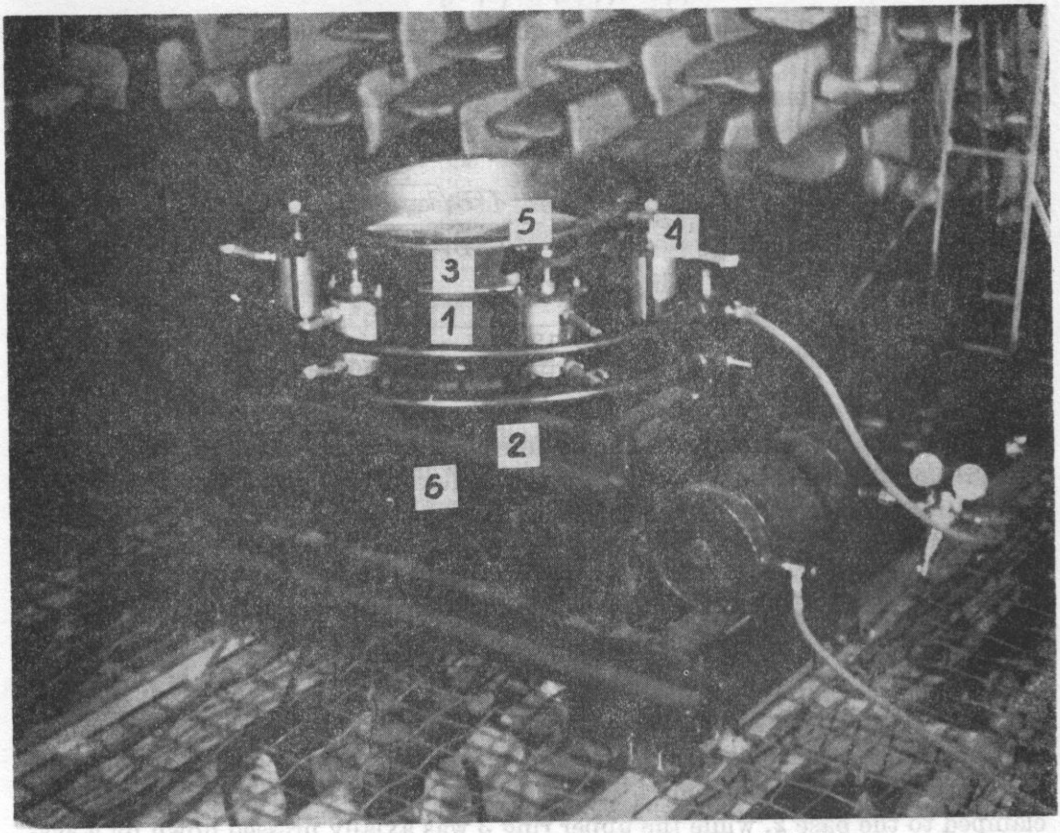


Fig. 1. The system for investigating vibroacoustical plates
1 – the lower clamping ring, *2* – the base, *3* – the upper clamping ring, *4* – a system of clamping servos, *5* – the tested plate, *6* – the head of a vibration exciter

The experimental determination of the equivalent surface of the plate required the assessment of the sound power radiated by the plate and the amplitude of the vibration velocity in the anti-node of the plate vibrations. Since the basic aim of these investigations was the essessment of radiation at the eigenfrequencies of the plate, it was necessary to determine preliminarily these frequencies by defining the vibration modes. The modes were visualized by the method of Chladni figures, consisting in covering the vibrating plate surface with a fine material layer [8]. As a result of the plate vibrations, this material gathered in the vibration nodes. The radiation of acoustic energy by the plate was investigated in the free field conditions (in an anechoic chamber) and in the reverberant field conditions (in a reverberation chamber).

The procedure consisted of three main stages. The first stage involved preliminary investigation aimed at the determination of the frequencies of the vibration nodes and their comparison with the calculated values. These investigations

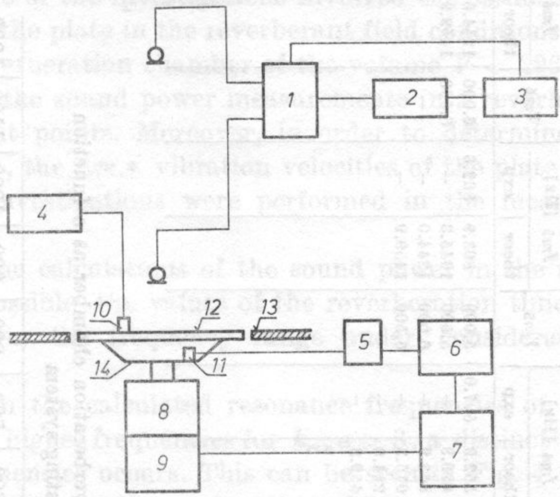


Fig. 2. The block diagram of the system for measuring and exciting vibration of the plate, which was used in the investigations in the free and reverberant field conditions

1 – a *BK* 4408 two-channel microphone selector, 2 – a *BK* 2607 measuring amplifier, 3 – a *BK* 2305 recorder, 4 – a *BK* 2607 measuring amplifier, 5 – a *BK* 2650 precision preamplifier, 6 – a *BK* 1026 generator, 7 – a *BK* 2707 power amplifier, 8 – the head of a *BK* 4813 exciter, 9 – the encasing of a *BK* 4801 exciter, 10 – an *HM* 0002 contactless electromagnetic vibration velocity sensor, 11 – a *BK* 8305 accelerometer, 12 – the plate tested, 13 – the baffle, 14 – the exciting piston

were performed in an anechoic chamber. In order to assure regular vibration distributions on the plate, plates were chosen from the point of view of good isotropic properties. As a result of this selection, a steel isotropic plate of the thickness $b = 0.9$ mm and the diameter $2a = 497$ mm was used in the further investigations. Table 1 shows the values of the eigenfrequencies of the plate, both obtained experimentally and from theoretical calculations according to relation (3). The following numerical values were taken for the calculations: $E = 2.06 \cdot 10^{11}\ \mathrm{N \cdot m^{-2}}$, $\nu = 0.27$, $\varrho = 7.86 \cdot 10^3\ \mathrm{kg \cdot m^{-3}}$. The values of the serfies γ_{mn} were taken after MORSE [11].

The second stage of the investigations involved the estimation of the sound power radiated by the plate at the resonance frequencies. These investigations were performed in the free field conditions, where the plate was excited to vibrate at the frequencies of axially symmetrical and axially nonsymmetrical resonances. The examples of Chladni figures for chosen eigenmodes are shown in Fig. 3.

The sound power was determined on the basis of the directional characteristic by measuring the sound pressure level as a function of angle for a constant distance from the plate centre. This distance was 2 m. In order to determine experimentally the equivalent surface of the plate for the axially symmetrical

Table 1. The calculated and experimentally determined values of the eigenfrequencies of the plate

m	γ_{m1}	f_{m1} [Hz]		γ_{m2}	f_{m2} [Hz]		γ_{m3}	f_{m3} [Hz]		γ_{m4}	f_{m4} [Hz]		γ_{m5}	f_{m5} [Hz]		γ_{m6}	f_{m6} [Hz]	
		theor	exp		theor	exp		theor	exp		theor	exp		theor	exp		theor	exp
0	1.015	35.6	28.1	2.007	139.2	123.2	3.000	311.0	286.3	4.000	552.9	507.0	5.000	863.9	862.6	6.000	1244.0	—
1	1.468	74.5	—	2.483	213.0	—	3.490	420.9	409.0	4.500	699.8	—	5.500	1045.3	—	6.500	1459.9	—
2	1.879	122.0	—	2.992	309.3	—	4.000	552.9	520.7	5.000	863.9	—	6.000	1244.0	—	—		
3	2.290	181.2	155.5	3.501	423.5	—	4.500	699.8	—	5.500	1045.3	—	6.500	1459.9	—			
4	2.700	251.9	—	4.010	555.7	—	5.000	863.9	—	6.000	1244.0	—						
5	3.110	334.2	295.0	4.519	705.7	—	5.500	1045.3	—	6.500	1459.9	—						
6	3.523	428.9	—	5.028	873.6	—	6.000	1244.0	—									
7	3.934	534.8	—	5.537	1059.4	—	6.500	1459.9	—									
8	4.345	652.4	—	6.046	1263.1	—												
9	4.756	781.6	—	6.5	1459.9	—												
10	5.167	922.6	—															
11	5.578	1075.2	—															
12	6.0	1244.0	—															
13	6.5	1459.9	—															

Table 2. The values of the reverberation time of the reverberation chamber as a function of frequency after mounting the testing system

f [Hz]	100	200	250	300	350	400	450	500	600	700	800	900	1000	1100	1200
T [s]	6.6	7.9	7.0	7.3	6.7	6.1	6.2	6.8	6.0	5.6	5.1	5.2	5.0	4.6	4.3

modes, the *r.m.s.* vibration velocity of the plate was also measured at its centre. The directional characteristics were measured using a BK 3922 turn table and registered with a BK 2305 level recorder.

The third stage of the investigations involved the estimation of the sound power radiated by the plate in the reverberant field conditions. This estimation took place in a reverberation chamber of the volume $V = 196\ \mathrm{m}^3$. The estimation was based on the sound power measurements in a reverberation chamber at six measurement points. Moreover, in order to determine the equivalent surface of the plate, the *r.m.s.* vibration velocities of the plate centre were also measured. These investigations were performed in the measurement system shown in Fig. 1.

In order for the calculations of the sound power in the reverberant field conditions to be possible, the values of the reverberation time of the chamber were measured over the frequency range under consideration. These are given in Table 2.

It follows from the calculated resonance frequencies of the plate shown in Table 1, that for higher frequencies for $k_{mn}a > 8$, a distinct concentration of the resonance frequencies occurs. This can be seen in Fig. 4 which shows the number of vibration modes N as a function of frequency, corresponding to the frequency range 20 Hz. The vibration distribution is much more complex then, as a result of which, the vibration velocity measurements at the plate centre may be insufficient. The measurements were expanded, therefore, to include the vibration velocities at several points of the plate in order to provide the mear value.

After the preliminary investigations, the measurements were decided to be taken at 5 points, one of which was at the centre of the plate, while the other four were on the circumference at the distance $r = 80$ mm from the centre of the plate.

From these measurements the mean square value of the vibration velocity was calculated from the relation

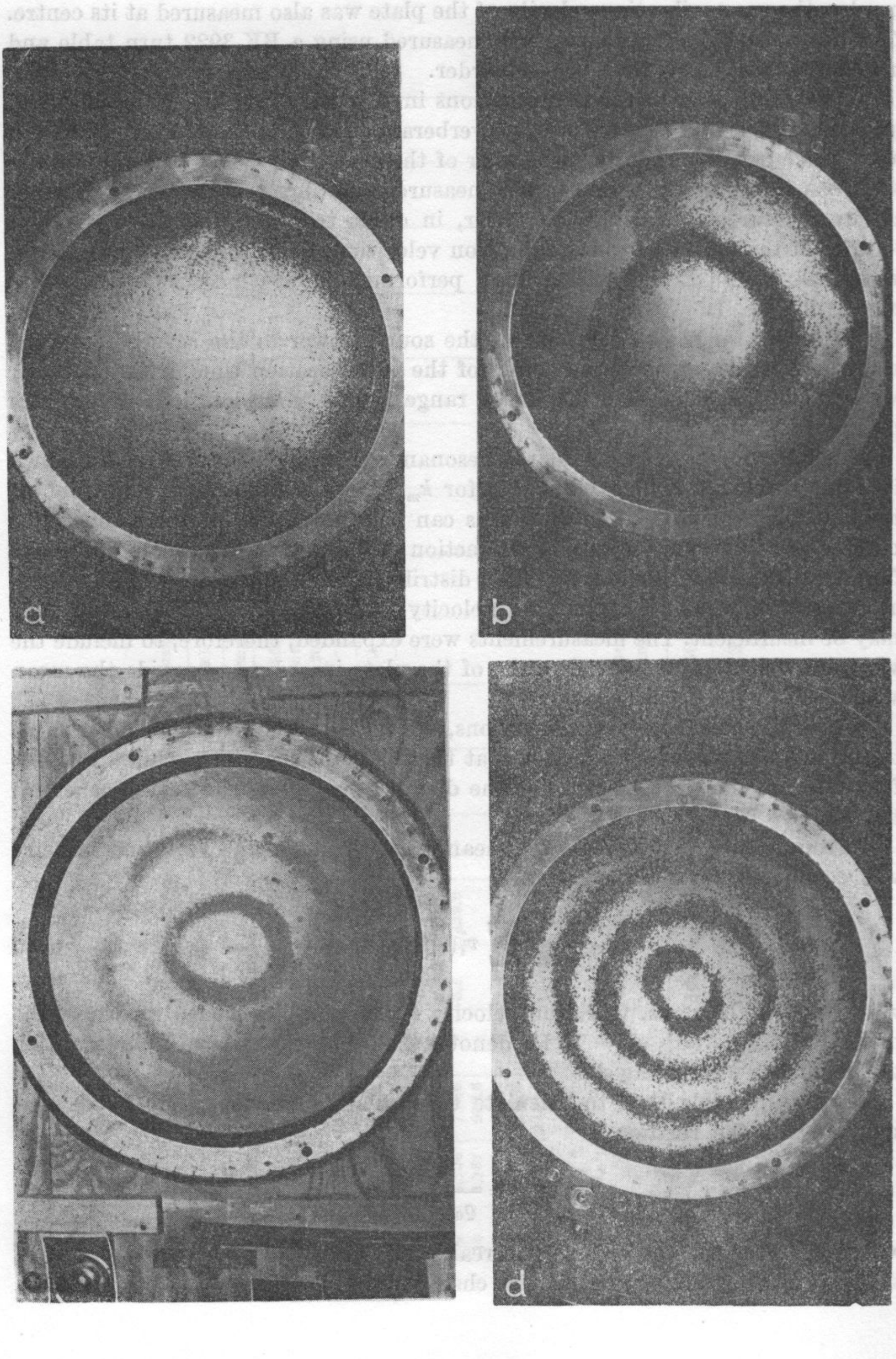
$$\bar{v}^2_{(\Delta f)} = \frac{1}{n}\sum_{i=1}^{n} v_i^2(\Delta f), \qquad n = 5, \tag{14}$$

where $v_{i(\Delta f)}$ is the *r.m.s.* vibration velocity of the plate at the ith measurement point ($i = 1, 2, \ldots, 5$), $\Delta f = 10$ Hz denotes the frequency bandwidth over which the amplitude was averaged.

It was possible thus to calculate the radiation efficiency [1, 2], defined according to the relation

$$\sigma_{(\Delta f)} = \frac{W_{T(\Delta f)}}{\varrho_0 c \bar{v}^2_{(\Delta f)} S_r}, \tag{15}$$

where S_r is the surface area of the real plate, and W_T is the sound power of the plate measured in a reverberation chamber.


a
b
d

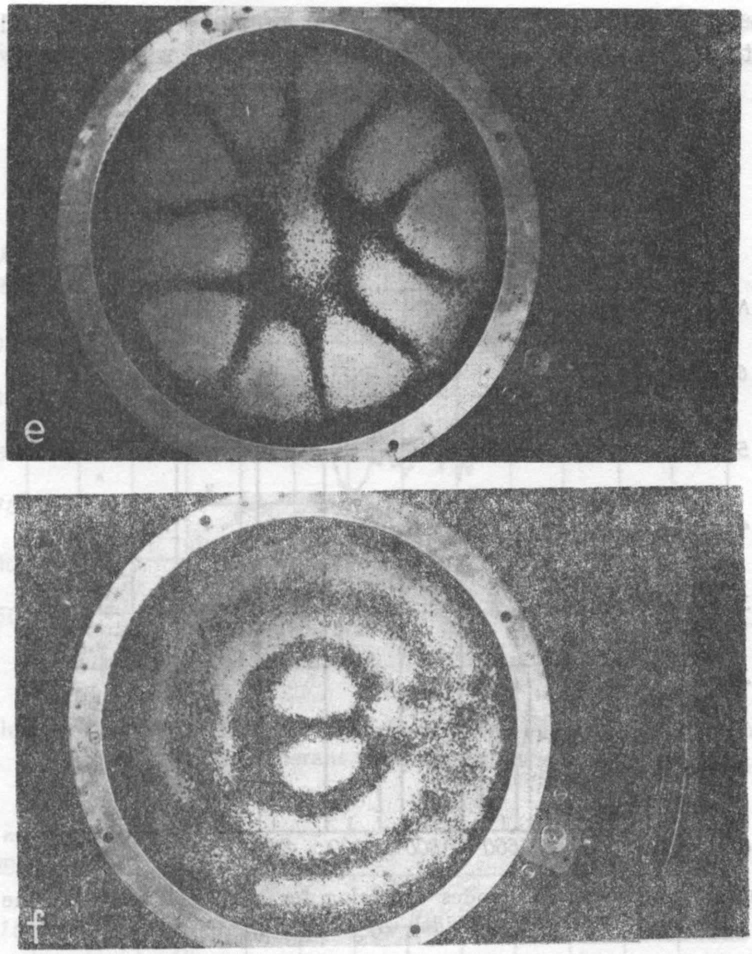


Fig. 3. Examples of the Chladnì figures obtained for several vibration modes of a circular plate clamped on the circumference (with diameter $2a = 497$ m, thickness $b = 0.9$ mm)

a) $f_{01} = 28.1$ Hz; b) $f_{02} = 123.0$ Hz; c) $f_{03} = 286.3$ Hz; d) $f_{04} = 507.9$ Hz; e) $f_{51} = 295.0$ Hz, f) $f_{13} = 499.0$ Hz

4. Estimation of the radiated sound power and the equivalent surface of the plate

a. The measured and calculated results obtained from the investigations performed in the free field conditions

On the basis of the directional characteristics determined, the squared values of the sound pressure were calculated at a given frequency. From the measurement of the distribution of the sound pressures measured on a hemisphere at the distance $r_m = 2$ m from the centre of the plate, 38 values of the sound pressures p_i were determined for averaging the pressure at each of the frequencies under investigation. Hence, the mean sound pressure levels were determined according to the relation

$$L_{\bar{p}} = 10\log \bar{p}^2/p_0^2,$$

where

$$\bar{p}^2 = \frac{1}{38}\sum_{i=1}^{38} p_i^2, \quad p_0^2 = 4\cdot 10^{-10}\,\frac{\mathrm{N}^2}{\mathrm{m}^4}. \tag{16}$$

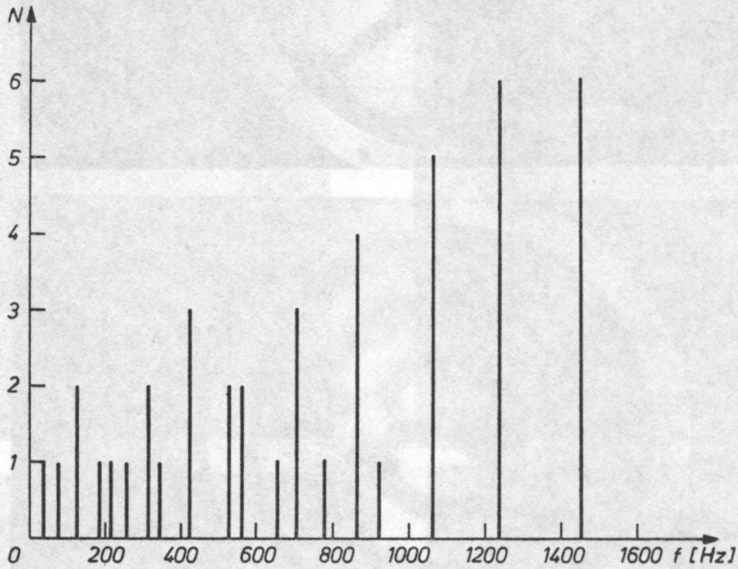


Fig. 4. The number of vibration modes occurring for a circular plate in the frequency bandwidth $\Delta f = 20$ Hz

The sound power radiated was calculated from the relation

$$W_{\bar{p}} = \frac{\bar{p}^2}{\varrho_0 c} S_p, \tag{17}$$

where $\varrho_0 c = 415\ \mathrm{kg\cdot m^{-2}\cdot s^{-1}}$ is the characteristic impedance of the air, and $S_p = 2\pi r_m^2 = 25.1\ \mathrm{m}^2$ is the area of the hemisphere for the measurements of sound pressure. Hence, the sound power level was expressed in the form

$$L_W = 10\log\frac{W_{\bar{p}}}{W_0} \quad [\mathrm{dB}], \tag{18}$$

where $W_0 = 10^{-12}$ W.

The values of the average sound pressure levels and of the sound power are shown in Figs. 5 and 6.

On the basis of the measured *r.m.s.* vibration velocities at the centre of the plate, the values of their levels were calculated, according to the relation

$$L_{v0} = 10\log\frac{v_0^2}{v_{\text{ref}}^2}, \tag{19}$$

where $v_{\text{ref}} = 5\cdot10^{-8}\ \text{m}\cdot\text{s}^{-1}$. These values are given in Fig. 7.

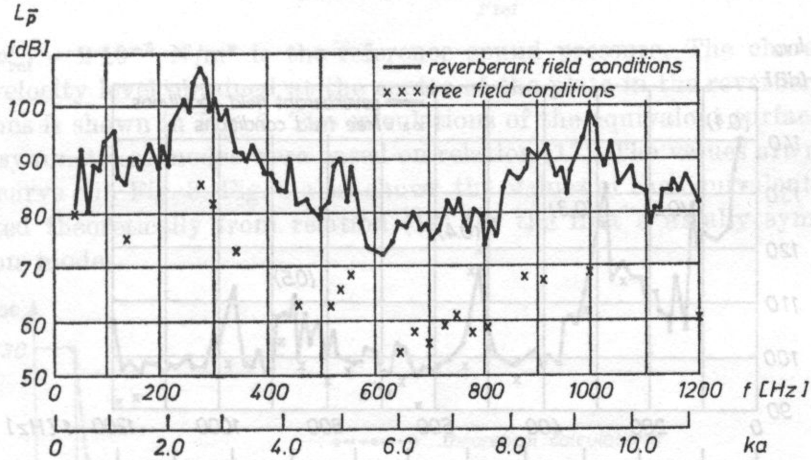


Fig. 5. The levels of the averaged sound pressure radiated by the plate in the free field and reverberant field conditions

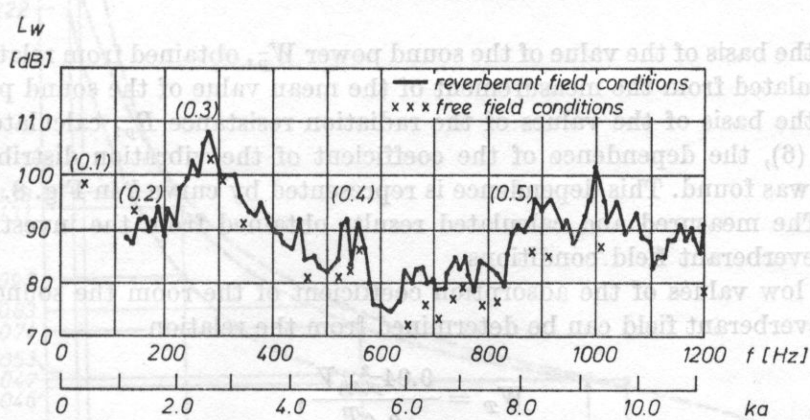


Fig. 6. The levels of the measured sound power radiated by the plate in the free field and reverberant field conditions

Relation (7) in the following form was used for the calculation of the equivalent surfaces

$$S_{\text{eq}} = \frac{W_{\bar{p}}}{R_p v_0^2}. \tag{20}$$

After division of the value obtained for the plate surface S_{eq} by its real surface S_r, the coefficient of the vibration distribution, defined by formula (8), was calculated,

$$\varkappa = \frac{W_{\bar{p}}}{R_p S_r v_0^2}, \tag{21}$$

where $S_r = 0.25\ \text{m}^2$.

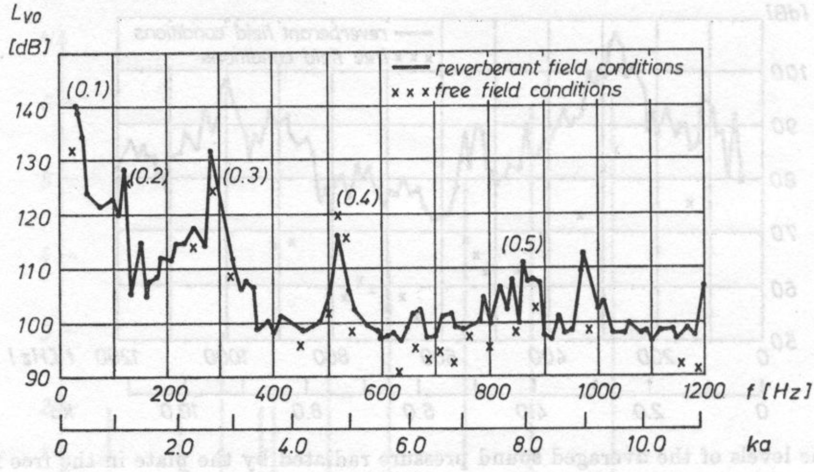


Fig. 7. The levels of the vibration velocity at the centre of the plate tested, measured in the free field and reverberant field conditions

On the basis of the value of the sound power $W_{\bar{p}}$, obtained from relation (17) and calculated from the measurement of the mean value of the sound pressure, and on the basis of the values of the radiation resistance R_p, calculated from formula (6), the dependence of the coefficient of the vibration distribution $\varkappa$ on $k_{mn}a$ was found. This dependence is represented by curve b in Fig. 8.

b. The measured and calculated results obtained from the investigations in the reverberant field conditions

For low values of the adsorption coefficient of the room the sound power in the reverberant field can be determined from the relation

$$W_T = \frac{0.04\bar{p}^2_{(\Delta f)} V}{\varrho_0 c T}, \tag{22}$$

where T is the reverberation time of the room, V is its volume, and $\bar{p}^2_{(\Delta f)}$ is the mean square value of the sound pressure, determined by averaging the results obtained at 6 positions of the microphone at the frequency band $\Delta f = 10$ Hz:

$$\bar{p}^2_{(\Delta f)} = \frac{1}{n} \sum_{i=1}^{n} p^2_{i(\Delta f)}, \quad n = 6. \tag{23}$$

Then, the sound pressure level in the reverberant field can be calculated from the relation

$$L_{WT(\Delta f)} = L_{\bar{p}(\Delta f)} - 10\log T + 10\log V \quad 14 \quad [\mathrm{dB}], \tag{24}$$

where

$$L_{\bar{p}(\Delta f)} = 10\log \frac{\bar{p}^2_{(\Delta f)}}{p^2_{\mathrm{ref}}}, \tag{25}$$

where $p^2_{\mathrm{ref}} = 2\cdot 10^{-5}$ N/m² is the reference sound pressure. The characteristic of the velocity level obtained at the centre of the plate in the reverberant field conditions is shown in Fig. 7. The calculations of the equivalent surface for the axially symmetrical modes were based on relation (17). The values are represented by curve *c* in Fig. 8. Fig. 8 also shows the values of the equivalent surface, calculated theoretically from relation (12) for the first 5 axially symmetrical vibration modes.

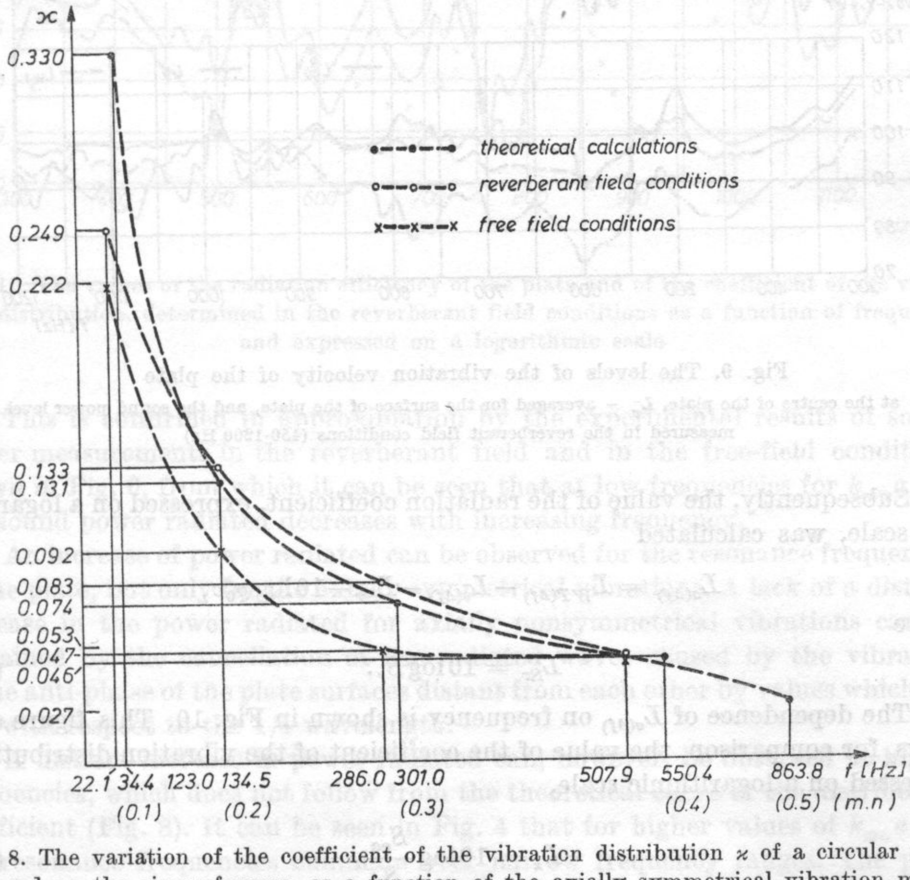


Fig. 8. The variation of the coefficient of the vibration distribution ϰ of a circular plate clamped on the circumference, as a function of the axially symmetrical vibration modes
a) calculated from formulae (8) and (13), b) measured in the reverberant field conditions, c) measured in the free field conditions

5. Measured and calculated values of the radiation efficiency of the plate

In order to verify the usefulness of the equivalent surface for the higher vibration modes, for which the vibration distribution of the plate is more complex, the radiation efficiency, defined by formula (15), was measured. The mean value of the vibration velocity was calculated from formula (14).

Subsequently, the levels of the mean velocity

$$L_{\bar{v}(\Delta f)} = 10\log\frac{\bar{v}^2_{(\Delta f)}}{v_{\text{ref}}}, \tag{26}$$

were measured, whose dependence on frequency is shown in Fig. 9. This figure also shows the vibration velocity level at the centre of the plate $L_{v0(\Delta f)}$ and the sound power level $L_{WT(\Delta f)}$ determined from measurements in the reverberant field, according to formula (24).

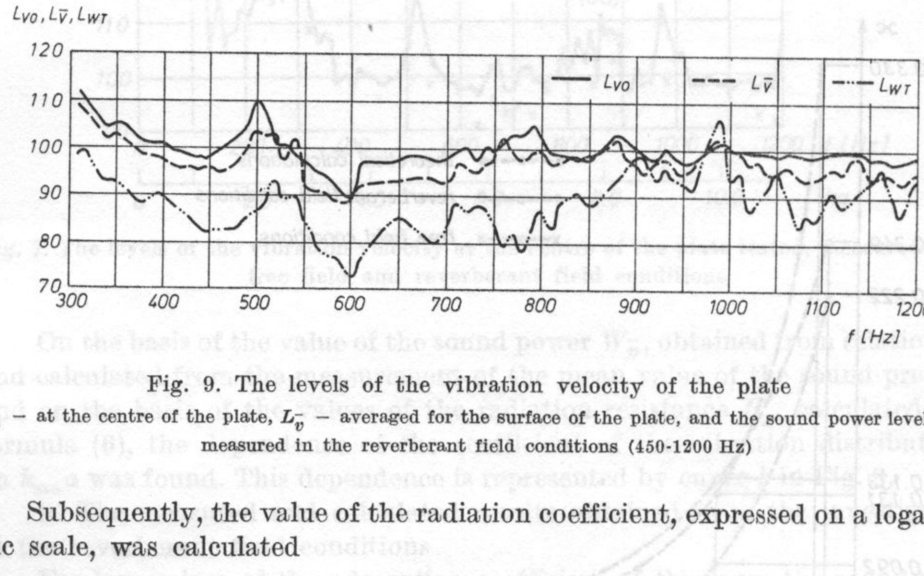


Fig. 9. The levels of the vibration velocity of the plate
L_{v0} – at the centre of the plate, $L_{\bar{v}}$ – averaged for the surface of the plate, and the sound power level L_{WT} measured in the reverberant field conditions (450-1200 Hz)

Subsequently, the value of the radiation coefficient, expressed on a logarithmic scale, was calculated

$$L_{\sigma(\Delta f)} = L_{WT(\Delta f)} - L_{\bar{v}(\Delta f)} - L_{Sr} - 10\log \varrho_0 c, \tag{27}$$

where

$$L_{Sr} = 10\log S_r. \tag{28}$$

The dependence of $L_{\sigma(\Delta f)}$ on frequency is shown in Fig. 10. This figure also shows, for comparison, the value of the coefficient of the vibration distribution, expressed on a logarithmic scale,

$$L_{\varkappa} = 10\log\frac{S_{\text{eq}}}{S_r}, \tag{29}$$

where S_{eq} is the equivalent surface of the plate, determined from relation (20).

6. Discussion of the measured and calculated results

It follows from theoretical analysis that there are two opposite tendencies affecting the plate radiation. On the one hand, the coefficient of the vibration distribution, $\varkappa$, decreases with increasing frequency, which results from Fig. 8; on the other, the radiation resistance increases as the frequency increases, and for $k_{mn} a > 2$ it is equal to $\varrho_0 c$.

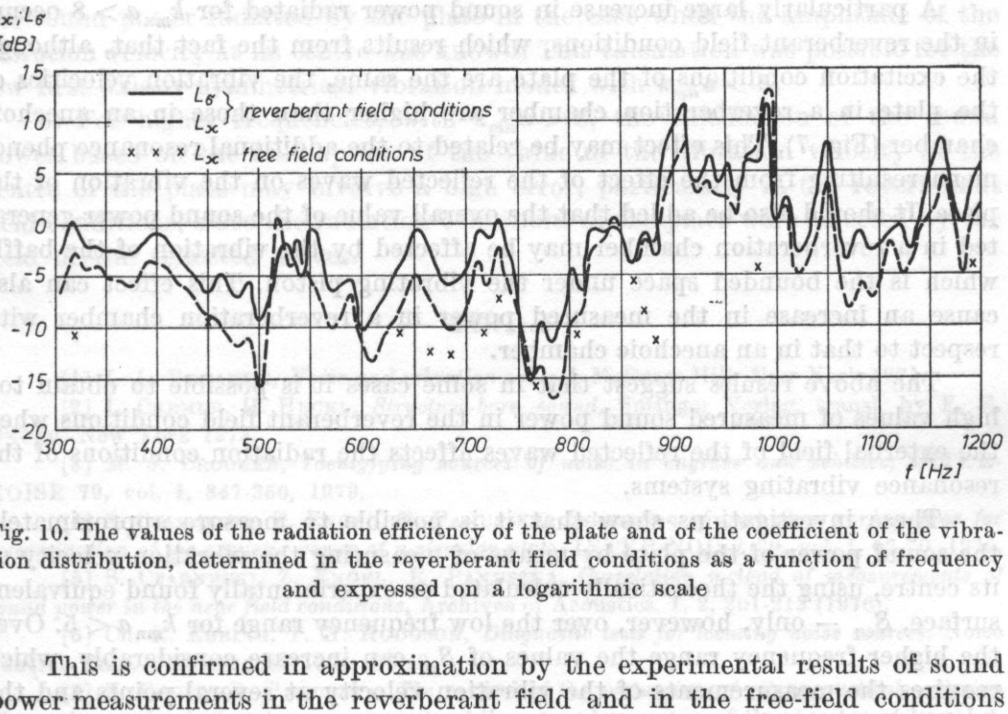


Fig. 10. The values of the radiation efficiency of the plate and of the coefficient of the vibration distribution, determined in the reverberant field conditions as a function of frequency and expressed on a logarithmic scale

This is confirmed in approximation by the experimental results of sound power measurements in the reverberant field and in the free-field conditions shown in Fig. 6, from which it can be seen that at low frequencies for $k_{mn} a < 5$ the sound power radiated decreases with increasing frequency.

An increase of power radiated can be observed for the resonance frequencies of the plate, but only for the axially symmetrical vibrations. A lack of a distinct increase in the power radiated for axially nonsymmetrical vibrations can be explained by the cancellation of the radiated waves caused by the vibration in the anti-phase of the plate surfaces distant from each other by values which are low with respect to the 1/4 wavelength.

A distinct increase in power radiated can, however, be observed at higher frequencies, which does not follow from the theoretical curve of the distribution coefficient (Fig. 8). It can be seen in Fig. 4 that for higher values of $k_{mn} a > 8$ the resonance frequencies cumulate over narrow frequency ranges. The plate vibrates in this case in several modes, which increases considerably its equivalent surface, since the distances between the vibration anti-nodes of the particular parts of the plate are comparable to 1/4 wavelength, which decreases the cancellation effect mentioned above.

This is confirmed by the results shown in Fig. 10, which give large values of the coefficient $\varkappa$, expressed on a logarithmic scale for some frequencies over the range 900-1200 Hz. Similar results are obtained for the radiation coefficient $\varkappa$, expressed on a logarithmic scale, which is obtained from the measurements of the mean vibration velocity of the plate by averaging the results from several points.

A particularly large increase in sound power radiated for $k_{mn}a > 8$ occurs in the reverberant field conditions, which results from the fact that, although the excitation conditions of the plate are the same, the vibration velocities of the plate in a reverberation chamber are higher than those in an anechoic chamber (Fig. 7). This effect may be related to the additional resonance phenomena resulting from the effect of the reflected waves on the vibration of the plate. It should also be added that the overall value of the sound power generated in a reverberation chamber may be affected by the vibration of the baffle which is the bounded space under the vibrating piston. This effect can also cause an increase in the measured power in a reverberation chamber with respect to that in an anechoic chamber.

The above results suggest that in some cases it is possible to obtain too high values of measured sound power in the reverberant field conditions when the external field of the reflected waves affects the radiation conditions of the resonance vibrating systems.

These investigations show that it is possible to measure approximately the sound power of the plate by means of measuring the vibration velocity at its centre, using the theoretically calculated or experimentally found equivalent surface, S_{eq} – only, however, over the low frequency range for $k_{mn}a < 5$. Over the higher frequency range the values of S_{eq} can increase considerably, which requires the measurements of the vibration velocity at several points and the application of their mean value.

7. Conclusions

1. The highest values of the sound power level radiated by the plate occurred for the axially symmetrical resonance vibrations. In the case of the axially nonsymmetrical resonance vibrations, and also the frequency range between the resonances, the values of the sound power level radiated were lower by about 10 dB.

2. It was found from experimental investigations that over the higher frequency range ($k_{mn}a > 8$) a concentration of the resonance frequencies of the plate occurred, causing an increase in the radiation efficiency of the plate at some frequencies above 900 Hz.

3. The results of experimental investigations of the sound power in the free field and reverberant field conditions showed a good agreement over the low

frequency range. Over the higher frequency range the values of the sound power measured in the reverberant field conditions were higher than those measured in the free field conditions.

This might have resulted from the effect of the reflected waves which caused an increase in the vibration velocity of the plate at some frequencies, and also from the radiation of the lower part of the plate and the baffle.

4. The values of the equivalent surface of a circular plate determined for the axially symmetrical vibrations permitted an approximate estimation of the sound power radiated by the plate in the case when the amplitude of the vibration velocity at its centre was known. This calculation was possible for the few first axially symmetrical vibration modes with $k_{mn}a < 5$.

5. For higher frequencies, with $k_{mn}a > 8$, the calculation of the sound power based on the assumption of the value of the vibration velocity at the centre of the plate may involve a high error, particularly in the reverberant field conditions, since the radiation conditions of the plate were affected by the field of the reflected waves.

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