

INFLUENCE OF REVERBERATION FIELD CONDITIONS ON SOUND POWER EVALUATION WITH THE AID OF INTENSITY METHOD

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The results of investigations presented in the article were aimed at explaining how the conditions of acoustic field described by the room absorption and reverberation time can influence the calculated sound power value determined by the sound intensity measurements. As the sound source the rigid piston in the large baffle was used. The measurements of sound intensity and sound pressure were done in the free field and diffuse field conditions. The total absorption of the room was changed.

1. Introduction

The development of the measuring technique and new tools in sound intensity measurements has enabled the elaboration of several methods for sound source localization, investigation of sound propagation paths and measurement of absorbing and insulating properties of materials. Most of the commonly used methods were based on sound pressure measurements due to ease of their application. The evaluation of acoustic parameters with the aid of sound pressure methods required the measurements to be done in conditions of free sound field or diffuse sound field. Only in these conditions the sound power could be calculated precisely. In comparison with sound intensity which was the vector value, the sound pressure gave less information about the sound energy transportation in acoustic fields. For practical application of sound intensity in investigations of sound radiation of machines, particularly important was the exactitude of sound evaluation. From the theorem that sound power can be calculated on the basis of sound intensity had no influence on the estimated value of sound power, but in practice this influence existed and was important. The disturbances of the acoustic field might be caused by waves reflected from the walls, or the acoustic field could be interfered by the waves coming from the external sound sources, e.g. the other machines working nearby. Many authors have tried to solve the problem of influence of the acoustic field conditions. The indices have been introduced, based on the difference of sound pressure level and sound

intensity level measured at the same point of the field [6, 8]. The work was the attempt to establish how the condition of acoustic field described by the room absorption and reverberation time could influence the calculated sound power value using the sound intensity. The results of comparison of sound pressure and sound intensity distribution in the field of plane piston source taken in different acoustic fields were also shown.

2. Application of sound intensity to acoustic parameter evaluation

The sound intensity methods could be used in examinations conducted in all kinds of media: gases, liquids and solids. The largest number of applications was performed in gaseous media, especially in the air. Among the main applications were estimation of sound power of the sound source, radiation efficiency, localization and sorting of sound sources in machines considering activity of their noise emission, investigations of sound propagation paths, measurement of sound absorption and specific acoustic impedance of materials and structures, and also examination of sound insulation properties, such as investigation of sound leakage in composed partitions and machine enclosures. In all the applications mentioned above the sound power was the quantity used for the acoustic parameter evaluation. Figure 1 presents the main applications of sound intensity to evaluate the acoustic parameters.

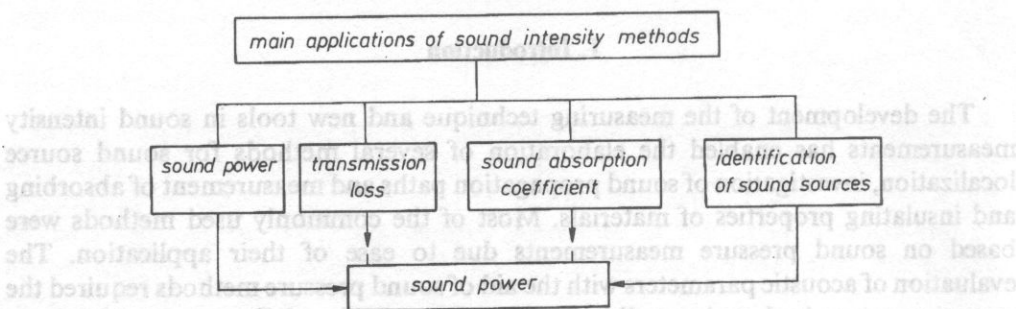


Fig. 1. Main applications of sound intensity in vibroacoustics

3. Sound power evaluation

Evaluation of sound power radiated from the sound source is based on the theorem that integral of the sound intensity normal component to the measuring surface surrounding the source is equal to the radiated sound power:

$$W = \oint_S I_n dS, \quad (1)$$

where S — surface surrounding the source, I_n — value of the sound intensity component normal to the surface S .

In practical cases the integral in Eq. (1) may be replaced by summation

$$W = \sum_{i=1}^k I_{ni} \cdot \Delta S_i, \quad (2)$$

where k — number of elementary fields into which the surface S has been divided, ΔS_i — elementary field. It has been assumed that I_{ni} was the mean value for the elementary surface ΔS_i .

Taking the same value for all elementary surfaces $\Delta S_i = \Delta S$ we obtain

$$W = \Delta S \sum_{i=1}^k I_{ni}. \quad (3)$$

The measurements are usually done for the array of fixed points or by sweeping method. For both methods the derived value of sound intensity is considered to be the average value for the appropriate surface element ΔS_i .

The Eq. (1) indicates that the net flow of acoustic energy integrated over the surface which does not contain sources or sinks is equal zero, even in an environment that contaminated by other sound sources. It implies that the sound power of the source inside the enclosing surface can be precisely determined. In practice, however, contaminating sources make the sound field quite reactive and cause large problems in estimating the sound power.

Statistical theory of pressure acoustic fields claims that the reverberation fields exist in the regions of both the direct sound waves and waves reflected from the walls. The size of direct waves region for chosen sound source depended on the total room absorption. The direct waves field around the source is less reduced the lower the room absorption.

Sound power of a circular piston sound source calculated from the following formula:

$$W = \frac{1}{2} S_z v^2 Z, \quad (4)$$

where Z — acoustic radiation impedance, $[\text{kg m}^{-2}\text{s}^{-1}]$, S_z — surface of the sound source, $[\text{m}^2]$, v — vibration velocity of the piston surface, $[\text{ms}^{-1}]$.

Acoustic radiation impedance of circular plane rigid piston was found from equation [1]:

$$Z = \rho c \left[1 - \frac{J_1(2ka)}{ka} + j \frac{H_1(2ka)}{ka} \right] \quad (5)$$

where ρ — density of the medium surrounding the piston source, i.e. air, $[\text{kg m}^{-3}]$, J_1

$(2ka)$ — Bessel function, $H_1(2ka)$ — Struve function, $k = \frac{2\pi}{\lambda}$ — wave number, $[\text{m}^{-1}]$,

a — dimension of piston, [m].

The real part of sound power was then evaluated from the equation

$$W = S_z v^2 \rho c \frac{k^2 a^2}{4} \left[1 - \frac{J_1(2ka)}{ka} \right]. \quad (6)$$

The results of calculation of sound power levels are shown in Fig. 5.

The sound power of a circular piston sound source was estimated by sound pressure measurements in a free field conditions and in a diffuse field. In the free field, in an anechoic chamber, the values of sound power levels in all examined frequencies were found from the formula

$$L_w = 10 \log \frac{p_m}{p_0} + 10 \log \frac{S}{S_0}, \quad (7)$$

where p_m — average value of sound pressure on the surface S_p , [Pa], p_0 — reference value of sound pressure, $p_0 = 2 \cdot 10^{-5}$, [Pa], S_p — area of measuring surface, [m²], $S_0 = 1$ m²,

$$p_m^2 = \frac{1}{n} \sum_{i=1}^n p_i^2 \quad (8)$$

where p_i — sound pressure value at point i .

In the diffuse field, in a reverberation room, the sound power value was the calculated from the formula

$$W = \frac{R}{4\rho c} p_m^2, \quad (9)$$

where $R = \frac{S_r \alpha_p}{1 - \alpha_p}$ room constant, [m²]; S_r — total area of inner room surfaces, [m²]; α_p — mean value of the sound absorption coefficient of room walls.

For very small values of $\alpha_p < 0.06$ it was possible to approximate (10) by the expression

$$R = S_r \alpha_p = A. \quad (11)$$

Then from (9) and (11) we obtained

$$L_w = 10 \log \frac{p_m}{p_0} + 10 \log \frac{A}{A_0} - 5 \quad (12)$$

where $A_0 = 2 \cdot 10^{-5}$ m².

Pressure methods for sound power evaluation in partially diffuse fields required to establish the kind of acoustic field in which the measurements were done. This establishing was based on several factors calculated from the reverberation time and total room absorption. They show the difference between the existing acoustic field and that under the free field conditions. Similar procedure was proposed for the sound

power evaluation using sound intensity measurements in partially diffuse fields. Although the evaluation of sound power by sound intensity methods seemed to be straight — forward and simple, it had to be preceded by a complicated field control procedure to determine the appropriate parameters of the measurements. This field control was based on a number of field indicators proposed by HUBNER [5, 6]. All these procedures were given in two international standards, still in development [11, 12].

4. Description of experiment

As the sound source was used a rigid plane circular piston placed in large baffle. For that kind of sound source it was possible to calculate precisely the sound power from its theoretical model, as it was shown in Sec. 3.

The piston was forced to vibrate by the exciter with the same constant velocity of vibrations equal to 0.05 m/s for every investigated frequency in the range from 100 to 2000 Hz. A constant value of vibratio velocity was maintained, owing to the application of a feedback with vibration control system of the vibration exciter. Investigations were conducted for harmonic signals of frequency. The piston was prepared to have a sufficient stiffness. It's wave number was equal to $ka = 1$ for the frequency of 775 Hz. To avoid the influence of noise produced by the exciter cooling ventilator, the exciter was placed in an enclosure separated from the baffle. Figure 2 presents the measuring system schema. For such chosen sound source the agreement

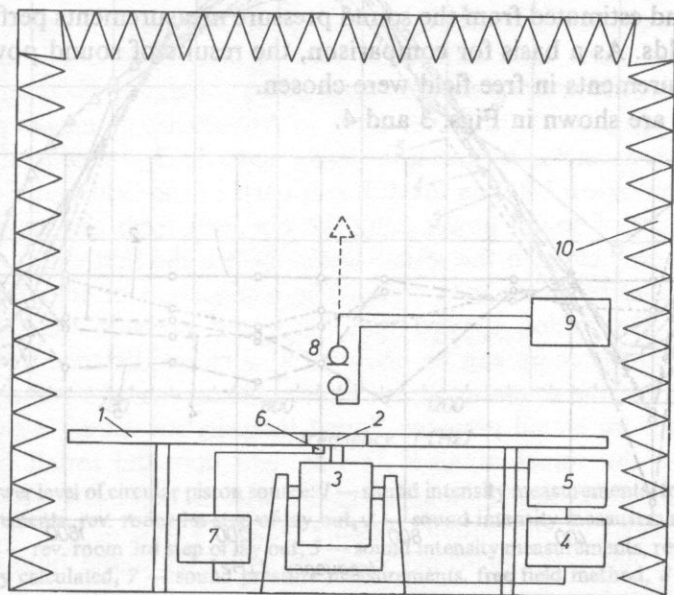


Fig. 2. Measuring system diagram: 1 — baffle, 2 — circular piston, 3 — vibration exciter, 4 — harmonic signal generator, 5 — power amplifier, 6 — accelerometer, 7 — charge amplifier, 8 — sound intensity probe, 9 — sound intensity analyzer, 10 — lay out with randomly distributed damping material.

between theoretically calculated sound power values and those evaluated from the measurements should be expected in the frequency range from 100 to 800 Hz. The investigations were also conducted for higher frequencies only to compare the results in different acoustic fields. The investigation took place in anechoic chamber and in the room with diffuse acoustic field. To obtain different acoustic conditions the total absorption of the room was changed. The changes of absorption were checked by the measurements of reverberation time in each case of layout with randomly distributed damping material. Three measuring surfaces were chosen in the form of parallelepiped. The dimension of the elementary surface ΔS for the first measuring surface were 0.1×0.1 m, and for the remaining measuring surfaces were 0.15×0.15 m. The area of the third surface was 2.72 times greater than the second one and 11.5 times than the first surface. The sound intensity component perpendicular to the measuring surface and sound pressure levels were measured.

5. Results and discussion

A. Comparison of the sound power values

The main aim of the presented investigations was the evaluation of how the acoustic field conditions influence the measured sound power of piston sound source. The results obtained allowed also to formulate the conclusions concerning the influence of the size and position of the measuring surface upon the evaluated sound power. Additionally, the sound power of the piston sound source was calculated theoretically and estimated from the sound pressure measurements performed in free and diffuse fields. As a basis for comparison, the results of sound power evaluated from the measurements in free field were chosen.

The results are shown in Figs. 3 and 4.

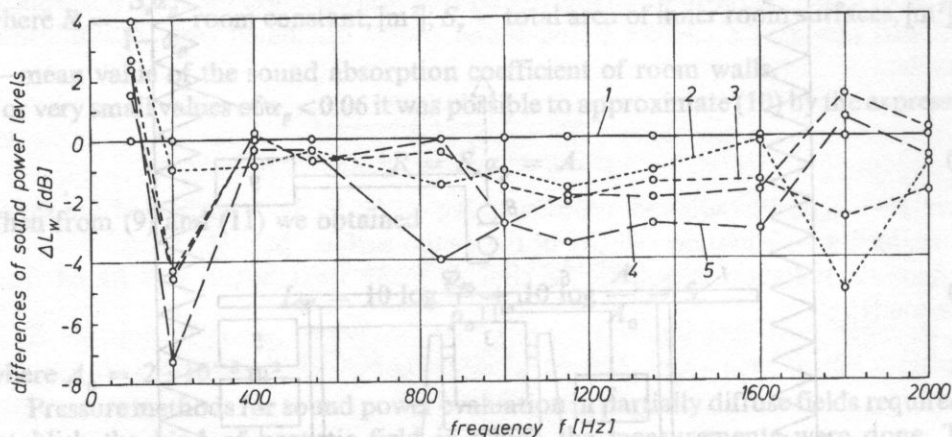


Fig. 3. Differences of sound power level values obtained from measuring surface no 1 in free field and obtained: 1 — in free field (anechoic chamber), 2 — reverberant room (1st step of lay out), 3 — reverberant room (2nd step of lay out), 4 — reverberant room (3rd step of lay out), 5 — reverberant room (with no lay out)

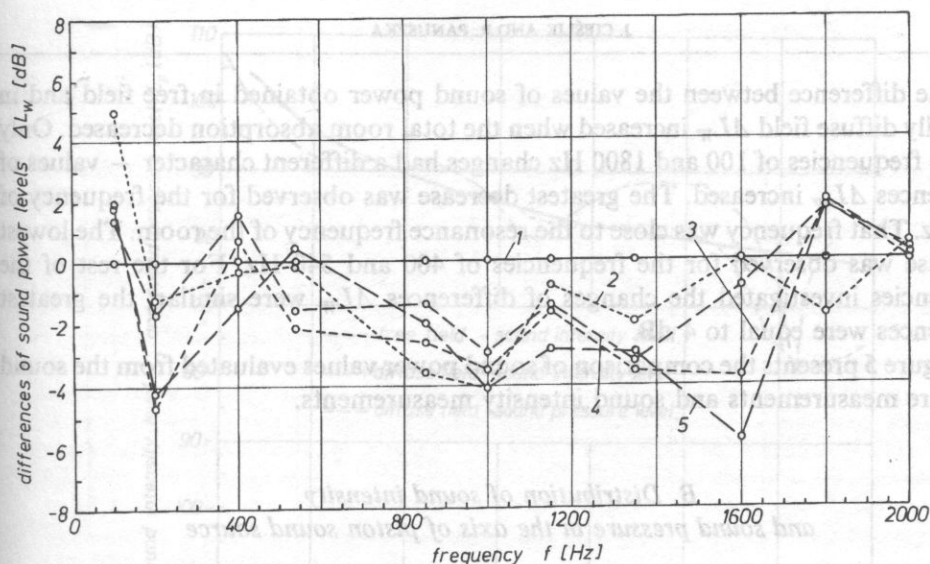


Fig. 4. Differences of sound power level values obtained from measuring surface no 2 in free field and obtained: 1 — in free field (anechoic chamber), 2 — reverberant room (1st step of lay out), 3 — reverberant room (2nd step of lay out), 4 — reverberant room (3rd step of lay out), 5 — reverberant room (with no lay out)

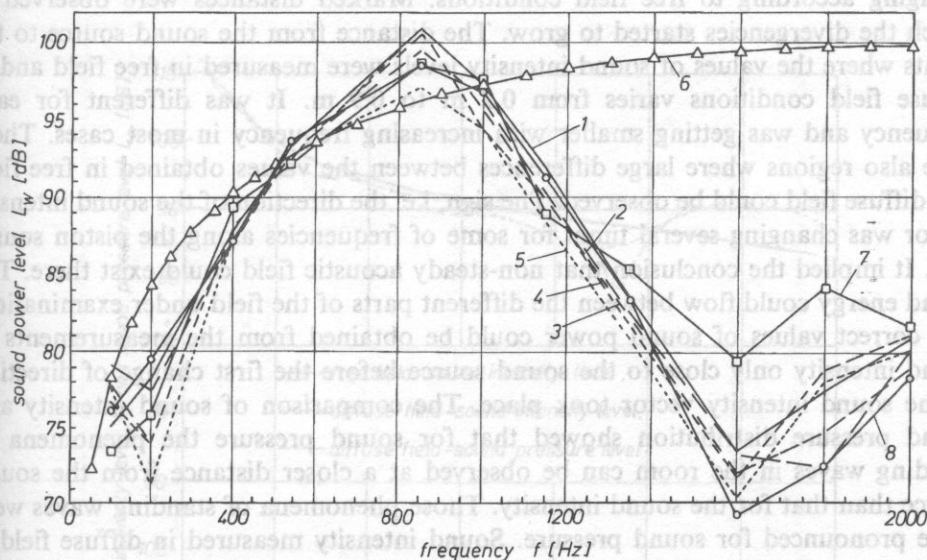


Fig. 5. Sound power level of circular piston source: 1 — sound intensity measurements, free field, 2 — sound intensity measurements, rev. room 1st step of lay out, 3 — sound intensity measurements, rev. room 2nd step of lay out, 4 — rev. room 3rd step of lay out, 5 — sound intensity measurements, rev. room no lay out, 6 — theoretically calculated, 7 — sound pressure measurements, free field method, 8 — sound pressure measurements, diffuse field method.

The difference between the values of sound power obtained in free field and in partially diffuse field ΔL_W increased when the total room absorption decreased. Only at two frequencies of 100 and 1800 Hz changes had a different character — values of differences ΔL_W increased. The greatest decrease was observed for the frequency of 200 Hz. That frequency was close to the resonance frequency of the room. The lowest decrease was observed for the frequencies of 400 and 540 Hz. For the rest of the frequencies investigated the changes of differences ΔL_W were similar; the greatest differences were equal to 4 dB.

Figure 5 presents the comparison of sound power values evaluated from the sound pressure measurements and sound intensity measurements.

B. Distribution of sound intensity and sound pressure in the axis of piston sound source

The results of distribution of sound intensity and sound pressure levels in the axis of the piston for the frequencies of 400, 540, 1000, and 1150 Hz are shown at the Fig. 6 to Fig. 9. The analysis of intensity levels distribution that for each of investigated frequencies there is the region where the values of sound intensity are changing according to free field conditions. Marked distances were observed at which the divergencies started to grow. The distance from the sound source to the points where the values of sound intensity levels were measured in free field and in diffuse field conditions varies from 0.2 m to 0.9 m. It was different for each frequency and was getting smaller with increasing frequency in most cases. There were also regions where large differences between the values obtained in free field and diffuse field could be observed. The sign, i.e. the direction of the sound intensity vector was changing several times for some of frequencies along the piston source axis. It implied the conclusion that non-steady acoustic field could exist there. The sound energy could flow between the different parts of the field under examination. The correct values of sound power could be obtained from the measurements of sound intensity only close to the sound source before the first change of direction of the sound intensity vector took place. The comparison of sound intensity and sound pressure distribution showed that for sound pressure the phenomena of standing waves in the room can be observed at a closer distance from the sound source than that for the sound intensity. Those phenomena of standing waves were more pronounced for sound pressure. Sound intensity measured in diffuse field is usually closer to the values obtained in free field than the sound pressure. The change of direction for sound intensity vector was in most cases connected with a significant change of the sound pressure value. Value of the field indicator calculated as the difference of sound pressure level and sound intensity level, $L_p - L_I$, at those distances rapidly started to increase.

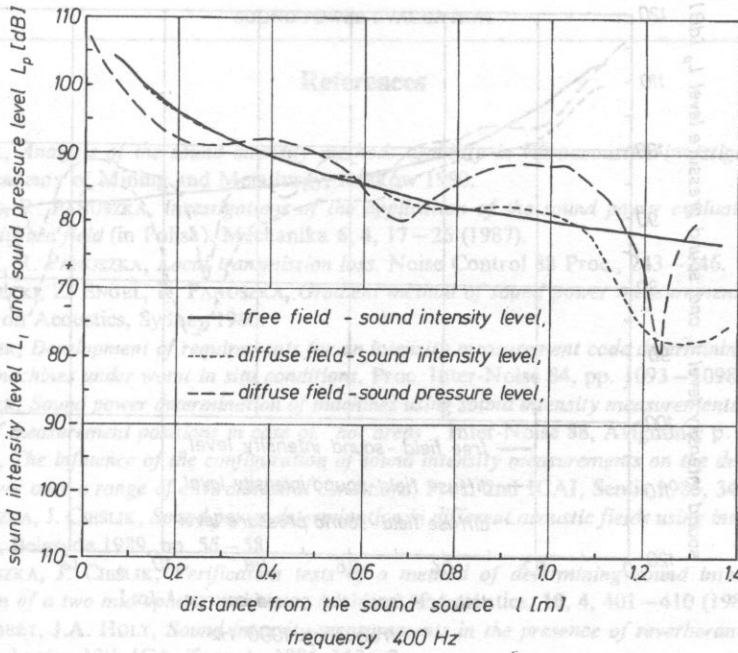


Fig. 6. Sound pressure and sound intensity distribution in the axis of circular piston sound source.

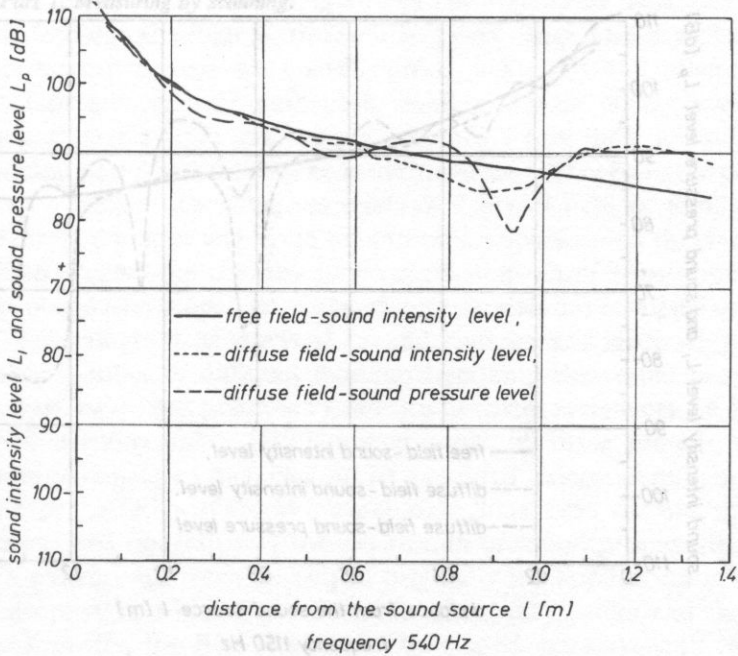


Fig. 7. Sound pressure and sound intensity distribution in the axis of circular piston sound source.

The difference between the values of sound power obtained in free field and in partially diffuse field is small. It increases when the total room absorption decreased. Only at two frequencies (100 and 200 Hz) the values of differences decreased. The greatest difference was observed at the frequency of 200 Hz. That frequency was close to the natural frequency of the room. The lowest decrease was observed for the frequencies of 400 and 800 Hz. For the rest of the frequencies the change of difference was not so great.

Figure 8 presents the comparison of sound intensity values evaluated from the sound pressure measurements and sound intensity measurements.

The results of distribution of sound intensity and sound pressure levels in the axis of the piston for the frequencies of 1000 and 1150 Hz are shown at the Fig. 6 to Fig. 9. The analysis of the distribution that for each of

investigated frequencies the sound intensity levels were measured in free field and in diffuse field. The results of sound intensity levels were measured in free field and in diffuse field are shown in Fig. 6 to Fig. 9. It was difficult for each frequency and distance smaller with increasing frequency in most cases. There were also regions where the differences between the values obtained in free field and diffuse field were large.

It is concluded that non-steady acoustic field exists in the axis of the piston source. It implies that the sound energy is not uniformly distributed in the axis of the piston source. It implies that the sound energy is not uniformly distributed in the axis of the piston source. The correct value of sound power could be obtained from the measurements of sound intensity and sound pressure close to the sound source. The first change of direction of the sound energy is observed at a closer distance from the sound source than the sound pressure. The sound pressure is usually closer to the value of the sound intensity level, L_p . The change of direction of sound energy is observed at a closer distance from the sound source than the sound pressure. The sound pressure is usually closer to the value of the sound intensity level, L_p . The change of direction of sound energy is observed at a closer distance from the sound source than the sound pressure. The sound pressure is usually closer to the value of the sound intensity level, L_p .

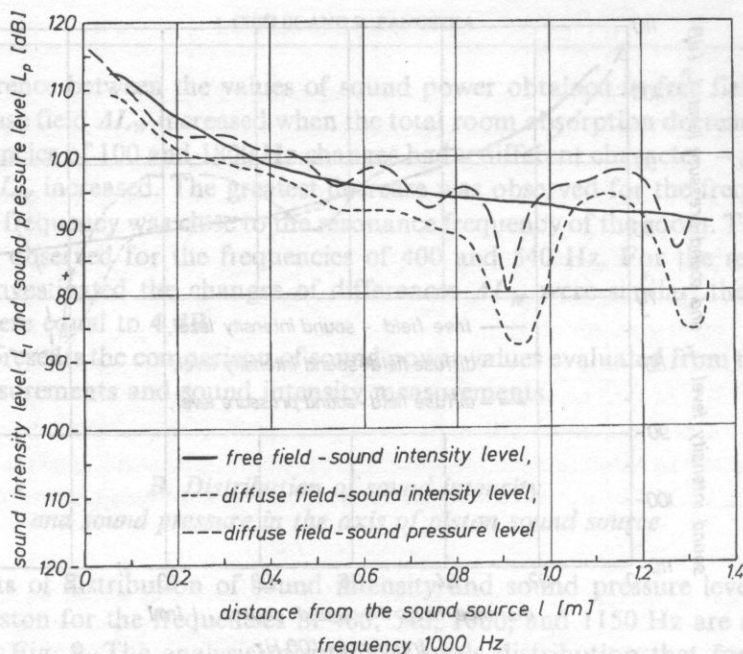


Fig. 8. Sound pressure and sound intensity distribution in the axis of circular piston sound source.

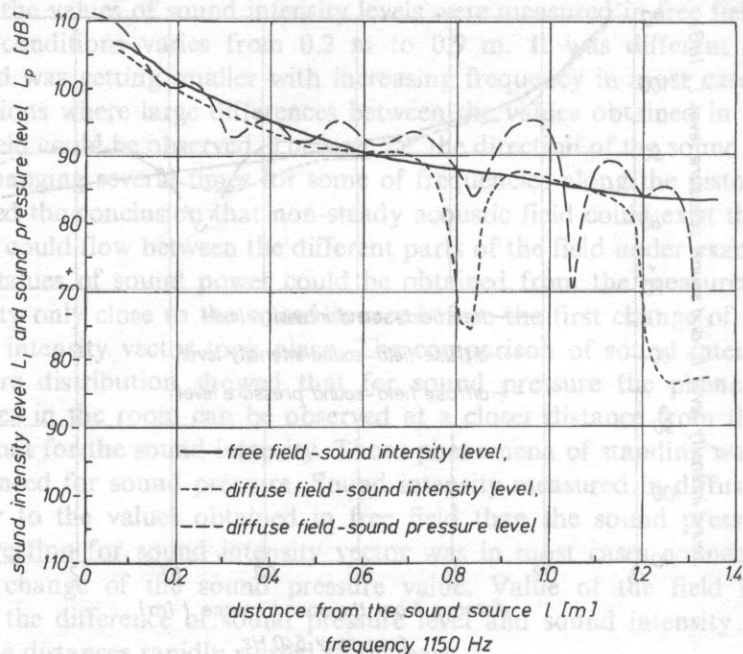


Fig. 9. Sound pressure and sound intensity distribution in the axis of circular piston sound source.

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Due to the continuing technological progress, research has been done and is still done in the motorcar, the aerospace and the manufacturing industries and others, to obtain absorbing materials who are simultaneously small in volume, have a low density, are rather cheap and have a good acoustic performance. From the other side it is often desirable to optimize the behaviour of acoustic materials for very specific purposes. To achieve highly qualified absorbing materials in a broad field of applications it is necessary to develop precise and, from time to time, sophisticated theoretical models which take into account the complex physical phenomena of acoustic materials.

The study of acoustic properties of absorbing materials has occupied scientist since the work of BERANEK [1, 2], MORSE et al. [3] and ZWICKER and KOSSEN [4] in the early 1930's. A large number of different models, describing the sound propagation in porous materials, have been published since. An excellent review can be found in the publication of ATTENBOROUGH [5]. However most of these models require the introduction of parameters which can not be measured independently, making them inappropriate for the design of absorbing materials. Since 1956, the Biot theory allows in a very general and rigorous way the description of sound propagation in porous materials. This theory has been developed originally for application in the field of underwater geophysics, where the densities of the fluid (water) and the solids are comparable. Recently, the Biot theory has been used extensively to calculate the acoustic properties of porous materials used in noise control engineering, where the