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CONSTRUCTION OF AN ANECHOIC UNDERWATER SOUND MEASURING TANK

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ABSTRACT

This is a translation of an article in Acustica, vol. 10, 1960, pp. 281-287. The article describes a water basin with dimensions 7 m × 4 m × 4 m, which was constructed for measurements with water-borne sound. The walls of this basin are coated with absorbers effective in the frequency range from 5 to 70 kc. These broadband absorbers consist of a system of parallel, wedge-shaped rubber plates made up of three layers of rubber glued together. The center sheet is perforated with circular holes (diameter 4 mm, hole density 4%). There are three types of wedges differing in length (7, 15, and 20 cm) covering the frequency range from 5 to 70 kc. The reflection factor related to amplitude remains below 10% in this frequency range. The excellent acoustical properties of the anechoic measuring basin are confirmed by the very small standing wave ratio for all frequencies.

ADMINISTRATIVE INFORMATION

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CONSTRUCTION OF AN ANECHOIC UNDERWATER SOUND MEASURING TANK

1. INTRODUCTION

A tank for free-field measurements of underwater sound was constructed in the new addition of the Third Physical Institute of the University of Göttingen. This tank has surface dimensions of 4 m × 7 m and a depth of 4 m. In order to perform measurements without undesirable reflections from the boundaries, it is necessary to make the walls of the tank, as well as its bottom and surface, reflectionless. To achieve this one needs a broad-band absorber, corresponding to the dimensions of the tank, which causes sufficiently small reflections in the frequency range from about 5 kc up to high frequencies.

Wedges constructed of porous material have been in use for quite some time for anechoic lining of chambers designed for measurements of air-borne sound. The pores of these wedges possess an appropriate propagation impedance such that the particle velocity of the sound field experiences losses and, consequently, damping of the wave occurs. In water, which has a high acoustic impedance, this principle is less effectively applied since, because of the resonance of the material structure in water, the relative motion between the medium and the absorbing material and, consequently, the acoustic damping are strongly reduced. Here the material must have a preferred response to the pressure component of the sound field; the governing facts in this process are the losses arising during the elastic deformation. As a rule rubber-like elastic substances are used, and their acoustic properties are matched reasonably well with those of water. In the case of underwater sound absorbers [1] built to date, essentially two approaches are taken:

1. Alteration of the compressibility by admixture of fillers such as saw dust, cork dust, and others.

2. Alteration of the compressibility by the introduction of macroscopic cavities into the otherwise homogeneous absorption material.

The second method has the advantage of being exactly controllable and reproducible. Examinations of the acoustic behavior of a single cavity in a large piece of rubber [2] have shown that such a cavity has a definite natural frequency of oscillation, that the latter is easily calculable, and that damping of the system is rather high. The absorber below is based on large suitably distributed cavities in rubber.

2. THE ABSORBING MATERIAL

The basis of the absorber is a plate consisting of three layers glued together. The central layer has circular holes punched into it. After bonding, the areas covering the holes above and below act as plates capable of oscillating. If their circumferences are firmly held in place, they have a natural frequency given by

$$\nu_{m,n} = \frac{\pi d}{4a^2} \sqrt{\frac{E}{3\rho(1-\sigma^2)}}\ \beta^2_{m,n},$$

where d is the thickness of the plate, r its radius, E the modulus of elasticity, σ the transverse contraction number, and ρ the density; $\beta_{m,n}$ are the solutions to the differential equation describing the plate oscillation, the first few values being

$$\beta_{01} = 1.015; \quad \beta_{11} = 1.468; \quad \beta_{21} = 1.879.$$

If the plate thickness becomes comparable to the radius of the hole, the natural frequency of oscillation will approach a limiting value which is no longer dependent on the plate thickness. Thus a vibration of the cavity is achieved like that examined more closely in the paper [2] referred to in the introduction.

The losses are essentially determined by the loss factor of the shear modulus, which is large in the case of rubber-like elastic substances. For this reason and for reasons of stability in water, rubber is especially suited for the production of such an absorbing material.

As an example of a measurement, Fig. 1 represents, in the complex plane, the impedance Z (related to the wave impedance $\rho \cdot c$ of water) of such a plate for the case of vertically incident sound. The

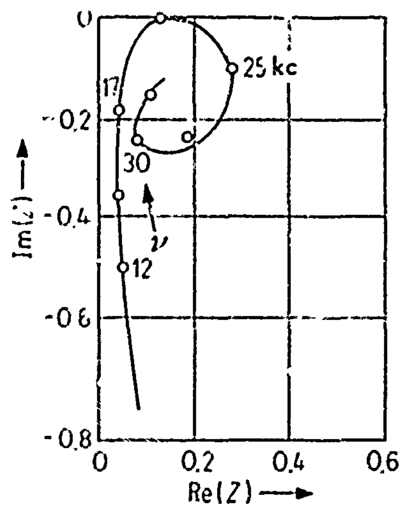

Fig. 1 - Impedance of a Perforated Plate (hole diameter 3 mm) for the Case of Perpendicular Incidence of Sound, Represented in the Complex Plane of Impedance

holes have a diameter of 3 mm; the number of holes in this example amounts to 5 per square centimeter. As a result of attenuation, the resonance curve is very broad. Even if not particularly well, a matching to the surrounding medium can only be achieved near the resonance frequency (about 25 kc).

3. THE PARALLEL-PLATE SYSTEM

In order to obtain a broad-band absorber based on perforated rubber plates, one has to arrange these in a parallel-plate system according to the "Fafnir" method [1] in such a way as to cause sound to impinge on the array parallel to the plate surfaces and to penetrate the channels between the plates. According to theory [3], the wave entering the channel decays exponentially, the energy being absorbed by the walls made up of such dissipative plates.

Besides being determined by the impedance of the walls, the amount of damping of the wave is essentially governed by the dimensions of the channel, the ratio of channel width to wavelength of the incident sound being the decisive factor.

Such a parallel-plate system was set up in a shallow tank for testing purposes. It consists of a number of perforated rubber sheets arranged next to one another in parallel fashion, thus forming a row of channels on which sound rays fall perpendicularly. A probe-type receiver is moved through the central channel perpendicularly to the row, that is, in a fashion parallel to the walls, to record the sound pressure behavior in front of and inside the channel. A SELL-transducer with a 7 mm diameter diaphragm was used as a receiver. The measurements were carried out with continuous tones.

Figure 2 represents two examples of sound pressure curves recorded with the test microphone. A normalized pressure amplitude is drawn as a function of position. The boundary between the free water and the parallel-plate system distinctly divides the curve into two regions. In front of it a standing wave is created by superposition of the incident wave with that reflected at the boundary layer. In the interior of the channel there exists a wave exponentially decaying along its direction of propagation.

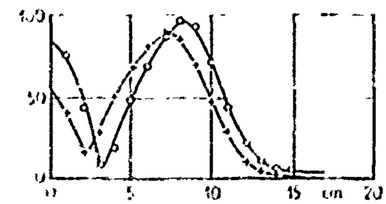

Fig. 2 - Examples of the Sound Pressure Flow in Front of and Inside the Parallel-Plate System for Two Frequencies

Damping in the interior of the channels (db/cm) was thus determined from many curves of this kind, and the reflection coefficient of the array was computed from the standing wave ratio.

In Fig. 3 some results of the measurements are presented. The following parameters were varied: size of the holes (3, 4, and 6 mm); density of the holes, that is, the ratio of the surface area of the holes to the total surface area of the plates (7, 10, and 37%); the material of the plates and the channel width (2 to 4 cm).

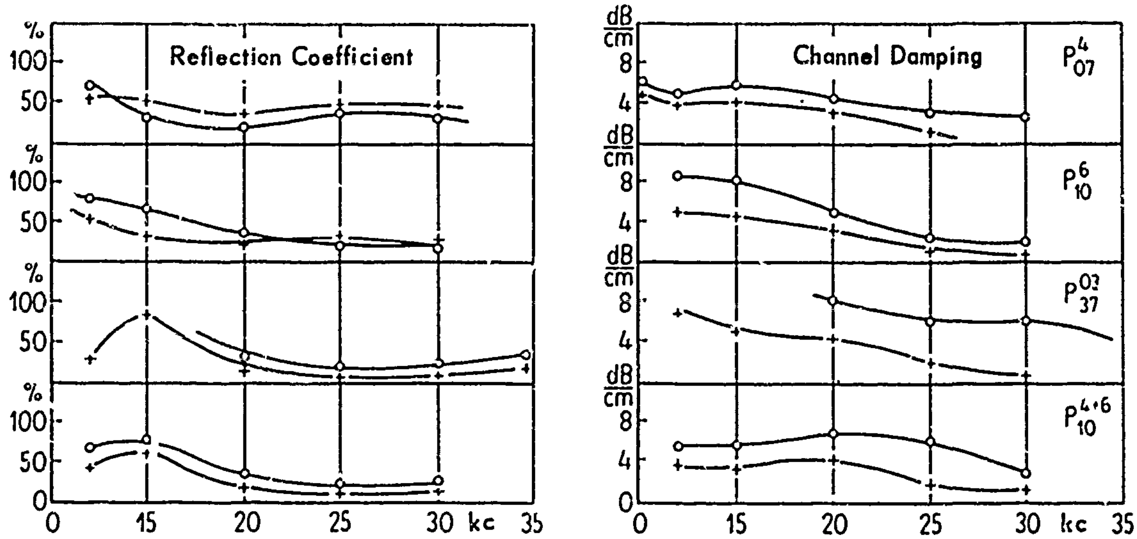

Fig. 3 - Magnitudes of Channel Attenuation and Reflection Coefficient of Different Parallel-Plate Systems, P^m_n, as a Function of Frequency

o Plate Separation 3 cm
+ Plate Separation 2 cm

Upper Plate Index m → Hole Diameter
Lower Plate Index n → Hole Density

If channel damping is plotted as a function of the ratio of channel width to wavelength b/λ, the results can be well arranged (Fig. 4). As a consequence of the high damping of plate oscillations, the

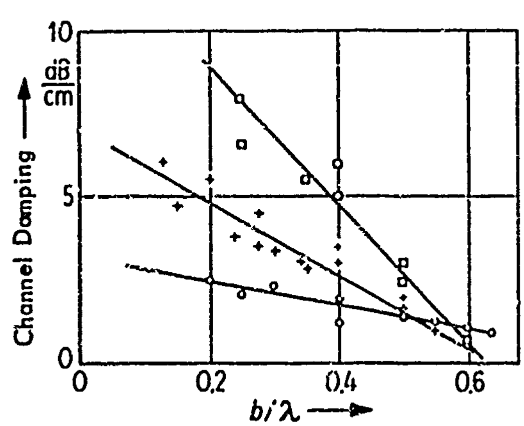

Fig. 4 - Sound Damping in Channels of Parallel-Plate Systems, P^4_{04}(o); P^4_{07}(+), P^3_{37}(□), as a Function o. Channel Width b/λ Related to the Wavelength λ

resonant frequencies in the holes are strongly suppressed so that it is the geometrical properties of the channel that essentially determine the amount of damping. The scattering of the measured values about the straight line, which correspond to the decrease of damping with increasing b/λ, is caused by the specific frequency dependence of each individual plate vibration over the cavities.

The three systems of plates represented here essentially differ in the density of holes. Damping increases with increasing hole density since a larger portion of the total plate surface is set into vibration. If one approximates the law that describes the attenuation of the wave in the channel in terms of b/λ by the function

$$D = A\left(B - \frac{b}{\lambda}\right)$$

in the interval $0.1 < b/\lambda < 0.7$, then the factor A expresses the effect of the hole density. With increasing hole density A approaches 40 % as a limit (Fig. 5).

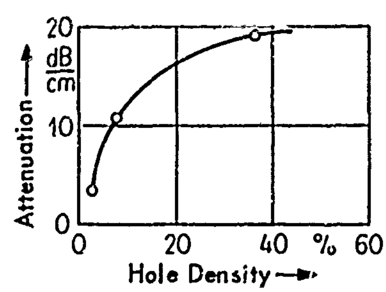

Fig. 5 - Effect of Hole Density (in %) on Attenuation in a Parallel-Plate System

As the thickness of the covering layers increases, the suppression of the individual natural frequencies associated with the holes becomes stronger and stronger, and the approximate formula above is better satisfied. This is shown in Fig. 6 by a comparison of two plate systems

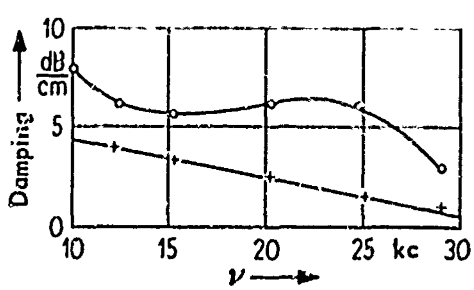

Fig. 6 - Effect of Layer Thickness on the Frequency Response of Attenuation in the Channel

o 2 mm
+ 3 mm layer thickness

made up of sheets of difference thickness (channel width 2 cm, hole diameter 5 mm).

Most of the results shown in Figs. 2 to 6 were derived from rubber plates. The investigations also covered rubber-like plastics, but rubber proved to be superior in both its damping capabilities and, in particular, its stability in water.

4. CONSTRUCTION OF THE ABSORBER

As shown in the preceding paragraphs, the parallel-plate system represents an effectively absorbing "medium" which, with proper dimensioning, causes a high attenuation in the penetrating wave for a wide range of frequencies. It is important in the construction of the absorber that, besides being able to attenuate sound energy, the absorbing array be acoustically matched to the free medium in an optimal fashion so that front reflections caused by the differing propagation impedances of the two media remain small.

For broad-band absorption this matching is best achieved by a gradual transition between medium and absorber. If the transition region has a thickness of the same order of magnitude as the wavelength of the incident sound, the discontinuity in the propagation impedance which would normally cause a strong reflection now vanishes.

The goal in the present case was to obtain an absorber for the frequency range from 5 to 70 kc, having a reflection coefficient $r < 10\%$ over this band. According to the results of Section 3, this large frequency band cannot be effectively damped with the use of a single channel width. Instead, it is necessary to construct an array whose channel width changes stepwise so that the formation of unattenuated rays is also avoided for high frequencies. The gradual changeover is effected by "serrating" the rubber plates.

Further measurements have indicated that, because of their high sound attenuation, plates with a large density of holes are undesirable. No sufficient suppression of frontal reflection can be achieved even by gradating the transition region between the media as a result of the large decrease of impedance caused by the high compressibility of the plate material. A plate with a hole density of only 4% (hole diameter 4 mm) proved most convenient.

The exact dimensions of the absorber, whose final construction is shown in Fig. 7, were arrived at by measurements in a shallow tank. In this process the reflectivity of a plate system 1 m wide was measured by means of a pulse response set-up and was compared with the reflectivity of an air pocket of the same size.

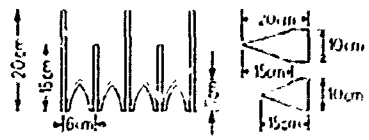

Fig. 7 - Design and Dimensions of the Broad-Band Absorber

To clarify the performance of the structural features of the absorber shown in Fig. 7, the damping $D(x, \lambda)$ of the wave is illustrated in Fig. 8 as a function of its penetration x into the absorber for different frequencies ν and the associated wavelengths λ.

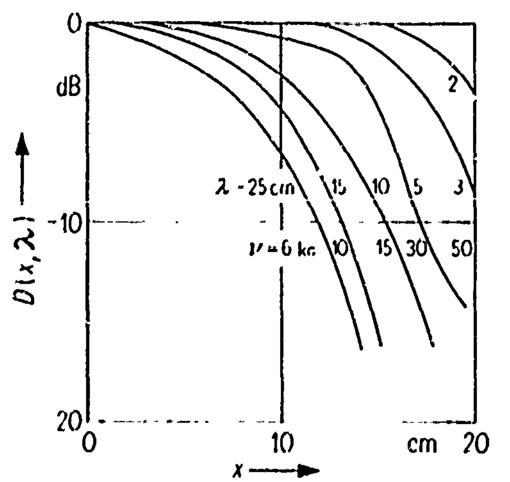

Fig. 8 - Damping of Wave Entering the Absorber as a Function of Distance Covered (0 cm: Front Edge of Absorber.) Parameter: Wavelength or Frequency of Sound Wave

It is clearly seen that with increasing frequency the front part of the absorber having wide channels becomes less and less effective. The "center of gravity" of the attenuation thus moves toward the inner part of the absorber with increasing frequency.

Figure 9 shows the measured reflectivity of this absorber. Curve (a) was recorded in a shallow tank and is supplemented by a measurement on sound spreading three-dimensionally in a deeper tank (b). In both cases the ordinate represents the reflection coefficient related to amplitude. Measurement (b) had been intended

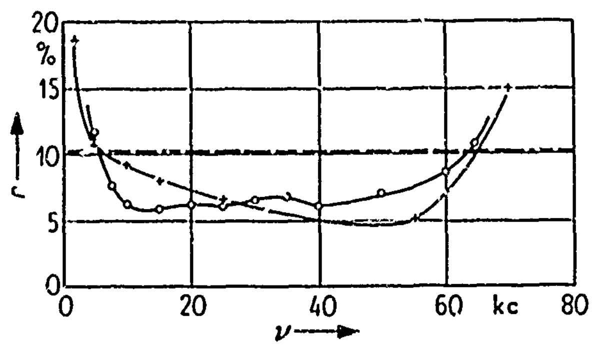

Fig. 9 - Reflection Coefficient of the Broad-Band Absorber According to Fig. 7 as a Function of Frequency

o Measurement (a) in Shallow Tank
+ Measurement (b) in Deep Tank

to confirm the transferability of the shallow tank measurements to the free field, but it served simultaneously as a test measurement on the absorption material which had meanwhile gone into production.

In lining a deep underwater sound tank with absorbing material, one is not only concerned with the reflection coefficient for sound of perpendicular incidence. Since the sound emitted by a transmitter usually strikes the walls at all kinds of angles, multiple reflections may influence the measurements. For this reason the angular dependence of reflection and the scattering strength of the absorber were investigated for the case of oblique incidence.

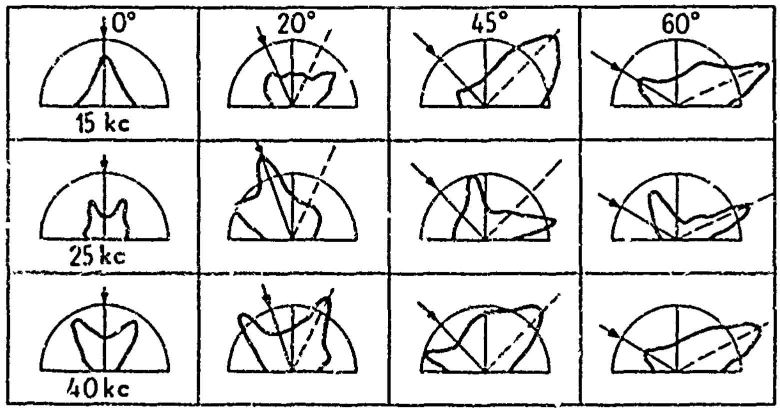

Fig. 10 - Scattering and Geometrical Reflection of the Broad-Band Absorber (of Fig. 7) for Different Angles of Incidence and Frequencies. The Semicircle Corresponds to a Reflection Coefficient of 10%

Figure 10 shows the results of these measurements. The reflectivity, as related to amplitude, generally remains below 10% (semicircle). As a result of the grid-like structure of the absorber, scattering can occur in preferred directions which do not coincide with the geometrical reflection. For very oblique incidence, especially for angles $\varphi > 60°$, the geometrical reflection exceeds the 10% limit by a small amount. As is evident from Fig. 10, all undesired reflections, however, remain small in general and only exceed $r = 12\%$ in exceptional cases.

5. LINING OF THE UNDERWATER SOUND TANK

For lining the measuring tank 116 square meters of absorbers were used on the bottom and the four walls. The water surface was partially covered with an additional 20 square meters of floating absorbers.

The individual absorber parts were produced by different companies. Rubber of the type "sealing sheet" made by the Continental Company at Hannover was utilized. The perforation of the inner sheet and the cutting of the wedges were done by the I. Rehm Company in Peine. The bonding was carried out by H. Lutze of Einbeck. It was especially important to have homogeneous, air-free bonding. Measurements on sample pieces revealed that those bonds made with Continental's cold-bonding cement were satisfactory in every respect; subsequent hydrostatic compression of the absorbers was unnecessary.

Single strips of rubber wedges were then glued into frameworks made of hard polyvinyl chloride, and these were installed in the tank in square units 0.5 m long on edge. In Figs. 11 and 12 the structure of an absorber unit and the partially lined tank can be seen.

Fig. 11 - Picture of an Absorber Unit
(50 cm × 45 cm)

Left: Front View
Right: Rear View

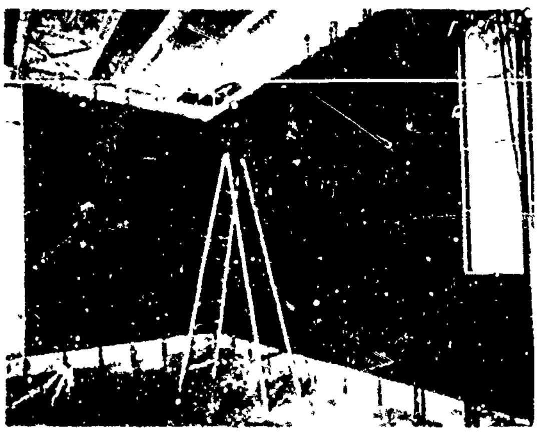

Fig. 12 - The Partially Lined Measuring Tank

6. TESTING OF THE TANK

To test an anechoic chamber, one generally investigates deviations from the 1/r-law, that is, deviations from the sound pressure decrease associated with a point source in free space. This procedure can not be applied to this tank for two reasons:

1. There are no spherical sources of sufficient power in the frequency band from 5 to 70 kc. The decrease in sound pressure with distance is determined by the frequency-dependent directivity characteristic of the source if the receiver is not moved exactly along a lobe center line.

2. Being only partially covered with absorbers, the water surface acts like a mirror. Thus a field of interference between direct and reflected sound rays is created.

For these reasons, in testing this tank, the source was set up close to the surface in such a way that its major lobe pointed vertically downward. This way, the microphone only receives the direct ray and reflections from the absorbent walls. The very small lithium sulfate microphone is lowered with constant velocity from a point next to the projector toward the bottom of the tank.

The acoustic quality of the water tank is measured in terms of the ratio of standing pressure waves caused by interference between the direct ray and reflections from the walls. Figure 13 shows the

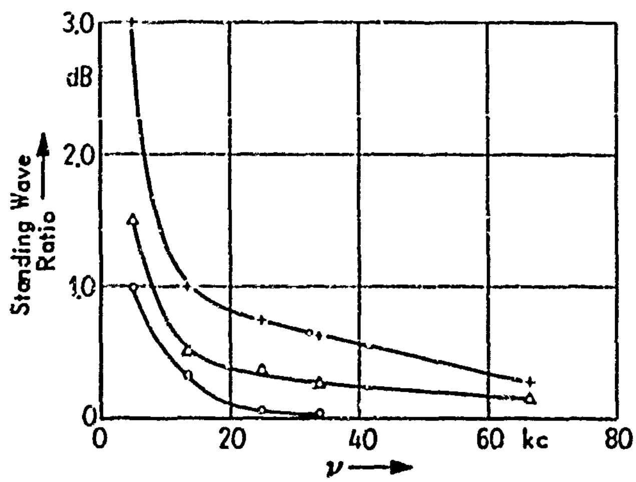

Fig. 13 - Standing Pressure Wave Ratio in the Anechoic Tank as a Function of Frequency. Parameter: Distance between Transmitter and Receiver (+.. 3 m, Δ.. 2m, 0.. 1m)

behavior of the standing-wave ratio (in dB) as a function of frequency between 5 and 70 kc, the parameter being the distance between transmitter and receiver. At a separation of 3 m the microphone is close to the bottom of the tank. The curves clearly show that the lower frequency limit of the absorber lies at 5 kc. Between 5 and 10 kc the standing-wave ratio drops steeply for all separations. These measurements were performed with a continuous tone.

Figure 14 shows examples of pressure level recordings. Superimposed on the standing-wave ratio of the sound field is an additional fluctuation of long period caused by the fact that the sound receiver is moved through the lobe structure of the projector not along the center line.

Curves 14 d-h were recorded for pulses of high repetition rate. Thus, fluctuations of long period are somewhat suppressed. The standing-wave ratio also vanishes in the same measure so that one may read from these curves the pure rate of decay of sound pressure with distance.

Even better results can be expected for measurements along the space and wall diagonals of the tank since, in such a set-up, the first main lobe reflections from the walls do not strike the receiver. However, such measurements can only be

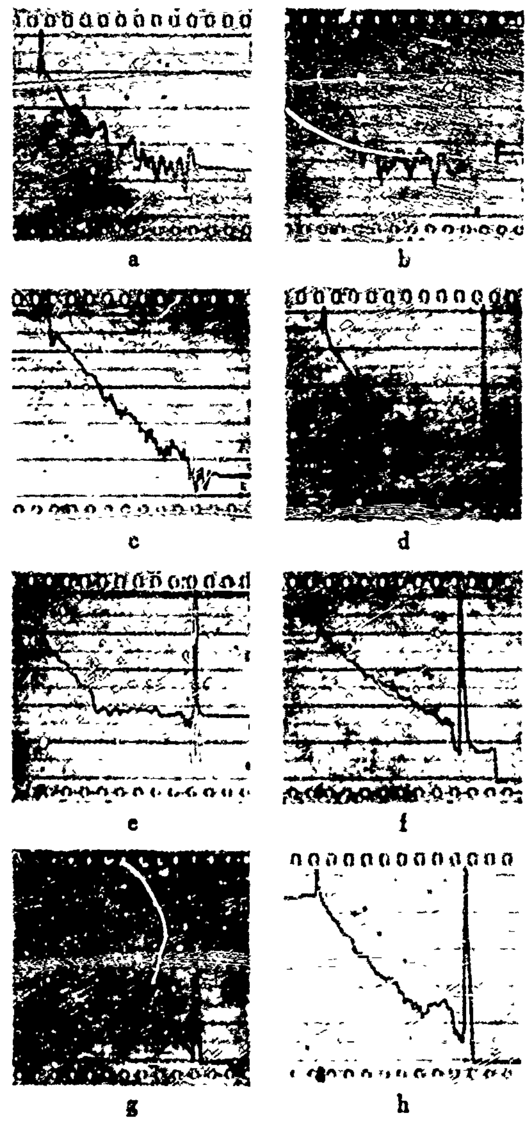

Fig. 14 - Recordings of Sound Pressure Level as a Function of Source-to-Receiver Distance at Various Frequencies

a, b, c - Continuous Tone

d, e, f, g, h - Pulse Train

performed if the entire water surface is covered with absorbing pads, which was not yet the case at the time of the above measurements.

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